THE AUTOMOBILE ENGINEER

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A TECHNICAL JOURNAL DEVOTED TO THE THEORY AND PRACTICE OF AUTOMOBILE CONSTRUCTION.

VOLUME I.

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INTRODUCTORY.

MANY times it has been said that the rapidity of development of the Automobile Industry has been unprecedented in history, and it is a truth which requires no demonstration. But probably few men, besides those who have actually been engaged in the production of automobiles from their first introduction into this country, realise to the full how great has been the growth of correlative knowledge.

Engineering was once a single science, and an engineer could know it in its entirety, but to-day there may be as wide a difference between one engineer and another as there is between a diver and an aeronaut.

Ship building, bridge building, machine tool making, locomotive work, and civil engineering all vary so widely as to require special and quite distinct courses of training. In the same way the science of automobile construction is a distinct and separate entity, demanding of a man years of specialised study before he can hope to have a thorough grasp of it.

The literature of the older branches of engineering is vast, if taken in the aggregate, but newer sections are not so well provided for as older ones. Especially is this so with regard to periodical publications, for though there is a plentitude of journals devoted to the interests of motorists, not one of them can devote full attention to purely technical matters.

There are so many calls upon their space that they can only give room for technical news or discussions intermittently, and when they do so it is necessary for the articles to be written in a form which will render them understandable to the ordinary reader, wherefore the mathematics or the science upon which their argument depends must be deleted, or reduced to the simplest language.

Other journals which are devoted to other branches of engineering have not space to deal adequately with automobile design and construction, each having its own peculiar interest, which must always come first.

Thus we are confident that in introducing *The Automobile Engineer* we shall procure the heartiest support of all those to whom the theory and practice of automobile engineering are of interest and of importance. We do not confine ourselves to Great Britain alone, but anticipate confidently a welcome in every automobile works throughout the world.

Our aim is to become the first authority on the building of self-propelled vehicles, whether they be for use on road or on rail, in air or in water. We shall give publicity to all research which appears valuable, and shall afford facilities for the threshing out of controversial technical subjects. It will also be our endeavour to describe and criticise current designs with a thoroughness hitherto unattempted, which we are able to do by our knowledge of the subject from an engineer's point of view. In short, we shall assist engineers towards the production of ever-improving automobiles by ever-improving means.

By publishing, discussing, or criticising each new idea, by making known to designers what their colleagues are employed upon, by linking together the ideas of England, France, America, and Germany, we shall take an important place in the future history of automobilism.

THE ARCHED BACK AXLE.

IT would be interesting to know how many patents have been granted for inventions connected with automobiles, and still more instructive to know what proportion of them have come into use. New industries have always exercised a peculiar attraction for inventors, and particularly those of that numerous class who possess great ingenuity and but little knowledge of the principles of mechanics.

The ability to invent is undoubtedly something quite apart from education. The most learned men need never have a single original thought, and the most obscure and commonplace people are the more likely of the two to hit upon one of those money-making ideas, such as a new window catch or a popular puzzle, but with machines as intricate as internal-combustion engines and power transmission systems, a valuable discovery is more likely to be made by a clever engineer of average intuition than by the most brilliantly ingenious individual lacking mechanical training.

Of course there are notable exceptions to this generalisation, but, broadly speaking, the best inventions are found to be the work of highly-trained men, and the worst those of pure ingenuity. Naturally, by far the greatest number of patents are neither very good nor very bad, but lie between the two, and in this indifferent class there are to be found many devices which depend to a great extent upon their precise construction for their success or non-success.

Some of these are the work of untrained ingenuity, and some of training without ingenuity. As an example of the first class the original Knight engine patent is excellent. The cleverness

of the design shown therein was undeniable, but a successful engine could hardly have been built from it. It is not too much to say that the present position of the Knight invention is due to the training which the inventor acquired after obtaining the first patent, which enabled him to make it practical.

On the other hand there is a striking instance of the practical success of a bad idea well carried out to be found on more than one car at the present time, and that is the construction known as the arched back axle.

Arched back axles are in use by an increasing number of automobile makers, presumably because the original one was well made, and therefore satisfactory in use. The arched axle, if equally well made, is no worse than the straight one, and it has so happened that so far it has been adopted only by those firms who can and do make all their parts excellently. The arched design is naturally more costly to carry out than the straight one, owing to its more complicated nature, but it has no inherent weakness beyond that there are more parts in it.

It can be called a bad design because it is not a necessary one, and not even advantageous in the least degree, and it is absolutely amazing that the fallacy underlying it has not been perceived by the designers of the otherwise fine cars in which it is to be found.

Two reasons are given by the mistaken advocates of the arched axle : one is that an arch is stronger than a straight beam of equal weight, and the other that with an arched axle dished wheels can be used. We do not, of course, deny the truth of this last statement, but the argument for the dished wheel is just as fallacious as that for an arched axle *per se*.

The strength of an arched bridge depends upon its abutments ; that is, upon the masses of earth or masonry which flank it, because before the centre of the arch can fall under either its own weight or an imposed load, it must lose some of its curvature, and in order to straighten it, must force apart the points upon which it rests. That is to say, an arch exercises a thrust in a horizontal direction against its abutments, and if the latter be removed the strength of the arch, as an arch, is removed with them. The axle of a car cannot therefore be made stronger by arching it, as it has nothing to thrust against.

Then as regards the dished wheel the most common argument in its favour is that it is stronger against side stresses than a flat wheel of equal weight. This is quite true, and if it is assumed that cars always take corners at such speeds that the side thrust is resisted by the outside wheels only, then no doubt the dished wheel is an excellent thing, but as there are many circumstances where severe side stresses are imposed on wheels from the inside outwards (as for instance when taking a curve with only the inner wheel gripping through the condition of the tyres or the road), and as the dished wheel is just as much weaker than the flat one in one direction as it is stronger in the other, the sum total of advantage diminishes to vanishing point. Even if this were not so a little extra weight in the wheels, or the use of wire wheels, would be much cheaper, much more simple, and even more efficient, because there must be some extra frictional loss in the division of the drive, and there is also the staggered spoked wheel, which would appear to be the obvious way out of the difficulty, if great strength and wooden wheels are both desired.

The ability to design a satisfactory arched back axle is proof of sound mechanical knowledge and we regret the waste of brain effort it represents much more than the lack of appreciation of the fallacy of its principle.

THE OVERLOOKED OBVIOUS.

It is the clever people that tend to overlook the obvious, and certainly motor designers as a class comprise more than the average number of brilliant men. If anyone doubts this fact, he need only look at the extraordinary mechanical development that has been wrought in the self-propelled road vehicle since its comparatively recent inception—in fact, it is no doubt due to the circumstances, that have necessitated a vast amount being done in such a short time, that certain of the least essential points have till now been left more or less to take care of themselves.

Take, for instance, the way in which such details as spring hanger brackets or footboard irons are secured to the frame. It is quite the exception to find these so designed that the securing bolts or rivets pass through the frame only on the neutral axis—in fact, the holes more often than not are placed close to where the stress is at its maximum, and where the homogeneous material of the frame would be of greatest value, yet it would not be difficult to design many of these details so that they might be secured to the frame by bolts or rivets passing only through the neutral axis of the beam. It is not contended that such a course would enable any material reduction of weight to be effected, although it is more than conceivable that it might render possible the use of slightly thinner material for the frames, and thereby facilitate work in the pressing, while the material would be subjected correspondingly to less pushing treatment.

The mention of footboard irons in the foregoing suggestion brings up another possibility that appears to have been overlooked by most car designers, for it never seems to have occurred to many that the frames taken in conjunction with these step hangers offer combined elements splendidly adapted to the truss form of construction, and it would be a simple matter to design the hanger so as also to serve the purpose of a truss post. Such a construction need not stand in the way of accessibility, and would certainly make for greater lightness and strength, while involving but little extra cost.

Another instance of the same tendency to accept the practice of the past for present-day application is seen in many of the radiators of to-day for the thermo-syphon cooling that is being adopted so increasingly. Although the designer is working with a principle absolutely different from that to which he has been accustomed, in too many cases little or no modification is made in the radiator, and the tank at the top remains the same size as when pump circulation was employed, regardless of the fact that the system will cease to work once the level of the water falls below the level of the return pipe.

Other instances on similar lines might be adduced *ad infinitum;* for example, the foot brake. Because in the old car designs it was found desirable to apply one of the brakes to the secondary gearshaft, we still continue to transmit braking stresses through unnecessary transmission. Readers with road experience will probably agree that it is desirable to transmit the ordinary braking stresses through the differential, for no extraneous compensating mechanism can balance the braking result in the same manner as does a brake acting through the differential, for while the compensated brake can ensure a balanced force on each wheel, that force is applied regardless of any difference in speed at which the wheels may be running. Granted that the foot brake should be applied through the differential, it may still be contended reasonably that it is superfluous to transmit braking stresses through all the transmission intervening between the gear box and the differential. No doubt the problem of braking will be very much modified in the immediate future by the increasing employment of front wheel braking sytems, and it is also possible that the growing popularity of the wire spoked wheel may also lead to the introduction of an accessible type of rim brake, on lines somewhat similar to that used in cycle design. In making this suggestion the writer does not claim to broach any new idea, for rim brakes on automobiles have been in use since the early days of the motor movement, and would have continued in use had the wheels of those days been suitable for employment with this type of brake.

Here again we have an example of another factor that tends towards conservatism in design, for the one part of a car is so bound up with others that it is extremely difficult to realise at once the influence that modification of one detail may have on other parts, or on the design as a whole; consequently, sometimes considerable periods may elapse before all opportunities brought about by any given change are fully taken. In making this suggestion, however, the fact that there may be other factors at work must not be overlooked. For instance, it might be argued that, as the tangent spokes of wire wheels for motor construction were always bent sharply over at the head, and the spoke led through a hole in a flange on the hub, our manufacturers had never been able to get away from the conventions originally set up by the cycle manufacturers; there is now no patent to stand in the way of employing a straight spoke; it has been used on some bicycles for many years, but has been little adopted for self-propelled vehicles. But there may often be reasons deeper than the surface. In this case, while the use of direct spokes might conduce to greater strength, the cost of manufacture would certainly be increased, seeing that the bent-over spoke is on the market, and can be produced at practically the same cost as the straight variety. Probably the cheapest way of carrying out the straight spoke idea is to turn the hub with flanges, and mill these latter out so that what is left of the flanges constitute the lugs that take the spokes; but even so, to carry out the theory in its

entirety these lugs must be milled and drilled at the correct angle, and in any case the milling out and drilling must inevitably add to the cost. Still, the idea has already been embodied in heavy motor vehicles.

The writer does not claim that any single one of these points would necessarily by itself be the means of obtaining any particular desired quality in full degree, but all are points that if taken with others may assist in reaching the ideal.

Doubtless every reader can furnish other instances showing points that call for the attention of the designer. The few cases that we have discussed are only quoted to help prove the same contention, and although commercial and patent considerations may on many occasions stand in the way, we cannot but think that those readers who give the matter thought, will admit that improvements might yet be effected in many points of design.

LACK OF ALL-ROUND KNOWLEDGE.

By the Editor of "The Autocar."

After fifteen years' close acquaintance with the motor industry nothing has surprised us more than the extraordinary lack of knowledge which the heads of one motor manufacturing firm have of the practice of another. It is true this is an age of specialisation, and that to succeed a man must be a specialist, but it often seems to us that in the world of motor engineering at least this specialisation has been narrowed unduly because there are so many talented designers and engineers who, instead of being specialists upon the subject of motor engineering as a whole, are specialists upon the one make of car with which they are intimately concerned.

In the early days of the industry, say, up to about 1902, every leading designer and motor manufacturer, besides many of the subordinates, both in the engineering and sales departments, had a very good general all-round knowledge of the leading makes of the day. Some of the more advanced not only knew what their own cars would do, but they knew just where they were better or worse than the cars of half a dozen other makers. This all-round knowledge was unquestionably one of the factors which did so much to make the evolution of the motor car so rapid. At the same time it was even then more limited than it should have been, but nowadays it is in many cases almost nonexistent. This is serious, because the motor car, though far from finality will, in all probability, develop towards the unattainable goal of perfection far more slowly in the future than it has in the past, and therefore improvements in the individual parts of the car are more and more necessary to enable a firm of makers to keep pace with the times.

As things stand now the average attitude is one of disbelief or sheer ignorance. For instance, if some particular car with a certain size of engine makes an exceptionally good performance, the verdict of nine-tenths of the members of the industry is expressed in a single word, "fake." Few of them take any trouble to find out why this particular engine has done so well. Too often the suggestion is made that some unfair practice has been resorted to, and all sorts of ridiculous tales without any basis whatever are circulated about it. One asserts its compression is abnormal; another that it knocks itself to pieces in no time, and has to be practically rebuilt after every performance. A third suggests that the fuel was not everyday petrol, and the thing is passed from one to another in this spirit; but there is no attempt to find out the truth. This applies again to any startling novelty, and the tales which are circulated in motor engineering circles concerning one of the latest are simply extraordinary. It might be thought that these were merely the attempts of one rival to belittle the goods of his opponent, but these figuments of the imagination are repeated among practical engineers to their fellows, and are not what one may call salesmen's tales. However, this is a very unpleasant side of the subject, and we do not want to dwell upon it. We would only say that an ounce of personal investigation is worth a ton of fabrication.

To turn from unbelief to ignorance we find very extraordinary results at times. For instance, one maker asserts quite honestly that his steering is perfect, but if one drives the car one may find it distinctly bad. It may be a vast improvement upon the previous steering of the particular design and manufacture, but it may be very much behind what it should be. The fact of the matter is, the maker has never driven any other car but his own, and he does not really know what good steering should be. The same thing applies to the smooth running of the engine. The so-called quiet and smooth engine may be quieter and smoother than the particular make has ever been before, but compared with a really refined engine it is a coarse machine. As to the alleged silent gears, it often happens that the most silent gear of the man who boasts would not be passed by the rawest tester of a firm which makes really quiet gears.

We might go all through the car and give a dozen other instances about ignorance as to the proper performance of vital parts. This ignorance is simply due to the fact that neither the maker nor his designers have had any experience of other good cars. They surpass their own previous efforts, and think all is well. It is not for us to suggest now they may enlarge their experiences, but this is fairly obvious to anyone who gives the subject a few moments' consideration. At first sight it may be imagined that, after all, if a firm can pay its way and make a profit there cannot be very much wrong with its methods, but we can very quickly show that this view is not correct. There is no doubt that a good many motorists do not know the difference between a thoroughly good car and one which is merely reliable. On the other hand it must be remembered that a vast number do know the difference, and are highly critical, and it is these people who have such knowledge and enthusiasm who have a tremendous influence upon the fortunes of a firm. If a car be turned out in any particular year with a noticeable vice, the amateur owner who has bought and driven anything between half a dozen and twenty cars in his motoring experiences very quickly tells his friends, and they spread the tidings that the car is bad. He knows what a car should be, and they know that he knows. In fact, the owner-driver often has a wider experience than the average maker when it comes to a good all-round practical knowledge of a number of different makes.

This is one of the bad results of ignorance of other makers' practice, and there is no doubt that much harm is done to certain makes in this way. But it does not end there. This lack of acquaintance with anything but his own car often leads a maker to all sorts of experiments, which have been tried and found wanting long ago by one or other of his competitors. Not only so, but by his ignorance of what they are doing and how their cars handle he often makes ludicrous mistakes. As an instance of this we may pursue our simile of the bad steering, and may mention a couple of examples. One firm had no ball thrusts to its steering pivots and the other none to its steering box. As is usual in such cases the steering was not what it might be. Both cars were two of the best makes of the day, and really good cars in almost every respect, and yet we could not find that either their designers or anyone connected with the firm had ever brought up the question of the lack of ball thrusts in the two different but vital places, nor did they know from practical driving experience that their introduction was such a real improvement.

Then again, the all-important quality of "holding the road" varies immensely on different makes of approximately equal specification. Those responsible for a car bad in this respect not only say that only big, heavy cars will hold the road, but show genuine surprise when told that certain cars of other makes, though of about the same dimensions and weight as their own, give no trouble on the same roads and at the same speeds which make their car tricky or even positively dangerous. They actually do not know that a car may weigh less than 26 cwt. and still keep nicely to a moderately good road at all speeds up to double the legal limit. Their's does not, and therefore they think the thing is impossible.

We might go into the matter at almost indefinite length, as the instances could be multiplied by the dozen, but we may say we have not given one which is a year old, and we think it will be admitted we have made good our plea for a specialisation of somewhat broader character than that which is current in the automobile engineering world of to-day, for after all it is of little use for designers to discuss papers and theories, if two-thirds of them have never handled a really good car in perfect trim and do not know how it should behave on the road.

UNIVERSAL JOINTS.

An examination of the theory of their action, and a comparison of various designs in use.

THERE is little doubt that, on an average, the least satisfactory part of any modern transmission system is the universal joint, or joints, employed therein. Serious trouble with any part of a modern car is unusual, but the rapid development of lost motion between the clutch and the road wheels is still too common an occurrence to call for comment. Any such back-lash is unpleasant for passengers in the car, on account of the snatching which must take place whenever the engine is exerting but a small driving or retarding force, and also, if there is slack between the gear-shaft and the bevel pinion shaft, the application of a propeller-shaft brake causes considerable shock, unless its engagement be made with a most unusual and inconvenient gentleness.

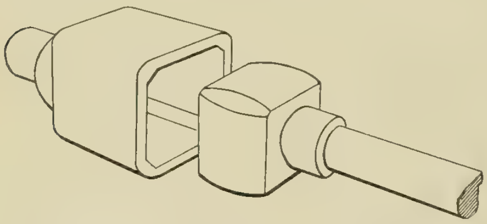


Fig. I.—The pot type used on a good many of the older cars, and still in use on some of the more modern vehicles.

Excluding the small amount of back-lash necessary between the actual teeth of the gears, the only points where it is able to develop are at the coupling between the clutchshaft and the gearshaft, or at the universal joint or joints, as the case may be, in the propeller-shaft. Lately some universal type of coupling has come into common use for connecting the clutch to the gearshaft, in order to facilitate the setting of an engine and gear in the frame, or perhaps to prevent trouble arising should the frame give a little in use and so bring crank-shaft and gearshaft somewhat out of alignment.

In the majority of cases these joints are perfectly satisfactory, because they should not be called upon to work out of line, except in the smallest degree, and where slack is found to exist it can usually be traced to faulty design or bad workmanship in the first instance. The joints used for propeller-shafts are, however, very different, because they are called upon to bear considerably greater stresses, and are very seldom running in line, even though the shaft may be so set that it is free from deflection in the normal position.

In the early days of shaft-driven cars, the universal joints employed were frequently of the most primitive description, due to the fact that, when used in other engineering work, universal joints are very seldom called upon to stand the same angularity, and to designers' failure to appreciate this. The excessive wear of many joints is largely due to insufficient lubrication, and designers have certainly realised this now, although, with a few notable exceptions, they took a long time to do so. However, they are still liable to overlook the fact that unless universal joints are fitted with some easily detachable form of cover, or with a lubricator, the task of looking after them properly is an extremely awkward and dirty one when the chassis has been used on the road, and that, therefore, it is a job likely to be neglected by most drivers. On the other hand, if the design includes some device rendering the replenishment of grease in the joint cases a quick and easy operation, it is certain to increase the average durability of the joints. However, the question of lubrication will be considered presently in detail, and, while concerned with the design of the joints themselves, we may assume that lubrication is properly attended to.

There has lately been evident a tendency to give up the old arrangement of a propeller-shaft having a universal joint at each end in favour of a long shaft carrying the bevel pinion at its lower end, and running in a tubular casing extending from the back axle to the gear box, terminating in a single universal joint. This construction possesses the advantage of neatness, and has also one or two other good points, but it has also some bad ones, chief amongst which may be placed the fact that it causes variations in the angular velocity of the bevel pinion.

Assuming that the gearshaft is driven at a steady speed, the propeller-shaft must, of course, run at the same speed, measured in complete revolutions per unit time; but while the angular velocity of the driving-shaft is constant (if the driven one be connected by a universal joint, and be running at an angle with the driving one), its angular velocity is in a state of constant variation. The angular velocities of the two shafts are equal four times every revolution, but between these four positions the velocity of the driven shaft reaches two maximum

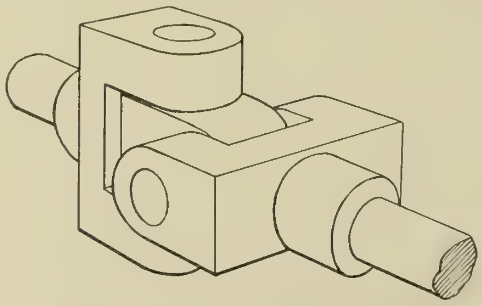


Fig. II.—The cross pin pattern in which the pins are not in the same plane.

and two minimum speeds, differing from the average speed by equal amounts and by a degree depending upon the amount of angularity between the two shafts.

This means that if we take a line to represent the angular velocity of the driving-shaft, use fractions of a revolution as horizontal co-ordinates, and angular velocities for vertical ones, then, if we proceed to plot the velocity curve of the driven shaft, we shall find that it crosses the velocity line of the driving-shaft four times in each revolution.

For calculation of the variation Rankine gives a formula :—

$$V_1 = V \cos i, \text{ and } V_2 = \frac{V}{\cos i,}$$

where V = the angular velocity of the steady-running shaft, V_1 the minimum velocity of the driven shaft, V_2 its maximum velocity, and i = the angle of inclination between the driven shaft and the horizontal.

Assuming that V = 1,000 revolutions per minute, and that i = 5°, we find that the driven shaft is turning at speeds varying from 996 revolutions per minute to 1004 every half revolution. If the inclination is as much as 10°, which it may be easily when the car is running fast on a rough road, then the minimum speed of the shaft becomes 984 revolutions per minute with a corresponding maximum.

The variations in speed of the driven shaft may perhaps be regarded as too

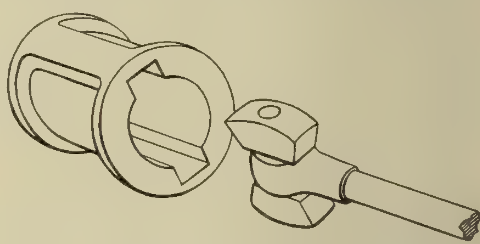


Fig. III.—The De Dion type of universal joint.

small to be worthy of notice, but it is very doubtful whether they ought thus to be neglected. Variation in the angular velocity of the bevel pinion must produce equivalent variations in the pressure between its teeth and those of the crown wheel. In fact, when a car is running over a moderately even surface, the teeth pressures will fluctuate quite considerably. This is bad for two reasons: Firstly, because it tends towards a reduction in efficiency; and, secondly, because it is liable to cause noise, or to have an aggravating influence on any noise-producing vibrations which may be the effect of quite a different cause. It is probable that some manufacturers who have taken especial pains to render their final drive as nearly dead silent as possible could throw some light on the real value of steady tooth pressure. In no case can it be a factor of great importance, but it might very likely just make the difference between that perfection of quiet running which is the aim of the designer of every pleasure car and what might be described as merely ordinary quiet running.

However, by using the older type of propeller-shaft with a joint at each end, this variation of velocity is entirely removed, provided that the gearshaft and the bevel pinion shaft always lie in parallel planes, and that the parts of the joint bearings at each end of the propeller-shaft are also both in a single plane (not crossed, for this construction simply doubles every variation). This arrangement removes the trouble, because equal angles will be made between the gear-shaft and propeller-shaft and the bevel-shaft and propeller-shaft, and while one joint is (so to speak) bent downwards, the other is bent upwards, and *vice versâ*, any variation imparted to the shaft being neutralised in passing through the second joint.

We have known cases where designers were acquainted with the irregular running of a shaft driven through a single joint, and have therefore employed two, but have then proceeded to nullify the effect of the second joint by the use of

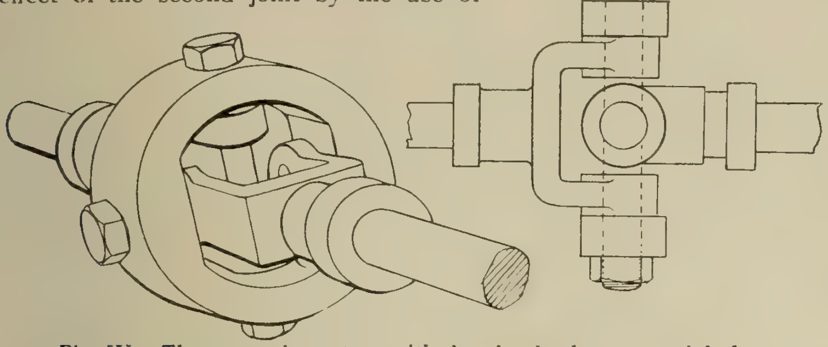


Fig. IV.—The cross pin pattern with the pins in the same axial plane.

a long torque resisting rod constraining the bevel pinion shaft, always to lie in a line which, if produced in the forward direction, would strike the centre of the forward universal joint, or, at all events, come to a point very near it, and, of course, the axle might just as well have a single universal joint at the forward end only, as two used in combination with a torque rod arranged in this fashion.

There is also another objection to the single joint pattern of drive that is common to every axle controlled by a torque rod which constrains it to move in an arc of a circle, and that is that every movement of the back axle up or down causes the crown wheel to move relatively to the pinion independently of the driving movement, and so gives the road wheels a varying velocity. With the double-jointed shaft this does not occur if the motion of the pinion-shaft relatively to the frame is parallel, or nearly so.

Some makers set the bevel-shaft in a practically horizontal line, and fix the whole axle to the road springs, allowing it no rotation in the spring pads. With this arrangement the motion of the shaft is much more nearly a parallel one, but it is an open question whether the clamping of an axle to the springs is not bad practice, for reasons altogether apart from the matter we have at present under consideration. One of the quietest cars ever built was the one brought by Mr. C. Y. Knight from America, and this had an axle of which the motion was controlled by a torque rod of immense length. It consisted of a single large tube, and terminated at a point but little behind the engine. This would give a very nearly parallel motion with the bevel pinion shaft, and was doubtless of some assistance in the reduction of running noise.

On the other hand, there is another aspect of the case, and that is that, with a good difference between the elevation of the gearshaft and the centre of the back axle, if the two shafts are parallel the angles made between them and the propeller-shaft will be slightly sharper than the angle which would be made by a single shaft leading from the gear box right down to the centre of the back axle. This means a slightly higher efficiency for the single joint arrangement on this account.

The whole matter is one of compromise. When one shaft is driving another which is in a state of constant movement, there must be some loss of power, and, taking the middle course, it appears that the best all-round results could be obtained by using a double-jointed shaft, and controlling the motion of the back axle in such a manner that the motion is neither parallel with the gear-shaft or radially to the forward joint, but in a manner which will give it positions intermediate between these two.

The designer of a universal joint has three common types from which to choose, and there is a fourth which was used with a fair amount of success some years ago, although it could hardly be called popular at the present time. It is shown in fig. I., and, if well made, may work and wear extremely well. The blocks and pots require to be machined with the utmost care, and should really be scraped to fit, as a very little slack will cause rather rapid wear. Both parts require to be hardened, and, generally speaking, while it is a good joint to use between the clutch and the gearshaft because it can be separated very easily, it is not to be recommended for propeller-shaft work.

Fig. II. is the type of joint which a short while ago was practically supreme, and its faults are the cause of the clanking noises which become noticeable in most cars fitted with it after a few months' use. The fact of the axes of the two pins being in different planes aggravates the variations in the velocity of the driven shaft. The joint, as a whole, is by no means easy to encase, and if the bearing surfaces are made sufficiently large it becomes very clumsy. It is worthy of note that this type of joint is referred to in a well-known engineering text book, published so long ago as 1901, as being "suitable for rough machinery where the angularity of the two shafts is not very great."

Fig. III. shows the type of joint made famous by its success as used on the cardan axles employed for so many years by the makers of De Dion cars. It requires to be well fitted in the first instance, and is very useful where a certain amount of telescopic motion is necessary. When slightly worn, disconnecting the joint and giving the square nuts a quarter turn on their pins before re-assembling removes, at any rate, a good percentage of the slack, but when wear takes place on the sides of the external container, practically nothing can be done short of replacing the whole arrangement.

The friction in this type of joint is higher than that of type IV., and unless surfaces are large the bearing pressures are very considerable. It also suffers from the disadvantage that it tends to pump out any lubricant contained in it, and this tendency can only be combated successfully by the fitting of a lubricator which will supply grease at the inner end of the pot. It is quite possible for a joint of this kind to be rusty at the inner end, although it receives frequent small charges of grease from the front end.

Fig. IV. promises to become, if indeed it is not at present, the most common form, and it has advantages over all the other types illustrated. There is no difficulty in case-hardening the bearing surfaces of the pins borne by the shafts, nor is there trouble in fitting them with good bushes, clamped in the floating ring in such a manner as to render their renewal quite simple and easy. Further, the joint lends itself to enclosure, and we shall refer later to the arrangement adopted on Wolseley-Siddeley cars, which is, perhaps, the neatest of all. (See fig. V.)

Telescopic motion on one of the joints is, of course, necessary, unless the motion of the back axle is controlled by an expensive parallel link arrangement, or by a troque and radius member pivoted on the exact centre line of the front universal joint. We have already pointed out that there are two good reasons why an axle should not be supported in this manner, and so the consideration of telescopic joints is important.

In practice there are two arrangements to be found: one, and far the most common one, is the De Dion joint we have just described (fig. III.), while the other is to allow one of the forks of the front or rear joint to slide on its shaft either on a feather or on a square, hexagon, or castellated shaft. Although the use of a simple feather may sound absurd, it is worth recording that a number of small Beaufort cars were given the necessary shaft freedom by allowing the foot brake drum, which carried half the universal joint, to move longitudinally on the gear-shaft, a mere quarter by three-eighths feather taking the whole drive. This affair, nevertheless, worked quite well, and was reasonably durable.

A square or hexagon end on the shaft sliding in one of the joint forks is, of course, much stronger than a feather, but it is none too easy to make a really close fit, while it weakens the shaft considerably. The castellated shaft makes much the best job of any, and should be almost everlasting if the sleeve is sufficiently long, but at the same time all sliding devices of this kind are not really good practice, because they cannot be renewed without the replacement of large and costly parts, and, all things considered, the De Dion joint is probably as good a compromise as has so far been designed.

We have said already that the average durability of universal joints leaves much to be desired, and that faulty lubrication is a potent cause of wear. Size, however, has a great deal to do with durability, and the great variations in the ideas of different designers makes an interesting study.

From an examination of the joints used by ten well-known makers, it was found that the pressure between the pins and bushes of the joints varied from 950 lbs. per square inch to 3,220 lbs. per square inch, whilst the average pressure for the ten examples was 1,791 lbs. per square inch. It is obvious that the lower the pressure is the longer the life will be, and with a load of 3,000 lbs. to the square inch it is not surprising that a bearing quickly wears away. Especially is this so when it is remembered that these pressures are calculated from the

driving torque, and that when gearshaft brakes are used the pressures may easily be at least three and even four times as great with reversal of stress.

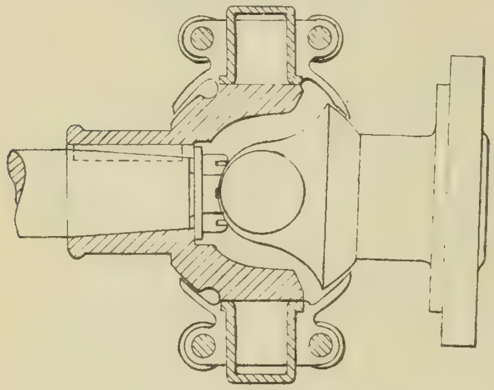


Fig. V.

For modern materials a bearing pressure of 1,800 lbs. may not sound very high, particularly for a bearing in which the motion is only intermittent; but having due regard to the fact that joints do undoubtedly wear too quickly, it would be wise were designers to reduce the pressure to 1,500 lbs. per square inch, and this is possible without using a clumsy joint. The pressure depends on the area of bearing surface, the mean diameter of the joint, and the torque of the engine. Thus—

$$\text{Pressure} = \frac{\text{H.P.} \times 33{,}000 \times 12}{D \times \pi \times A \times R}$$

where D = effective diameter of joint in inches; A = area of bearing surface of fork carried by any one shaft, in square inches, and R the minimum number of revolutions of the shaft at which the rated h.p. can be transmitted.

This formula lends itself to simplification if

$$\frac{33{,}000 \times 12}{\pi}$$

be taken as equal to 126,000, whence we get

$$\frac{126{,}000}{R} \times \frac{\text{H.P.}}{D\,A}$$

= the bearing pressure in lbs. per square inch.

It is better to obtain strength by making the joint of fair diameter than by making each bearing very large, because the larger the diameter of the lubricant-containing case, the further from the shaft is any oil or grease thrown by centrifugal action, and the only place of escape for it should be, and usually is, along the shaft. On the other hand, as the forks (and the ring, if type IV. be used) are subject to bending stresses, it is easy to overdo diameter.

From an examination of the same ten joints from which the average bearing pressure was calculated, it appears that a diameter of threequarters of an inch is a favourite size for the pin with a length of from one inch to one inch and a quarter, the total bearing surface per fork being from one and a half to two square inches. These ten cars are all of about 20 b.h.p., so a good rule would be to make each fork pin $\frac{3}{4}$in. to $1\frac{1}{4}$in. in diameter, according to the power of the car; to let the length of bearing (diametral thickness of the ring) be from 1in. to $1\frac{1}{2}$in., and then to obtain the diameter of the whole joint from the equation

$$D = \frac{\text{H.P.}}{R \times A} \times \frac{126{,}000}{P}$$

The effective diameter (D) of a joint is, of course, equivalent to the full span of a fork less the length of one pin's bearing surface (with fork bearings an inch long if D=4in., the total diameter of the joint = 5in.)

So far we have refrained from the consideration of any means whereby universal joints might be improved, except by an increase in their size; but in designing a joint, the fact that sooner or later it will require new bearing surfaces should not be forgotten. It is just as easy, and just as cheap, to make a joint in such a way that the renewal of bushes is simple. With the right type of joint such renewal is usually not difficult, but designers who use the De Dion type are very prone to omit everything which will permit its dismantlement short of actually shifting the back axle or gear box. As examples of good joints in which the bushes are easy to renew, we show the front joint used on Wolseley cars (fig. V.) and on Sunbeam cars (fig. VI.). These show two different ways of dividing the ring, and while there is but little if any difference in cost, the Wolseley is certainly stronger and less likely to work slack on the road, while the Sunbeam is neater, and the fitting of new bushes would be a somewhat less troublesome task.

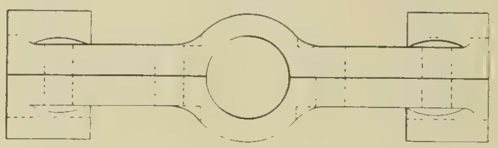


Fig. VI.

The Deasy designers have disposed of universal joint troubles by the expedient of fitting ball bearings to the pins, a small self-contained race being substituted for the usual bushes. Roller bearings would be still better, but we believe the Deasy joint has proved itself by experiment to be a great deal more than ordinarily durable. Of course, it is a costly design, and therefore is only to be considered for moderately high-priced cars where quality is of first importance.

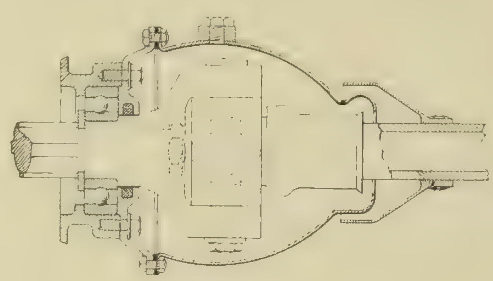


Fig. VII.

Another rather interesting joint is that used on the cars made by Brazil and Straker (fig. VII.); this has a very large "cross-bolt" intersected by a smaller one, and all the parts are of a large size. The bearing pressure works out fairly low, and the joint is cheap to make. At the same time it is not so easy or so cheap to enclose as the ring type, and when it wears it would not be so easy to set up again as a ring joint or a De Dion pattern.

The lubrication of universal joints is of greater importance than was believed by most designers up to quite a short time ago. At present it is receiving a great deal of attention, but in many cases it has been found that the design of an effective-looking oil-retaining cover was easy enough, but that when in use it did not quite come up to expectations as a dust excluder.

It may be laid down as a positive fact that any joint cover which depends upon leather for its security from the egress of lubricant or the ingress of mud, is foredoomed to failure, unless the leather be in cup form, or simply used as a washer between two metal surfaces, because leather is reduced by oil to a condition of extreme ductility, and is therefore easy to drag out of shape.

The Straker Squire cars made by Brazil and Straker have stationary joint covers attached to the back axle and gear box respectively. These are of a roughly spherical shape, and, of course, have fairly large orifices to allow for movement of the shaft. To close these the

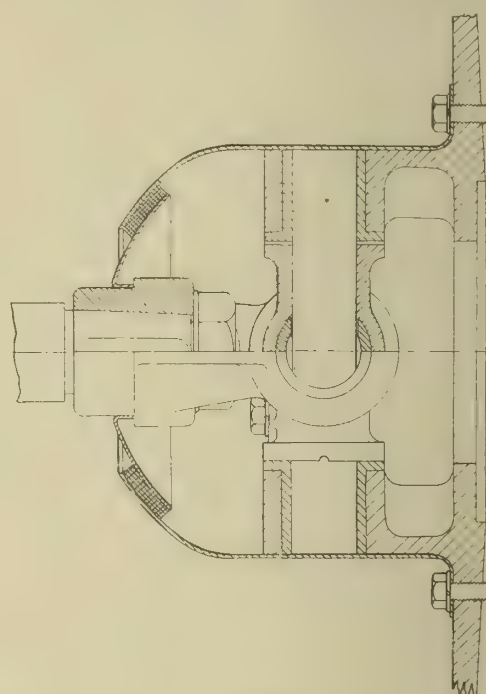


Fig. VIII.

shaft carries two conical sheet metal cups, which overlap the fixed covers. This leaves a slight annular opening, but it is well protected, and should allow but little foreign matter to enter, while lubricant is retained by the fixed covers quite effectively. (See fig. VII.).

A fixed cover is also used on Adams cars at the forward end, and it is formed solid with the gear box. The opening is closed by a leather cone, terminating in a ring, which fits the shaft, but allows the latter to rotate within it. There are some points of advantage in having the cover fixed, and it is claimed that this arrangement gives better lubrication, because the oil is not kept lying against the periphery all the time a joint is turning, as is the case, more or less, when the whole arrangement revolves as one piece.

The commonest forms of protector are two in number, and are both pressed from sheet metal, usually brass. In each case the main portion of the cover is

roughly hemispherical, and is bolted to a disc behind the joint. The point of difference between the two types lies in the connection made with the shaft. In one case the metal cover is left with a sufficient opening, and the final covering is performed by strapping a leather cone to both cover and shaft. In the other case a small pressing, similar in shape to the main casing slides on the shaft, and is kept in contact with the main casing by spring pressure: being a part of the surface of a sphere, it can, of course, move freely in any direction as the shaft moves. It is sometimes fixed inside the main case, and sometimes outside, and is usually separated from it by a leather ring, which serves to make the joint still less liable to leakage. As an example we show in fig. VIII. a sketch of the universal joint cover used on Argyll cars, and also on Humbers.

The effective protection of the De Dion type of joint is more difficult, and this is due partly to its shape, and partly to the presence of telescopic movement. It is still common practice to leave the end

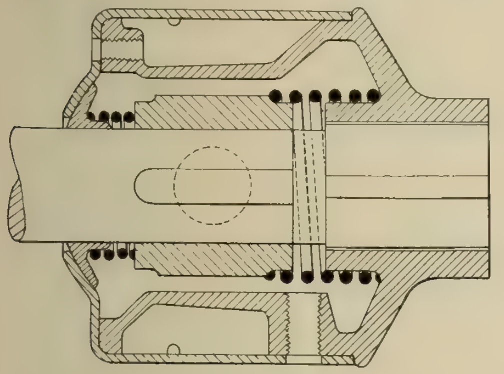


Fig. IX.

open, and simply to strap a piece of leather between it and the shaft. This can only be secure during the life of the leather. If the latter is really good quality and well dressed with dubbing in the first instance, it will last a great deal longer than it would if it were a poor quality of an unusually absorbent nature, but in no case can its life be considered satisfactory, owing to the deteriorative action of lubricant. A very neat method of enclosing this type of joint is used on Swift cars, and is shown in fig. IX. It should, by appearance, be an ideal arrangement.

There are also an increasing number of what might be termed self-contained joints, that is to say, joints in which the cover is formed as an integral part. Such a one is the front end joint used by the Wolseley Company, and shown in fig. V. This joint is lubricated by an oil-way, terminating in one of the bearing caps closed by a screwed plug, the idea being that oil can be supplied here with an ordinary can. The disadvantage is that this simple process may be rendered extremely awkward by an inconsiderate coachbuilder. Another good self-contained joint is used on Sheffield-Simplex, fig. X., and Crossley, fig. XI., cars. In these the shaft is enclosed in a tube which terminates in a spherical expansion. This ball end is retained in a correspondingly shaped casing, firmly attached to the frame. In both the cases mentioned, the tube containing the shaft acts also as a torque and radius member, but there is no obvious reason why the same device should not be employed on a double-jointed shaft, and it certainly gives the best protection to the working parts. Having the outside part of the case stationary, it is easy to arrange for it to be supplied with either oil or grease, as pipes can be carried to any convenient part of the chassis where they are not likely to be interfered with by body-work.

One objection which can be brought against the ball and socket cover is that of possible difficulty in compensating for wear, but as the two surfaces do not move over each other to any considerable extent, and as they should always be in a bath of lubricant, wear ought to be extremely slight, so slight, in fact, as to be of no account. The method of securing the ball used by Crossley Bros. is perhaps the better of the two, as it would be easier to take up in case of any slackness developing.

It is impossible to proceed further with the discussion of universal joint practice without entering upon a consideration of arrangements for the resistance of torque from the back axle, but it would perhaps be well to first consider ideal arrangements from the point of view of flexible coupling only. This would appear to be as follows:—

(1) Shaft jointed at each end.
(2) Shaft of bevel pinion controlled so that it is given a nearly, but not quite, parallel motion.
(3) Joints of the ring type.
(4) Joints and shaft enclosed by a light tube with enlarged spherical ends secured in spherical cases attached so as to remain stationary outside shaft and joints.

It now remains to be seen to what extent these ideals interfere with one another. Firstly, as there is no telescopic motion, the distance between the centres of the two joints must be the same for all relative position of gear box and back axle shaft which means that if the bevel shaft remains horizontal, it is at the greatest distance behind the gear shaft when in the normal position, and

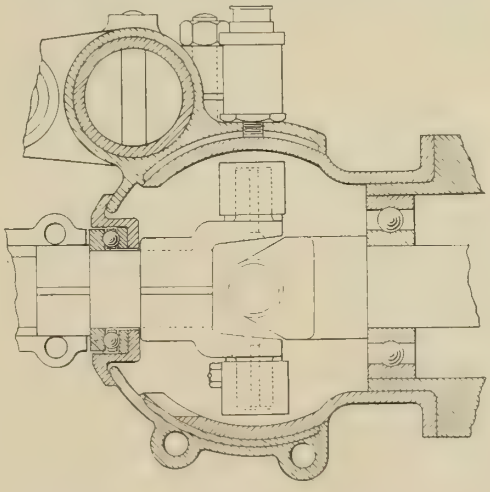


Fig. X.

approaches it if the latter either rises or falls. It is obvious that this means free shackles on the front ends of the rear springs, and motion definitely controlled. In order to obtain the desired motion of the back axle, some arrangement is required akin to a parallel motion, and there is no reason why the tube which encloses the cardan shaft should not form one of the links. To get the other link a point may be chosen above or below the centre of the forward end joint, and from it a radius rod can be joined up to the top or bottom of the differential case.

A somewhat similar mechanism is in use by the Maudslay Company, but in this instance the propeller shaft is not enclosed, and the radius rods are arranged in pairs and duplicated, one set being slung from each side member of the frame. It would in any case be necessary to duplicate one of the links of a parallel motion, in order to compensate for the loose front shackles on the springs.

No extra member would be required to resist torque, as the parallel motion links are well adapted for this purpose, and it is worth noting that on Maudslay cars the radius rods also serve as shock absorbers, their points of attachment being mounted on supplementary springs.

The only objection to this construction

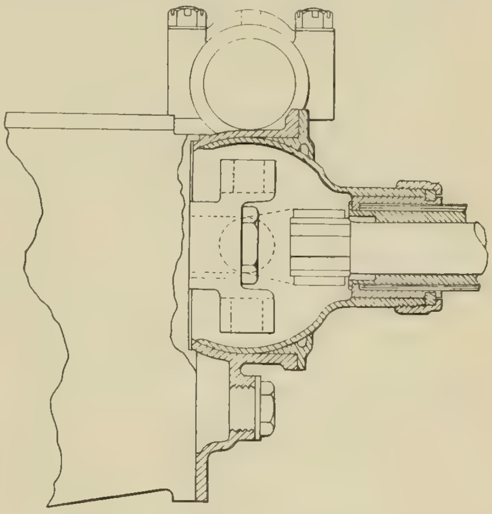


Fig. XI.

is its cost, and it is no doubt this important factor of design that has led to the production of so many cars with theoretically wrong systems of transmission, which often betray their inherent weaknesses by early and rapid wear.

Still, if an ideal is costly, and something very nearly as good is a great deal cheaper, then the latter may be much the better commercially. It might be pointed out easily that the Daimler cars have, for the past three seasons, been made with a simple pair of joints (one telescopic), and have had the axles bolted to the springs, these taking all driving and braking strains, as well as performing their ordinary functions. However, the whole question of suspension would have to be gone into before it would be possible to say definitely whether this is good practice or not, and such a discussion does not come within the scope of this article.

If the nearest approach to the system of shaft drive suggested as ideal is required to be obtained at a cost less than that of the parallel motion arrangement, probably fixing the axle to the springs, or using a very long torque member with the axle free in the spring pads, are about equally good as far as the drive is alone concerned. Either method necessitates the use of a telescopic joint, and for the encasing of this and of the other joint the designs illustrated are all about equal in cost, with the exception of the ball and socket, or Crossley type, which is decidedly expensive to make thoroughly well, and unless so made it could not be expected to prove satisfactory.

GRINDING AND GRINDING MACHINERY.

The growth of a new class of tool and its great effect upon economical repetition work. Descriptions of some of the machines most used for automobile work, and instances of the great saving in cost of production obtained by grinding as opposed to older methods.

THE use of grinding machinery in the production of a large proportion of the parts of modern motor cars has increased within comparatively recent years to an extent that could hardly have been foreseen. The tool room, as a separate and very special department, has, of course, been a feature of all first-class engineering works for many years, and for the work done in the tool room, grinding machines of great precision were an absolute necessity. But it was only a natural sequence that the use of hardened steel parts and very fine workmanship, which are to-day essential features in all cars of repute, should bring the grinding machine into the general machine shop, where, not only has it established itself permanently, but its scope also has widened astonishingly.

It is used not only for hardened surfaces, but for general details which have hitherto been produced on the lathe; indeed, although the possibilities in this direction have, perhaps, not yet been sufficiently realized, it looks as if, for many kinds of repetition work, it will entirely supersede the lathe: not only so, but in its latest developments it is ousting planers and milling machines, producing work of superior finish and with a rapidity hitherto unknown.

The whole subject, taken broadly, reads almost like a romance, and shows, incidentally, how a branch of industry can grow and flourish in the sunshine of demand, when assisted by the fertilizing influences of modern scientific methods.

As has been the case in many other departments of engineering, we have to thank America for leading the way in the very rapid development that has taken place, in which connection the well-known names of Brown and Sharp, Pratt and Whitney, Norton (represented here by Chas. Churchill, Ltd.), and Landis are prominent, but British makers are now coming well to the fore, and when we find such firms as Alfred Herbert, Ltd., Luke and Spencer, Ltd., H. W. Ward, Jas. J. Guest, George Richards and Co., Ltd., Wm. Muir and Co., and many others, devoting considerable attention to the business, it will be seen that would-be purchasers have an ample field for selection.

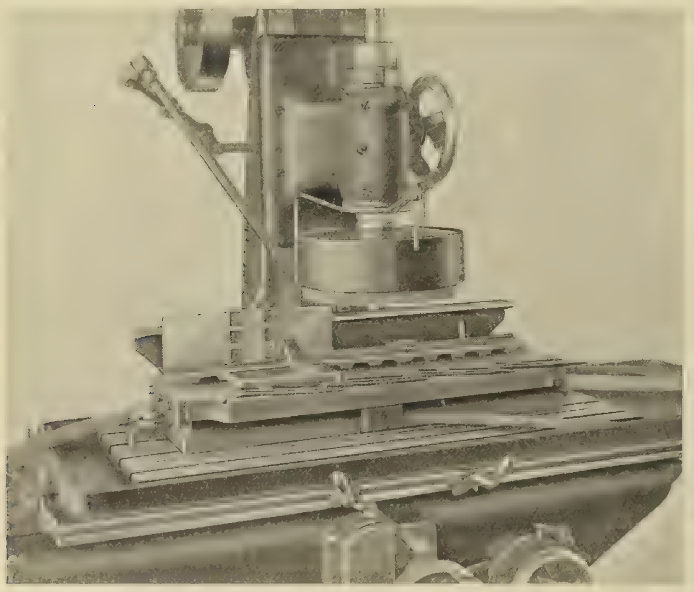
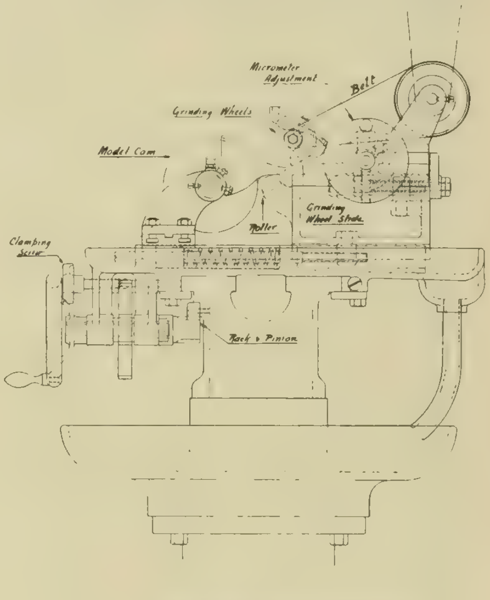


Fig. II.

To the engineer who is concerned in the economical production of interchangeable parts, and, therefore, especially to the automobile manufacturer, the advantages conferred by the grinding machine are so obvious as hardly to need mention; it may be said, in fact, that without its aid such work would be commercially impossible.

In accuracy it easily hold first place, as it will work—in the hands of a skilful operator—to a limit of one ten-thousandth of an inch. In rapidity of production, too, it competes on very close terms with the lathe, and, for flat surfaces, with planing machines, or vertical millers, in many cases beating both easily.



END VIEW OF CARRIAGE AND BED.
Fig. 3I.

As regards lathe work it is now being found both practicable and profitable to take drop forgings direct to the grinding machine without putting them in the lathe at all, or making any previous preparation beyond cutting to length and centreing.

Here is an instance in the production of crank shafts. A certain maker uses a 2in. 4-throw shaft, with a 1 in 10 taper at the tail end, made from a drop forging. In this case the shaft, after being cut to length and centred, is taken to the grinder, which in two hours finishes the pins and journals, removing ¼in. of stock, and finishing to a limit of .0025in. This is done by a Landis crank grinder, a machine that is specially designed for this class of work.

When it is remembered that only a few years ago the time taken to finish such a shaft in the lathe would be from 40 to 60 hours, it will be realized what a marvellous advance is marked by such an operation as that just described. Of course, the time mentioned does not include finishing the crank webs, but it seems well worthy of consideration whether it would not be just as well to leave these unfinished. As drop forgings they can be made sufficiently clean, and—apart from the question of balance, which is easily overcome—their remaining unpolished does not in any way detract from the satisfactory performance of the engine. It may, perhaps, not be out of place to remark here that it is in the careful consideration of details like this, and the elimination of all unnecessary operations, that the British automobile engineer may find it possible to reduce his costs, and so be able to meet threatened competition from abroad, and the constant cry for cheaper cars at home.

To give a few more examples of the rapidity with which work is produced, it is possible to finish pistons 4½in. dia. × 5½in. long at the rate of 15 per hour, $\frac{1}{32}$in. of material being removed In smaller work, gudgeon or piston pins, 3⅞in. × ⅞in., after case-hardening can be finished at the rate of 70 per hour, whilst valve stems can be done at the rate of 75 per hour.

Shafts and such like, which require to be case-hardened, have a tendency to distort in the process, and without the grinding machine this would be a serious matter taking much time to rectify, but as grinding is the last operation, this difficulty is no longer of any great moment, and for the same reason there is no spoiling of work by cutting keyways, which, of course, are done previous to grinding.

Crankshaft and other journal bearings, which used to require a tedious and expensive process of hand scraping, can now be so accurately finished by grinding as to entirely dispense with the hand work, and, moreover, experience tends to show that ground bearings wear better, owing no doubt to the fact that no high

Fig. III.

places can be left, so that a full, even bearing is secured.

So far, only round work has been referred to, such as would be dealt with ordinarily in a lathe, but the grinding machine

has been designed to deal with flat surfaces also, and in this direction its scope would seem to be even greater. In the first adaptations for face grinding the periphery of the wheel was used, the work being traversed backward and forward on a table under it, whilst the wheel itself was given a transverse motion to and fro across the work. For long, narrow work this arrangement proved excellent, but obviously it has its limitations. The latest development is a new vertical machine, in which a cup-shaped wheel is used face on to the work, and the variety of work that can be handled by this machine in short time is extraordinary.

It will do practically anything that can be done on a vertical milling machine, but by the aid of magnetic chucks it will deal also with very thin articles, such as knife blades, piston rings, pistol handles and levers, which a milling machine could not deal with at all.

There is scarcely an item in the modern automobile in the manufacture of which the grinding machine cannot be usefully employed. It would make too long a list to go through the whole of them, remembering, as we may, that anything from one to five thousand parts are required to make a car, but a few may be mentioned, viz., cylinders, pistons, piston rings, valves, crank shafts, cam shafts, cams, crank chambers, gear boxes, cardan and back axle shafts, gear boxes and covers,

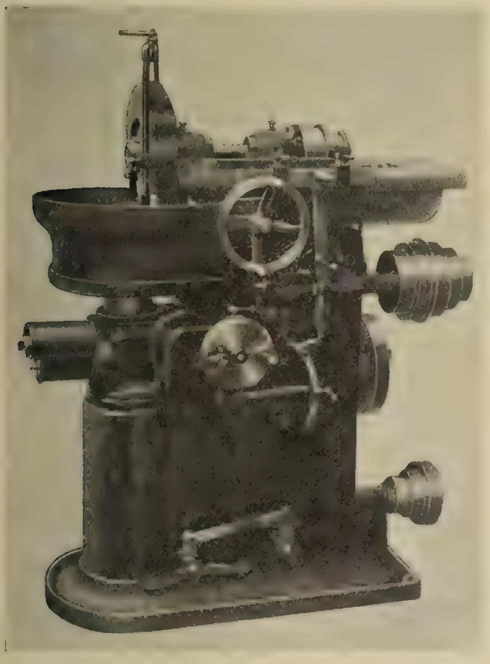


Fig. IV.

axle casings, bearings, bushes, pins, studs, and so on. It requires no great stretch of imagination to conceive that with a little care in the drawing office a car might be designed, in the making of which lathe work could be practically eliminated. It is true we cannot tap or screw with the grinding machine, but for the ordinary operations of turning and boring it is possible, and probably will be found profitable, by the use of drop forgings and otherwise suitably selecting our materials and reducing the margin of stock to be removed, to dispense with the lathe to a far greater extent than has yet been thought possible.

Who shall say to what extent our machine shops may not be revolutionized in this direction in the near future? Special machines will be called into existence for particular operations, and in time, when automobile engineers have given up their pursuit of new "talking points" for each new season's cars, (which necessitates the constant changes in design and scrapping of thousands upon thousands of pounds' worth of

Fig. V.

highly expensive drawings and patterns), we may at last find ourselves within measurable reach of the much-talked-of £100 car!

The romance of the grinding machine is found in the abrasive wheel, which is its essential feature, and without which it would possibly never have emerged from the comparative obscurity of the tool room.

The history of the discovery, development, and present-day manufacture of alundum and carborundum provides a study of engrossing interest, and shows, incidentally, to what a pitch of practical utility the electric furnace has attained, for without the great heat of the arc the reduction of these two valuable substances from their native clay would hardly be possible. And without them the grinding machine could not have advanced so rapidly as it has done. Emery, the only powerful abrasive we had in the early days, is not to be compared with alundum or carborundum in cutting, temper, or lasting power, and although still used to a very large extent for certain purposes, it has been, to a great extent, superseded for precision grinding.

In order to appreciate what the character of an abrasive wheel should be, it is necessary to clearly understand what takes place in the operation of grinding. The grinding machine is really a milling machine of a modified type, and the grains of abrasive material are the cutters. These little cutters remove only microscopic cuttings, but they do it at such a speed, and there are so many of them, that their output of work is comparatively large.

Of course, like steel cutters, they wear and become blunt, but, unlike steel cutters, they cannot be sharpened, so it is necessary to keep breaking them away and so exposing new and sharp grains in order to maintain the cutting power of the wheel.

In order to get the best results, the size of the grains must be carefully chosen to suit the particular material that has to be operated upon. The wheels, therefore, are not formed from solid slabs of abrasive material, but the latter is reduced to granular form, and then sifted out into various grades through wire mesh of definite dimensions, varying from 6 to 200 per inch. The very finest grades are got by throwing the finest powder into water and selecting in accordance with the time taken in settling.

Wheels can thus be manufactured of any degree of coarseness or fineness as desired, each of a perfectly uniform quality and texture throughout, the abrasive being mixed up with special cements, which, after being moulded to shape and dried, are subjected to a very intense fusing temperature, that makes them very strong. Wheels are made either hard or soft, and in many degrees of coarseness and fineness, and it is in the selection of the most suitable combination for the material being operated upon, that success in grinding depends. A wheel which is good for soft cast iron is not likely to succeed with hardened steel. A wheel that is too soft will break away and wear out too rapidly, whilst one that is too hard for its job will tend to glaze. But just here another factor appears, in that a wheel may glaze from being driven too fast or held up too hard to the work. So to secure the best results one must study up all the various points involved, or engage a first-class operator, who has done the studying and had experience as well. But, fortunately, the great manufacturing firms who are making the wheels have, by constant experimenting, accumulated great store of information and reduced the selection of wheels to finality. It is, therefore, only necessary to give full particulars of material to be dealt with, together with a few necessary details of the machine, in order to get the right wheel for the work.

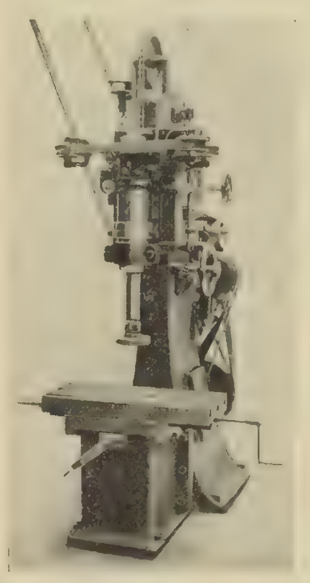


Fig. VI.

A volume might be written on this subject, *i.e.* in regard to proper selection of wheels, speeds at which they should be run, methods of mounting, protecting devices to prevent injury to the operators from bursting wheels—fortunately a rare occurrence—and so on, but space does not permit of an extended treatment of these details. Only as regards speed it may be noted that from 4,000 to 6,000 feet per minute is the usual speed for Norton grinders.

Coming now to the grinding machine itself, it will be found that quite a large number of different types are manufactured, but broadly they can be divided into classes, viz.:—

(1) Plain grinders.
(2) Universal grinders.
(3) Surface or face grinders.

(4) Tool grinders, and

(5) Machines for special purposes.

In the illustrations accompanying this article will be found examples by leading makers of all these types, which may be briefly described as follows:—Plain machines for cylindrical and taper outside work are those which are used for the larger details that can be carried on dead centres, such as shafts, spindles, pistons, etc. The universal machine is one which can be used for both internal and external

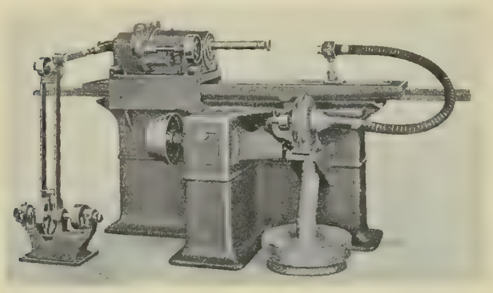


Fig. VII.

grinding, and can conveniently deal with smaller details, such as bushes, rings, pump barrels, piston pins, valves, and such like. Surface grinders, as their name implies, are suited to deal with flat surfaces, such as crank cases, gear boxes, inspection doors, etc. Tool grinders are used in the making and keeping in repair of nearly every class of tools. Of special machines there are not yet a great number on the market, but we think it probable that as time goes on the number will largely increase, owing to the success that has attended those already introduced. In order to ascertain to what extent special machines are being used in motor manufacturing the author communicated an enquiry on the subject to a number of the leading British car makers. The majority replied to the effect that only standard machines were being employed, but the Belsize Co. were kind enough to send a very interesting description of a special cam grinding machine, an end view of which is shown by fig I.

This machine, designed by the staff of the Belsize Co., and made by Messrs. Pollock and McNab, of Bredbury, near Manchester, was introduced in order to provide a rapid and accurate method of correcting the distortion of cam shafts that almost invariably results from the hardening process. The machine has a fast and loose headstock for carrying the cam shaft between centres, and a shaft along the front carrying the former cam. The head for carrying the grinding wheels is placed on a slide carrying a roller which is held hard against the former cam by a strong spiral spring in the carriage. In order that the whole space between headstocks may be available for traverse of the carriage, the grinding spindle carries a wheel at each end. The model is made with two shapes, one for inlet cams, and the other for exhaust. In operation, the cam shaft is placed in its centres and the carriage runs along the bed until one of the grinding wheels is in position for grinding the first cam on the shaft. The spindle is rotated by hand until the highest point of the former is bearing against the roller in the grinding spindle slide. The cam shaft is now rotated until the highest point of the first cam is against the grinding wheel, when it is clamped, and the machine is then started.

It has already been said that manufacturers are well catered for by tool makers, and some details of the most useful machines for automobile work will thus be both useful and interesting. Fig. II. shows a vertical surface grinder, by Pratt and Whitney, which is a quite new design. It has a cup-shaped wheel, 12in. diameter, which covers the full width of the work, ensuring perfect flatness and very rapid output. It is claimed that it will do its work from 12 to 20 times faster than any other machine yet made. The table, which permits of a working surface of $10\frac{3}{4} \times 36$in., is provided with straps for holding the work, but it has also a magnetic chuck for thin articles. There is, in addition, a rotary table, used for grinding rings, discs, collars, etc., and the motion of this can be engaged or disengaged without stopping the emery wheel. The vertical spindle runs at a fixed speed of 1,100 r.p.m., and the vertical slide which carries it has ten automatic feeds, obtained by a ratchet wheel, each tooth of which represents a movement of .0002in., so that feeds from .0002in. to .002in. are available. Provided with a special attachment this machine can be used for grinding cams, either solid with the shaft, or detachable, and produces work of extreme accuracy. Some idea of its capacity may be gathered from the samples of work shown in the illustrations. Single cams of usual size can be ground from the rough at the rate of 30 per hour, and fig III. shows a back-axle casing, with alternative covers of pressed steel, and cast malleable, which can be faced from the rough in less than half the time they would take on a milling machine, and finished to closer limits. The machine is of very massive construc-

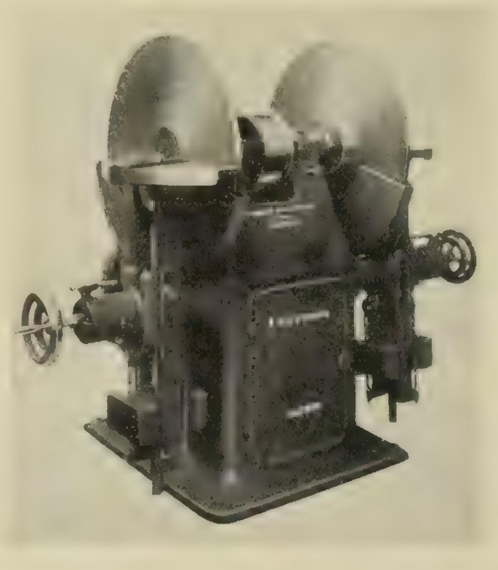


Fig. VIII.

tion, and is adapted to be driven either by belt or motor.

Fig. IV. shows a "Churchill" ring and surface grinder. This is another surface grinder, but of an entirely different type, the wheel being carried on a horizontal instead of a vertical spindle, and it has been designed more particularly for dealing with such items as piston rings, discs, dies and annular surfaces either flat, convex or concave. For this latter work it has an adjustment of 3 degrees on either side of zero. It has a magnetic chuck, specially designed for wet grinding, which requires direct current of 100 or 200 volts, equivalent to what would be used by an ordinary lamp, a simple switch sufficing to grip or release the work. There is a very fine range of feed, with micro adjustment and automatic stop, the rate of feed varying at will from .0001 to .00075 inch. Weighing upwards of a ton, it will be realized that the machine is not likely to suffer

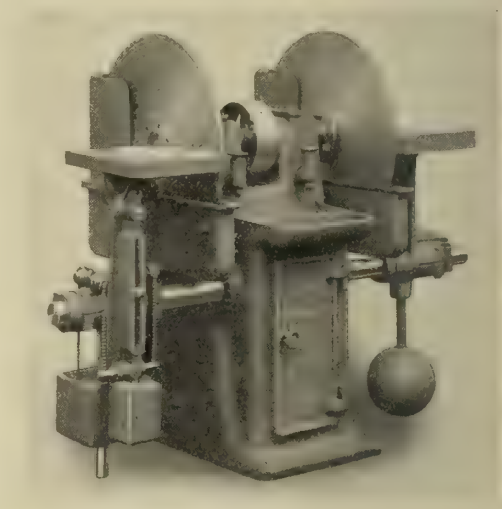


Fig. IX.

from vibration if placed on a suitable foundation.

Ward's surface grinder, fig. V., by H. W. Ward, and Co., Ltd., illustrates a third type, that is designed to handle work up to 3ft. long by 8in. wide. The general appearance is neat and compact, and its construction is well up to this maker's usual high standard of excellence. The table has automatic traverse in both directions, controlled by suitable stops, which are arranged in such a manner that they can be run past the reversing lever in front of the machine, without disturbing the setting. The cross feed, which operates at each reversal of the table, is instantly reversible, and adjustable to feed from .25in. downwards. The vertical feed to the wheel is by screw, worm and wheel, permitting graduations of .001 inch, and a useful feature is an automatic belt tightening device, compensating for adjustment of the wheel slide, and maintaining always uniform tension on the driving belt. A very neatly-designed diamond wheel-trueing device is also a feature of this machine, as can be seen in our illustration.

No automobile factory that makes its own engines can afford to be without a cylinder grinder. True it is that this machine is not found in every such factory, and that there are some people who go so far as to say it is an unnecessary luxury, which can quite well be done without, but once we admit the necessity, or even the desirability, of grinding in other departments, we are bound to recognise that absolute accuracy in our cylinder bores as an essential second to none, and uniform accuracy can only be obtained in a commercial way by grinding. We illustrate two useful types, the first being of the vertical spindle, and the second, the horizontal spindle type. Fig. VI. shows the former, turned out by Messrs. C. W. Burton Griffiths and Co., and it will be noticed that the design is a

particularly convenient one in regard to setting up the work and watching its progress. The wheel spindle is carried in cylindrical revolving sleeves, and is driven by a belt passing over flexibly-mounted guide pulleys. The sleeves are adjustable in relation to each other while running, in such a way as to cause the wheel spindle to be moved from a position of concentricity, and to travel in a circular path, the radius of which is variable up to its extreme limit by the simple movement of a hand wheel, while the driving mechanism of the outside sleeve is designed to give quite smooth action, free from jar or backlash. The table, 30in. × 12in., has a travel lengthwise of 18in., and a cross traverse of 4in. The machine will deal with cylinders up to 6in. diameter by 12in. deep, and is provided with adjustable vertical automatic traverse to any required stroke within its limits. The wheel spindle bearings are of ample size, and are adjustable for wear by means of taper housings.

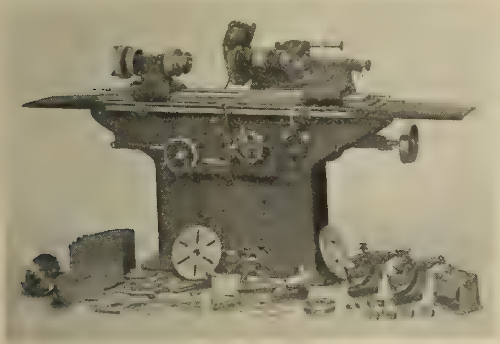


Fig. X.

Fig. VII. shows the second type, the Brown and Sharpe horizontal cylinder grinder. This machine would be useful for cylinders cast singly, and for a considerable variety of other internal grinding, in which it is difficult to revolve the work, but for cylinders cast in fours it would hardly be so convenient as the machine described above. The wheel spindle is of hardened and ground tool steel, running in adjustable bronze bushes, the front end being made taper to receive the wheel sleeves, so as to permit of quick change of wheels without the need for trueing after mounting. The spindle drive is by means of a belt from an intermediate shaft fixed on the floor behind the machine, and passing over loose pulleys mounted on a swing arm provided with adjusting screw for controlling the belt tension. The table travel is automatically controlled by adjustable dogs, which have the property of re-adjusting themselves, to prevent the table over-running the reversing points. An exhaust fan removes dust from the work, and deposits it in a tank partly filled with water.

Coming properly under the description, "surface grinders," are disc grinders, which have earned for themselves a most excellent reputation for all-round usefulness. From comparatively humble and unassuming beginnings, these machines have developed into true precision grinders, and are almost indispensible in any kind of engineering works.

Fig. VIII. shows a good example of this type by Messrs. Burton Griffiths. The discs are of mild steel, mounted on a hardened and ground shaft running in two very large self-oiling bearings, between which is the belt pulley. The body is of extremely massive construction, to ensure freedom from vibration. Tables are provided for resting the work upon, and these tables, besides having a pendulum movement across the wheel, can be set to any desired angle to the face. Quite an industry has sprung up in the manufacture of various kinds of abrasive discs for these machines, and it is now possible to get some which will work on aluminium without clogging. If care is taken to prevent abuse of this machine by apprentice lads, and even fully qualified fitters (who too often damage it by using it for sharpening chisels, or grinding off the ends of thin rods or bolts), it will earn its price in a remarkably short space of time.

Fig. IX. shows a somewhat similar machine, by Messrs. Alfred Herbert, Ltd., which has protecting guards and a fan for removing dust. In its main features it is like the one above described. The speed is 1,000 r.p.m., and horse-power required is about eight.

The machines already described serve fairly to illustrate the best modern types, for dealing with flat surfaces and cylinder bores. We may now turn our attention to those used for external cylindrical work, and internal work on which the parts themselves, as well as the wheel, are rotated during the operation of grinding. One of the most useful of this class is the "Universal" grinder, of which the illustration, fig. X., shows a 12in. × 36in. machine, by Messrs. Alfred Herbert, Ltd.—one of their latest designs, which they are supplying largely for motor work. This machine has a swing of 12 inches, and admits between centres a length of 36 inches. It will take wheels up to 12in. diameter by 1in. wide, and it is provided with three wheel speeds, viz., 1,751, 2,190 and 3,000 r.p.m., whilst there are twelve speeds for the work ranging from 20.6 to 372. For internal grinding it has also three speeds, viz., 11,400, 14,230 and 19,500. The wheel is carried on the end of the spindle to ensure ease of removal. The spindle itself is hardened, ground and lapped, and carried on adjustable phosphor bronze bushes, lubricated by continuous feeding syphon lubricators, provided with filtering pads. The wheel slide, supported by a heavy base, is graduated, and will swivel to any angle, and telescope covers are used to protect the slides from dust and water. The automatic cross feed has a range from .00025 to .004in., with automatic stops and adjustment to compensate for wear of wheel. A swivelling table provides for tapers up to 4 degs. each side of centre line, and up to 2in. per foot, while there is a fine adjustment for accurate taper setting, and the table can be clamped securely without using a spanner. The traversing table has self-oiling slides, carefully protected from dust. The automatic table traverse is driven by a single constant feed pulley, independently of the work speed. A variable feed device enables the rate of traverse to be instantly regulated by a handle at the front, without any danger of grinding a flat on the work when changing the feed. Considerable attention has been paid to the steady blocks, which are of simple form,

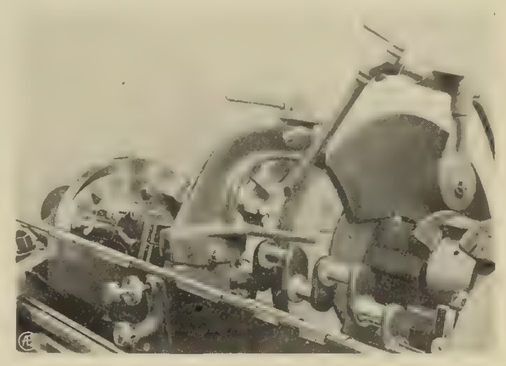


Fig. XII.

and may be made of wood, white metal, brass, or any other material, without requiring special castings. The horizontal steady block is adjustable by a horizontal, and the vertical block by a vertical screw, this adjustment being rational and easily understood. The lower block acts so as to hold the work back against the horizontal one, so preventing thin work crowding on the wheel. Adjustment of the blocks is independent, so that setting one does not upset the other.

For internal grinding various sizes of attachments fit into the same bracket interchangeably. The grinding spindle is driven by loose connection, and, being entirely free from belt pull, will run at high speeds without heat. It is impossible to go more fully into detail in an article like this, but enough has been said to indicate the care that has been expended on the design of this interesting machine, which fully upholds Messrs. Herbert's high reputation.

Fig. XI. shows a Landis crank grinder, which can be used for finishing single cranks up to 8ins., and multiple cranks up to 6ins. throw. It is made with specially heavy rigid heads, the main spindles are extra strong, and the face plates are also made very heavy and capable of resisting all the strains set up by this work.

The grinding wheel is mounted on a large diameter steel spindle, hardened and ground, running in bronze bearings,

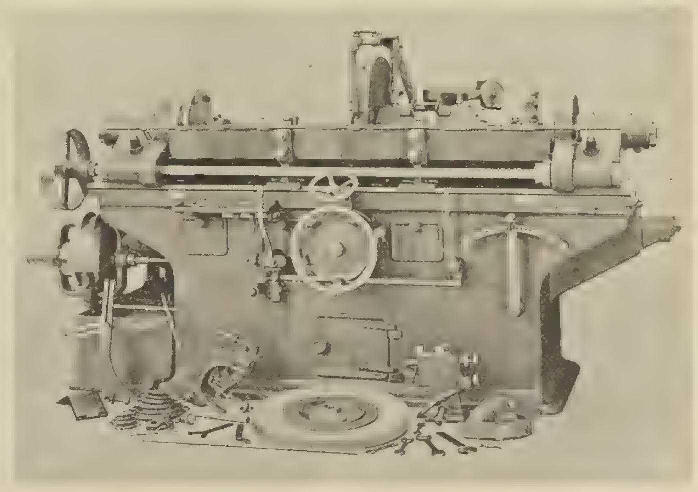


Fig. XI.

adjustable for wear. The wheel travel is automatic in either direction, controlled by easily-set dogs, operating against a sliding bar in reversing lever. By raising this bar the wheel can be moved beyond the reversing position on either hand, and returned again without resetting. The wheel traverse speed change device is obtained by simply shifting a lever, requiring no changing of gearing, clutches or belts. The automatic cross-feed of the wheel operates at each reversal of the carriage, and can be adjusted to reduce the diameter of the work at each reverse from .00025in. to .002in. It is regulated by pressing a thumb latch. This feed will finish work accurately to within a quarter of a thousandth part of an inch, and stops automatically when the work is finished to the size for which the mechanism is set. Every improvement and refinement that experience could suggest has been incorporated into the design of this fine machine, and applied by the Landis Company; that is saying a good deal. In the illustration, fig. XII., a view of this machine is obtained, which shows it at work on a stamped crankshaft, as referred to in the early part of this article.

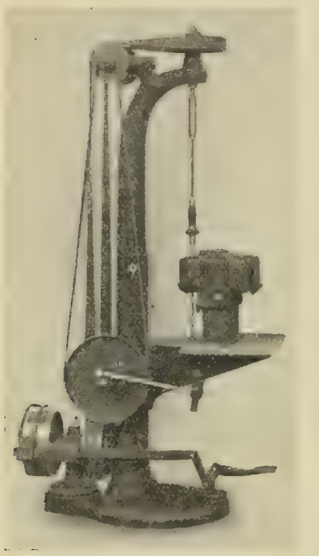


Fig. XIII.

A very useful type of plain grinder is manufactured by the Churchill Machine Tool Co. This has a swing of 4ins., takes 24ins. between centres, and wheels up to 10in. diameter and 1in. face. All wearing surfaces of the wheel slide are of large size, and well protected from water and grit. It has automatic cross-feed, with a maximum movement of 3in., working to within .00025in., and a swivel table for grinding tapers up 2in. per foot. The table travel has ten speed changes, with instantaneous change from roughing to finishing traverse. This is an excellent type of machine for small work, such as gear box shafts, pistons, etc.

An interesting example of a machine designed for one special purpose is seen in Messrs. Herbert's valve grinder, fig. XIII. At first sight it might seem to be superfluous to have a machine for such a purpose, for the grinding in of valves is usually looked upon as a quite trifling matter, but if we consider the case of a factory turning out, say, 20 cars a week, which means 80 cylinders, and 160 valves, the money saving effected by this machine is worth consideration. It is stated that at the Daimler Works, where the machine was first introduced, the work of the machine was found to be better than hand work, and that it took less than *one-third* of the time. The construction is clearly shown by the illustration, and consists of a vertical column, with self-contained countershaft, driving a vertical spindle, and a flat table for carrying the work, fitted on a dovetailed slide on the side of the column. In operating, the cylinder is placed upon, but not fixed to, the table, the valve is put in position, an automatic lifter is adjusted to suit length of valve stem, and the machine is started by depressing the foot lever. A screwdriver, which forms the bottom end of the flexible shaft, is pressed down by means of the handle into the usual notch on top of the valve, causing the latter to rotate. The direction of rotation is always the same, but by the automatic lifter the valve is periodically raised from its seat, thus ensuring perfectly uniform grinding. Being entirely self-contained, and occupying a floor space of only $3\frac{1}{2}$ft. by 2ft., the machine is easy to instal, and it requires very little power to drive it.

If there is one item about an engine that ought particularly to be ground, it is the cams. The special machine of the Belsize Co. has already been described, and Messrs. White and Poppe have a very successful little machine, made to their own design, which deals with single cams in a highly satisfactory manner.

The Pratt and Whitney vertical surface grinder, already described, has an attachment for grinding solid camshafts, and three of these, with eight solid cams on each, can be ground from the rough in one hour. An important point to which attention may be called is that with this arrangement the cams are always ground by a truly flat surface, which does not change, owing to wear of wheel, whereas in peripheral wheel grinding the steadily decreasing diameter of wheel, due to wear, does affect to some extent the contour of the cam, that is to say, with a given model the produced cam will be perceptibly different in contour when ground by, say, an 8in. wheel, than it would be were the wheel reduced to 4in.

For bushes of all sorts, whether of hardened steel or bronze or brass, and for rings, collars, and such like, it is always well to have a special machine on the job. These items in a large, busy motor factory, soon run into thousands, so there is no difficulty in keeping one or more special machines fully occupied. A machine made by James J. Guest, Birmingham, who also makes a variety of larger grinders, will be found very useful for this class of work, for it is substantial, rapid in operation, and simple to handle. The wheel spindle is of hardened and ground steel, running in dust-proof bearings, of phosphor bronze. Taper work is produced by swivelling the workhead or table. A small countershaft, carried on the cross slide, is used to avoid excessively high speed of the belt running to the machine. The feed wheel is graduated in thousandths of an inch on the diameter of the work, and suitable provision is made to ensure accuracy in sizing. The work is held by a self-centreing chuck, and the machine is provided with a steady. The maximum distance from chuck face to wheel spindle bearing is 20in, and swing is up to $5\frac{1}{4}$in. The countershaft pulleys run at 550 r.p.m., and the spindle speed is arranged to give a peripheral wheel speed of 5,000 feet per minute.

In concluding this article it may be well to point out that the author has endeavoured to confine his attention to a selection of machines of a character suitable to automobile manufacture, and the descriptions of the machines, necessarily very brief, are only such as will give a general idea of each. The selection has been made without any regard to the names and standing of the makers, but merely with the idea of giving a good and representative selection of machines by a good and representative selection of makers.

SOME POINTS IN THE DESIGN OF AERO-MOTORS.

By W. G. Aston.

WITH the rapid advancement in the development of the aeroplane, the question of the most suitable motor for the purpose becomes increasingly important, and is a matter that deserves the closest attention of motor car designers, for it is they who already have the facilities for the manufacture of light internal combustion engines, and it is from them that must come a supply for which every day finds a greater demand. As a problem of design and construction, the aeroplane engine is one of the hardest nuts to crack that has yet come the way of the motor designer, but a satisfactory solution to the difficulty is certainly rendered much easier of achievement by the aid of experience gained in the making of automobile engines.

A satisfactory engine for aero work has to satisfy the combined requirements of the car and the boat engine; thus, it must be light for its power, and be capable of working continuously at full load. The fact that in a motor car the engine rarely has to give its full output for more than a few minutes at a time, and that in between such periods of heavy loading there are long "breathing spaces" of light load, accounts for the success of many engines, which under less easy circumstances must inevitably have failed. Constructors of motor boats early realised this important point, and the wiser designers of aeronautic engines have already doubtless done likewise. There have been many instances of aeroplanes engined with motors taken out of cars and slightly lightened, but in most cases success has not attended this practice, and certainly in all the cases that have come under my notice it has not been possible to obtain flights of any duration.

Just as cycle engineers developed into motor manufacturers, so must the motor manufacturer inevitably take the leading part in the coming flying machine industry; hence it is clear that the aero motor

can hardly escape being a direct development of the car engine, but it cannot be too strongly emphasized that the divergence between the two is, and will continue to be, a very marked one.

The primary requirements of an aerial engine are : (i) Reliability, (ii) Economy, (iii) Lightness, and (iv) Absence of Vibration. I have placed these in what I consider to be their order of importance. Combined, these four desiderata necessitate a superlative degree of excellence in design, construction and material, which must tax the skill of the best-equipped manufacturer.

The necessity of extreme reliability and lightness in an engine whose function it is to sustain an aeroplane against gravity is too obvious to require enlargement, but economy is not, at first sight, so important a consideration as its position in the list would make it appear. I have used the word, however, in its more comprehensive sense, to embrace the consumption of fuel, lubrication and water (if water cooling be employed). Although, as regards petrol alone, the aeromotors of the present day are certainly extremely wasteful, it would be unfair to say that they are unduly so; still an extremely severe indictment may be justly brought against them in the matter of oil consumption. Many of the air-cooled aero-engines which have achieved success, notably the Gnome and the Anzani, use very nearly as much lubricant as petrol, so that they might almost be said to be "oil-cooled." This is, of course, a fault that cannot be exaggerated, as unless it can be cured or ameliorated to some degree, it must seriously curtail the distance that an aeroplane can traverse between consecutive stopping places. It is clear, in this connection, that the rough and ready method of supplying lubrication to the working parts, which has served its purpose in the car motor, must soon give place to a more rational and less wasteful system in the aero-engine, as much on the score of tending towards reliability and capacity for full output at long stretches as upon that of low consumption. If for no other reason the splash lubrication to the pistons must surely give way to something more highly specialised, on account of the desirability of auxiliary exhaust ports in the cylinder walls, which are uncovered by the piston at the bottom of its stroke. Even if the piston be of such length that these ports, even at the top of the stroke, are not open to the crank chamber, a very large volume of oil is wasted, and, if the engine be in front of the aviator, such oil is thrown forcibly back on to him, and is liable to cause him great distress, if not actually expose him to imminent danger. Several accidents have been ascribed to this cause, and, as seems likely, with some truth, for several times I have seen aviators descend after quite a short flight with their faces smothered with dirty oil, and hardly able to see out of their eyes. Even when the auxiliary wall exhaust is not used, occasional over-lubrication, which only the most carefully-considered can avoid, may result in pungent smoke being blown into the aviator's face, unless, as is unusual, exhaust pipes of some length are used.

Great lightness of motor, though admittedly desirable enough, is, beyond a certain limit, not so absolutely necessary as it has been made to appear. One hears commonly of claims made that certain aero-motors weigh only 3 lbs or less per horse-power, but as a rule, when one enquires into the matter, one finds these figures have been obtained by neglecting the weights of the ignition, carburetter, flywheel, and radiator. It would be absurd to gainsay the fact that on the same aeroplane a light engine can travel further without stoppage than a heavier one, because it can carry more fuel, but lightness, *per se*, may pass rapidly from a virtue into a serious fault, as beyond a reasonable limit it must involve a sacrifice of strength and reliability. Unimpeachable material is, therefore, of the highest necessity, for only by that means can weight be cut down and strength kept up. At the same time it must be remembered that lightness is only taken as a measure of power, and that merit in this respect is represented by the expression

$$\frac{\text{Power.}}{\text{Weight.}}$$

Assuming the existence of an engine of certain weight, it may as easily, and much more advantageously, be improved by having its power output increased, as by having its weight decreased. It seems to me that enough attention has not been paid as yet, at any rate in aero-engine design, to the acquisition of "more power" by means of various devices. It is true the problem is a difficult one, but it is surely worth while as being the more rational means of enhancing the value of the fraction $\frac{P}{W}$, since the denominator cannot be taken below a certain minimum. I believe that a certain French firm have succeeded in building a motor of 30 H.P. (continuous load output) weighing only 80 lbs. complete, but how unusually light this is may be seen from the following figures, which relate to the pounds weight per horse-power of the more successful aero-motors of to-day :—50 H.P. Darracq, 5 lbs.; 40-50 H.P. E.N.V., 4 lbs.; 60 H.P. Wolseley, 6 lbs.; 30-35 H.P. Green, $5\frac{1}{4}$ lbs.; 50 H.P. Gnome, $4\frac{1}{2}$ lbs.; 30 H.P. Anzani, 5 lbs. The above weights do not include ignition outfit, nor flywheels, except in the case of the Anzani and the Green.

Following upon the question of weight for power, and closely related to it, is the question of r.p.m. Unfortunately for the designer of aero-engines, high speed is not what is wanted, so that he is unable to obtain the power required by the simple expedient of increasing the r.p.m. when the engine is to be direct-coupled to the screw, for whereas the thrust of a propeller varies as the square of the speed the power absorbed increases as the cube. Therefore to obtain screw-efficiency it is necessary to use propellers as large in diameter and as slow-running as possible, so that a reduction gear is rendered necessary if the engine revolutions are to be kept high. In this connection it would be interesting to ascertain exactly what power is given out by many so-called 40 H.P. engines when direct-coupled to the propeller. The speed of the latter very rarely exceeds 1,000 r.p.m., whereas I have a suspicion that the B.H.P. is taken at a considerably higher speed. An engine capable of giving high power at a reasonable speed is therefore the ideal for aviation purposes, as any speed reduction gear must form a factor in the weight of the power plant, besides being wasteful of power, both by friction and possibly by increased head resistance

As regards freedom from vibration, this is a point far more important than is apt to be thought. The extreme lightness of the aeroplane fuseleage or framework to which the engine must be bedded down makes it particularly liable to develop defects when subjected to constant vibration, and this is not the only point. Unless the motor be fairly well balanced, it will cause all the truss-wires and stays which are present in large numbers in nearly all aeroplanes to vibrate excessively, and in doing so they exercise a head resistance out of all proportion to that which they offer when not vibrated. In a biplane, for instance, there may easily be 700 feet of guy wire, and this, when in a state of vibration, will offer a resistance equal to that of a flat plate seven (or more) times its diameter, so that the total increased resistance may be equal to that of plate some 3 feet square, or larger. I would not suggest that such wire vibration is entirely due to the engine, but it is very largely so.

Whether air or water-cooling be the better method is a question which only time can solve, but up to the present it would seem that honours are fairly evenly divided. On the whole, the water-cooled engine has been more uniformly successful, especially in long distance work, excepting, of course, flights performed with the Gnome rotary motor. The last type has been called by many the "engine of the future," but it must not be forgotten that although it has been so successful, it possesses certain inherent disadvantages that reduce the lead it would otherwise have over the orthordox type of motor. On the other hand it has been subject to far less development than the stationary cylinder type, and a few years may find it improved clear of the defects it at present suffers under. As to its balance, and freedom from vibration, there is no question, but as to its fuel efficiency there may be some doubts, for not only has it to drive itself round through the air—a matter which absorbs a percentage of power that cannot be neglected—but it must also be borne in mind that when the aeroplane fitted with this type of motor travels forward through the air, the engine offers a resistance practically equal to that of a disc some three feet in diameter.

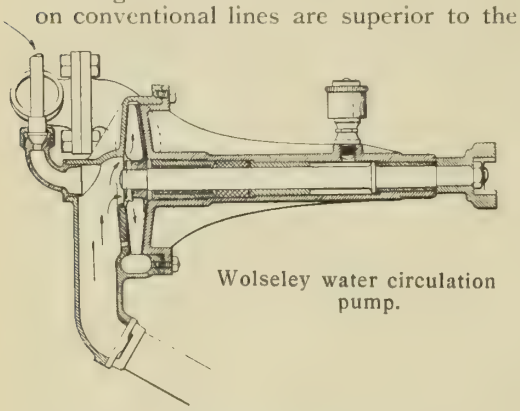
With regard to the question of water cooling, this, it would seem, is a matter that is dependent upon circumstances to a great extent. Thus, if an engine is to be placed immediately in the wake of the screw. it will be subjected to a blast of air having a velocity at least twice as great as that of the aeroplane, so that the problem of air cooling successfully is facilitated very greatly. On the other hand, if an engine be tucked away, in order to reduce head-resistance, water cooling seems inevitable. As to whether it is more economical to add to the weight of the power plant by introducing water cooling than to place the engine in such a position that it cools itself and imposes additional head-resistance, is a question which the designer of the future must solve. My own view is that air cooling is about to "come into its own," and not only for aero work.

THE 16-20 H.P. WOLSELEY.

A critical description of one of the most carefully designed and constructed of British-built cars.

THE 16-20 h.p. Wolseley is in many respects typical of the rest of the Wolseley models. In common with all its kind, it has been designed with a view to its use as a carriage, and no attempt has been made to obtain very high power from the engine, smooth running and true flexibility being regarded as far more important.

For general convenience few cars made on conventional lines are superior to the

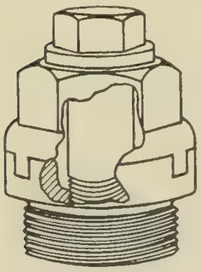


Wolseley water circulation pump.

Wolseley vehicles, and that the company have lost but little by their abstention from the production of a type giving a very high brake-horse-power in comparison with the cylinder dimensions is evidenced by their long-continued commercial prosperity.

Though the whole 16-20 h.p. chassis is well thought out, perhaps more care has been lavished on the engine than on other parts, and therefore we reproduce a full set of views showing the general arrangements, in part section, on pages 16 and 17. The cylinder bore is 3.9-16in., and the stroke 4¾in. The compression space is allowed a quite small variation, the maximum volume passed being 24.6 per cent. of the total volume swept out by the piston, and the minimum 23.4 per cent.

The cylinders are cast in pairs, and each pair is held down by the usual six studs, which also serve to position them. Though water is not circulated completely round each cylinder, the jackets are well proportioned, the large opening at the top of the casting, 3⅜in. by 1⅝in., and the wide pipe above it, giving a depth of water immediately above the top of the combustion head, making the formation of steam bubbles at this point extremely unlikely, if not impossible. The exit pipe is also given a gentle rise to the radiator, so the pump has merely to assist natural flow, and cooling ought to be satisfactory without it. The valve guides are cooled, as is shown in the sectional end elevation of the engine, and we have no doubt but that this precaution has a beneficial effect on the life of the valves and guide passages.

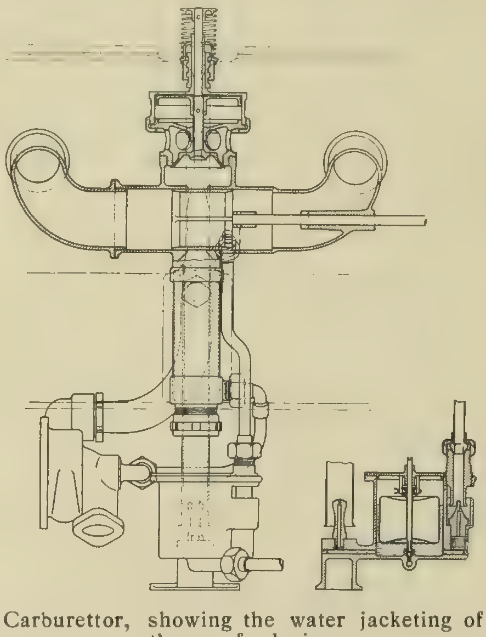


Method of removing valve caps.

The method of combining the caps, which close the boring aperture in the combustion head, with the studs that hold down the water pipe, is very neat, and it should be observed that in two cases the studs also serve to carry the ignition wiring case.

Every effort has been made to render the castings as simple as possible, and to make the machining of them complete. The hole cast in the top of the combustion head not only helps the founders, but enables a boring bar to be taken right through, and in practice each pair of cylinders is bored at a single operation on a double-bar machine. The top of the combustion chamber is faced, so that the only inside surface not machined is the small wall area round the valves, and this in common with the rest of the casting is thoroughly sand-blasted. The volume of the compression spaces are regulated by the shape of the undersides of the valve caps, and we understand that, owing to the almost complete machining of the surfaces of the cylinder, there is but little trouble in obtaining equal volume, for all four cylinders, within the fine set limits.

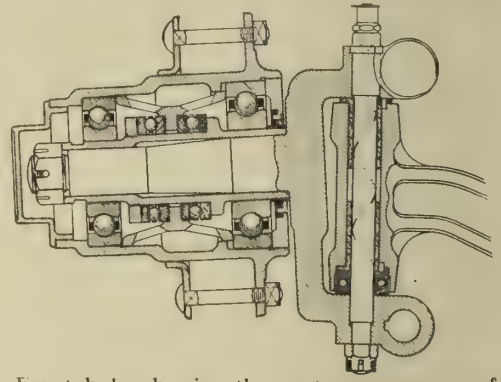
The valves have an area of opening equivalent to 7.8% of the piston area for the inlets, and 11.7% for the exhausts, these proportions having been determined by a long-continued process of experiment. It has been found that there is nothing to be gained by increasing the lift of the inlet valves to equal that of the others, and so the minor advantages of lower lift are obtained without sacrifice.



Carburettor, showing the water jacketing of the gas feed pipe.

The exhaust valve is set to open when the crank is 38° from the bottom position; it closes a little after the completion of the exhaust stroke when the crank has moved 7° on its downward travel. The inlet valve opens at a crank position 11° past the top position, and closes when the crank is 19° beyond the bottom. The tappets are steel, of round section, ending in tee heads, which bear against the cams. These heads slide in slots cut in the lower extremities of the phosphor bronze guides, and these parts are made of a strength sufficient to enable them to withstand the twisting which they may have to resist when the set screws in the top of the tappets are being locked. These set-screws are provided with fibre pads to deaden the sound of their contact with the valve stems, and the lock-nut is on an external thread, and not on the set screw itself; it is provided with a short sleeve concealed within the top of the guide.

There is a small point in connection with the valve caps, which is well worthy of mention. For their removal each car is provided with a hexagon cap, having dogs beneath to engage with the slots in the cap, and a central hole, through which a special short set screw is inserted and

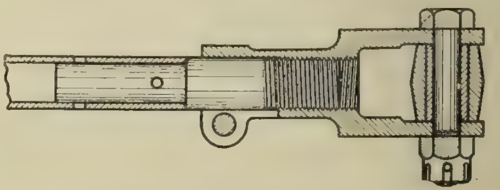


Front hub, showing the neat arrangement of the thrust washers.

fixed by the plug thread. This arrangement permits the use of shallow caps, which make for accessibility of plugs and compression cocks, and, at the same time, provides a more certain means of removal than can be obtained by the special spanners and devices in common use.

The pistons are cast iron, and the gudgeons are carried low down, so that, neglecting the rings, the area above the pin is equal to that below, and the pressure of the piston on the cylinder is equalised over its length. The small end has a phosphor bronze bush, and the gudgeon is locked by a taper set screw, which is secured from rotation by a split pin opened out against the side of the piston. A lip is cast on the bottom of the piston to act as an oil catcher, and each piston is provided with the customary oil groove below the ring level.

The crankshaft is 1½in. in diameter, and has three main bearings of an aggregate length of about 9in. The material is a special Vickers steel, which, we believe, is a nickel-chrome variety. The main and big end bushes are gun metal, lined with a babbitt mixture. The camshaft is formed in one piece with the cams, and the latter are ground finally in a special jig, and tested carefully to ensure that the time of valve opening and closing shall be within the prescribed fine limits. The



End of tie rod, with adjustment for setting steering.

bearings of the camshaft are gun metal simply, and of such a diameter that the shaft can be withdrawn endways. A skew gear wheel is also made solid with the camshaft, and of a size to allow its removal through the bearing houses, it being the means of driving the oil pump,

which is situated on the rear half of the upper portion of the crankcase. The distribution gears are situated in the front of the crankcase, the arrangement and the position of the water pump and magneto being shown in the plan of the engine. The gears themselves are all steel, and have single helical teeth, but otherwise call for no comment.

The engine lubrication system is one of the chief features of the car, and deserves to be studied very closely, as it is the original trough system, and the trough system seems likely to become very popular. The Wolseley Company were not the first discoverers of the trough idea, but they were the first firm to make really careful and elaborate experiments with it, and therefore they deserve to receive credit for such success as it achieves.

It is not our purpose at present to enter upon the arguments for and against the system, as a system, but the particular example of it we are about to describe is in all senses satisfactory, and designers in search of something better than plain splash and simpler than forced systems would do well to study it carefully.

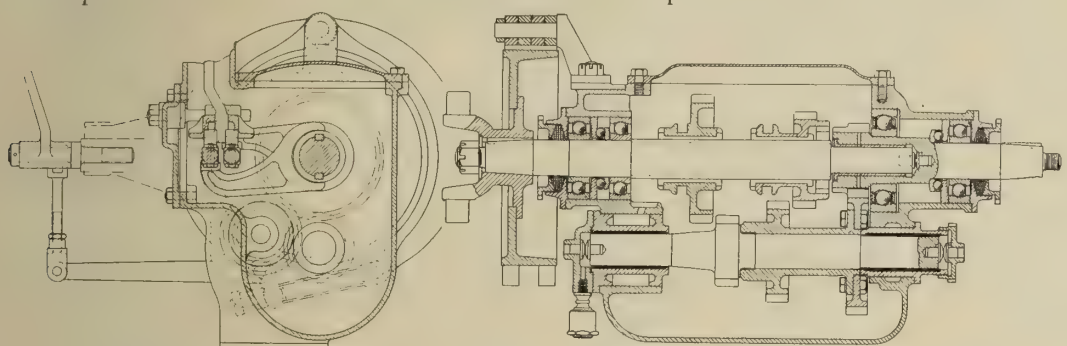
Under each big end is a trough cast in the aluminium base chamber of such a height that the body of the big end clears the walls well. Oil is fed to these troughs by a small gear pump, sucking from a sump, to which overflow from the troughs returns, and feeding oil to a horizontal pipe, whence it sprays through small holes into each trough.

The big ends are provided with scoops for the distribution of the oil. These scoops are shown in the engine section, and their small area should be observed, as well as the small depth of their dip in the troughs. The big ends, the gudgeons, and the camshaft are all supplied with oil flung by these scoops, but the main bearings are fed separately. The same pump as that which feeds the troughs also supplies oil to a pipe running along the outside of the crankcase, and the three main bearings are all fed direct from this. Oil from the pipe runs into a cup above each bearing, and overflow runs through a passage in the case, which conducts it below the bearing, and thence it returns to the sump. Needless to say, the oil is filtered on leaving the sump, it passing through a single thickness of gauze, in cylindrical form, arranged so as to be easily withdrawable for cleaning purposes.

Oil is also fed to the distribution gears by a lead from the pump, and runs back from them to the sump, but care is taken to prevent splash from the gears mingling with that of the big ends. A pressure gauge on the dashboard is connected to the pump end of the pipe which feeds the main bearings, and this serves to show whether the oil is circulating as it should do. The bearings of the outlying distribution shafts are also provided with grease cups.

To supply fresh oil a filler is fitted, and its position is shown in the plan of the engine. There is also a cock placed at the highest permissible level, and this is supposed to be opened when filling. Both these parts are easy to get to, even when the car is complete with body and shield, and the latter has a special hole cut in it to give access to the filter.

The advantages of this system of lubrication as carried out in this design are its simplicity and the small amount of driver's attention it requires. The disadvantages are that it is only after much trial and error that the makers have been able to determine the exact oil level, the power of the pump, the shape of the dippers, and the best means of preventing the main bearing feed from interfering with the splash action. The trouble is to get sufficient oil on the cylinder walls at all speeds, and never too much, and while this engine has no baffle plates between the cylinders and crankcase, it is instructive to note that some of the other Wolseley engines are so provided.



16-20 H.P. Wolseley gear, and foot brake. Scale, approximately 1 in. = 7.5 in.

Especial care is taken to prevent the exudation of oil from any bearings, and where these cannot be capped, then a stuffing box is provided. This is a refinement that might worthily be copied by other makers, for it has a double usefulness—it saves oil, as well as keeping the outside of the engine clean.

The carburettor has been designed mainly to give flexibility and the greatest possible power at slow speeds. It has two jets, the secondary one being contained in a special compartment connected to the inlet pipe proper just alongside the inlet pipe by a small pipe, ending in the throttle. As the throttle is opened an extra air is uncovered directly above the top of the vertical part of the inlet pipe, so that descending air meets ascending rich mixture. The order of firing is 3, 4, 2, 1, so there are two successive intakes up each branch of the inlet pipe. It is possible that this has an effect on the performance of the carburetter, as the alternating draught must assist the stronger mixing of the gasses. The air valve draws supplies through a ring of ports, and is controlled by a spring, the tension of the latter being in turn controlled by a dashboard lever, which operates a quick thread nut situated beneath the spring.

The bore of the main induction pipe is about $\frac{3}{4}$in. in the vertical part, and this portion is water jacketed. The upper duplex pipe is not jacketed, and has a bore of about $1\frac{3}{4}$in. The throttle is a piston open at both ends, and sliding in the horizontal pipe, so as to uncover jet tube and air inlet simultaneously. Petrol is fed by gravity, and the throttle is controlled by a lever above the steering wheel.

The exhaust and inlet pipes are held in position by four studs. The outside pair hold the exhaust pipe only, but the inner pair secure both exhaust and inlet pipes by means of bridges. The exhaust joints are furnished with copper and asbestos washers, but the inlet pipe has simple faced joints. The four nuts are brass, which prevents their rusting to the studs.

The ignition is performed by a Bosch DR 4. dual equipment, and the time of ignition is controlled by a lever above the wheel. This system requires the use of four plugs only, so the exhaust valve caps are provided with compression release cocks. The magneto driving arrangements are shown clearly in the plan of the engine.

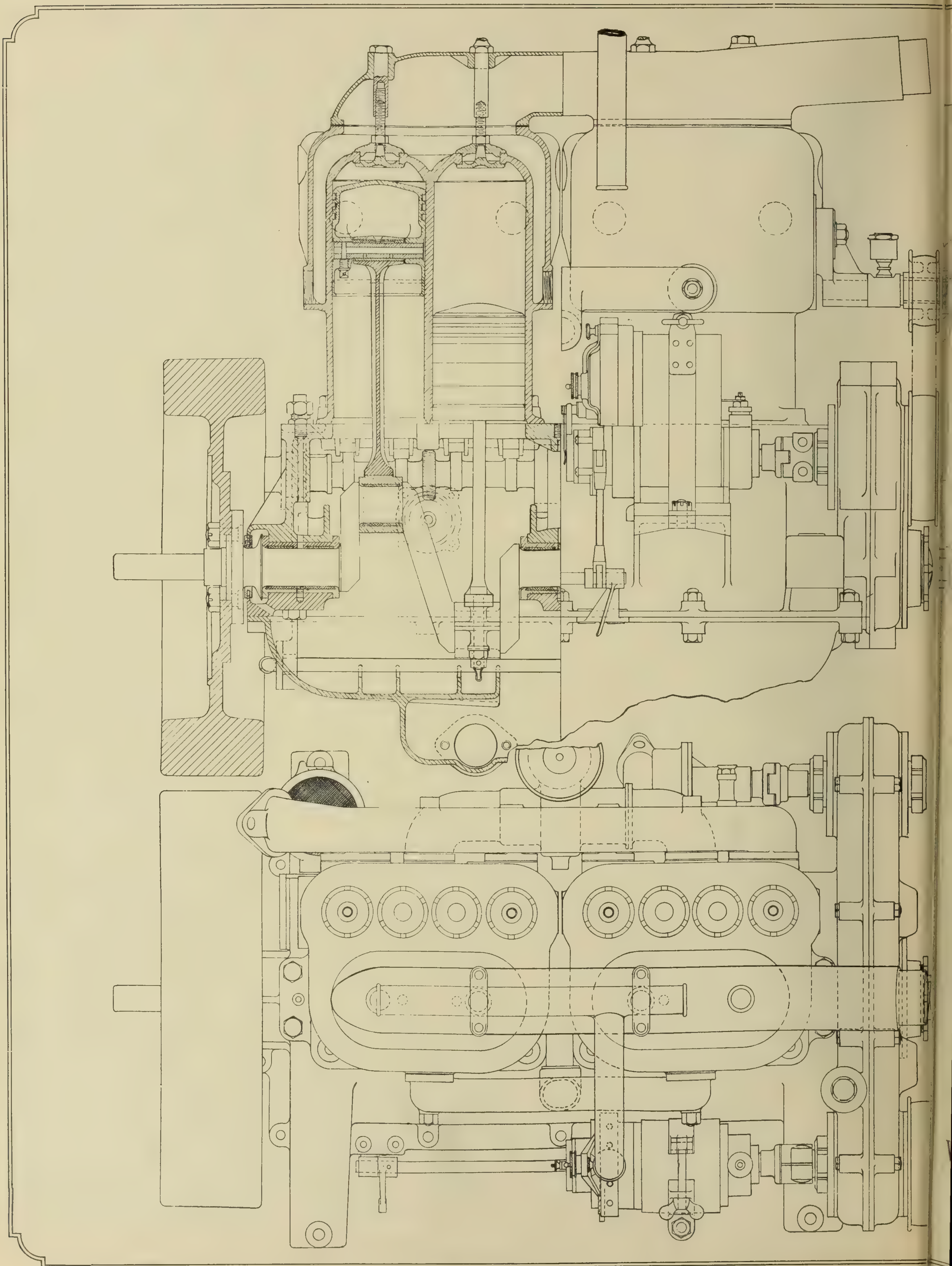
The water-circulating pump is of the centrifugal type, and the length of its gland should be observed. It should also be noticed that the grease cap is carried at a distance from the water sufficient to ensure the permanent viscosity of its contents, a point often neglected, but, none the less, of considerable importance to the user. The radiator is a honeycomb, and great care has been taken to provide for the complete draining of the water system, as there is a plug at the bottom of the radiator, and also a special pipe, ending in a capped nut, running from the bottom of the pump (the lowest point in the system) to a convenient position for manipulation. The fan is driven by a flat belt, and the tension of the latter is adjustable by sliding the spindle bracket on the studs, which secure it to the front cylinder.

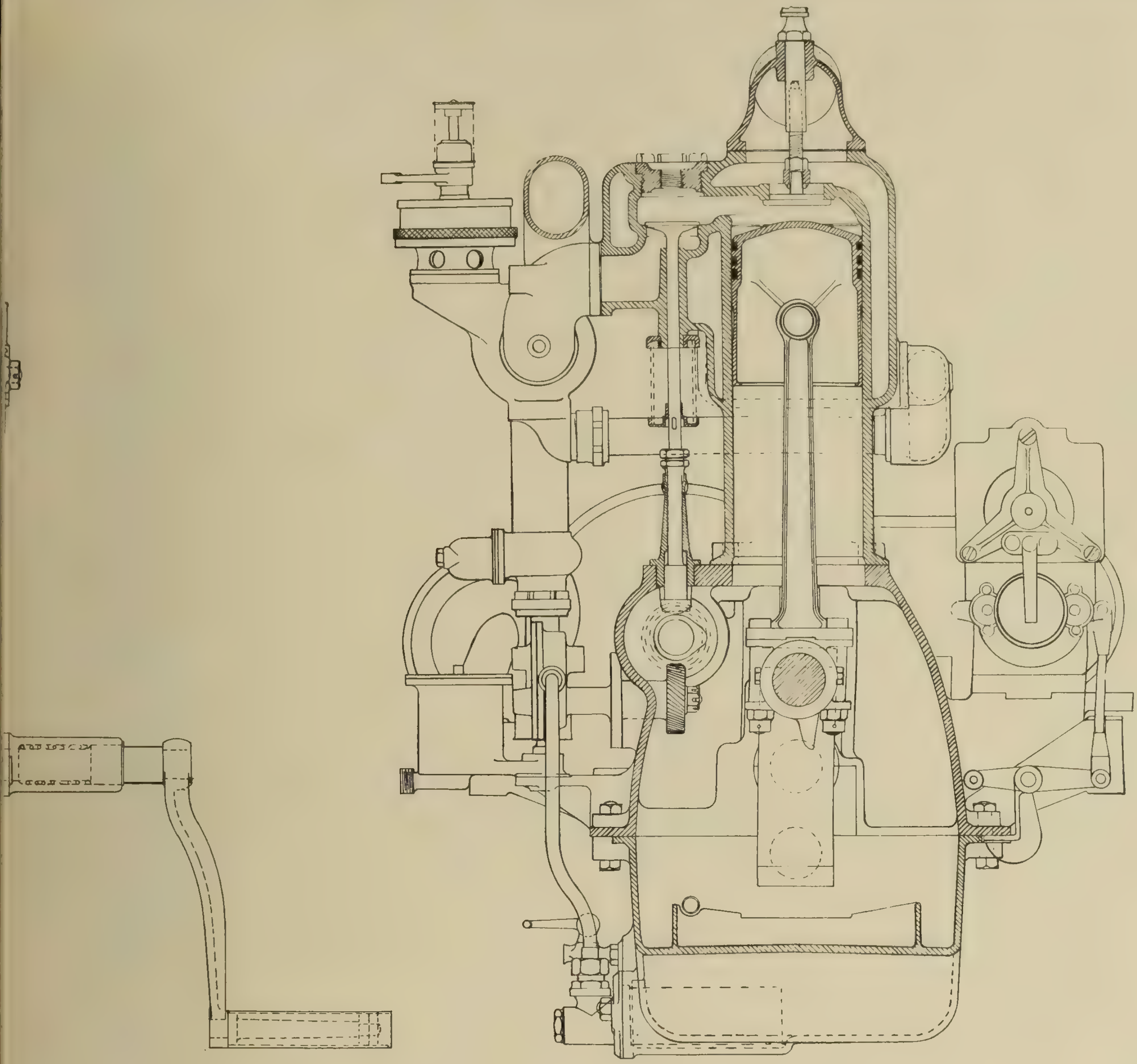
The engine, and the gear box also, is supported on an under frame of pressed steel, the section being channel with the open side downwards. At the ends the inner sides of the channel are opened and attached to the front and central cross members, this giving a wide bearing between the two parts, allowing the riveting to be very secure, and also making a rigid framework. The under frame members are also joined to the main frame by brackets situated close to the flywheel.

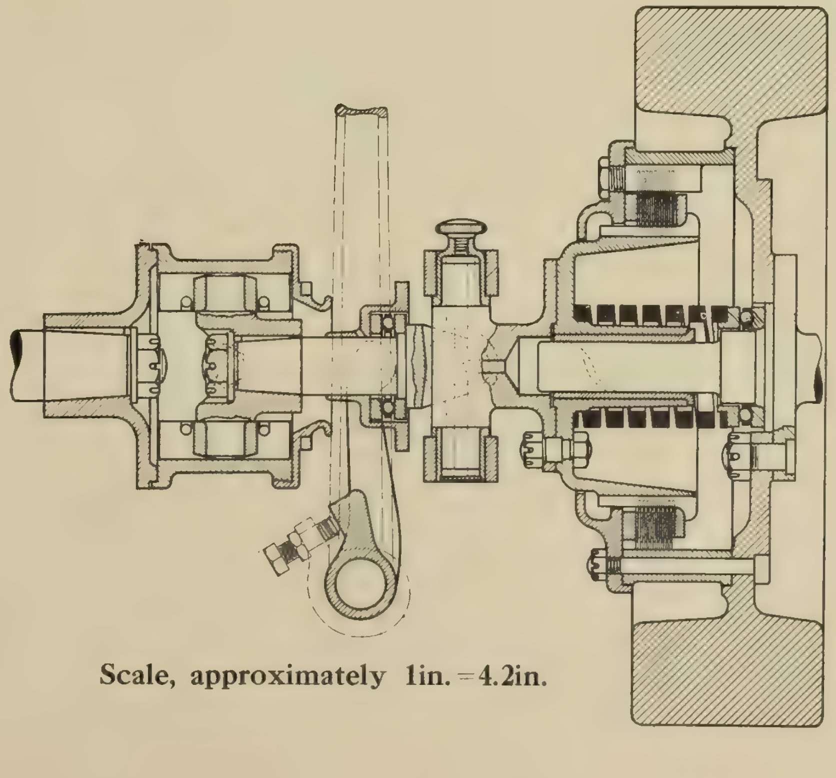
The clutch is carried on the crankshaft, and is almost completely explained by the illustration on page 16. The diameter of the plates on the inner member is $6\frac{3}{4}$in., and the inside diameter of the outer plates is $5\frac{3}{4}$in., while there are eighteen plates. The clutch case contains a fair quantity of oil, and a lead along the end of the crankshaft supplies oil to two of the ring universal joint pins.

The second joint behind the clutch release fork is presumably employed because it gives a very durable form of telescope, and it also makes the freedom between the crankshaft and gearshaft complete, which would not be the case if only the ring joint were used, for though this allows an inclination between the shafts, it does not permit them to run out of parallel, unless the two centre lines intersect at the middle of the joint.

The same design of joints are used for the propeller shaft, and the construction of the rear one is rather interesting. The front joint has simple hardened steel pins and phosphor bronze bushes, but the







Scale, approximately 1in. = 4.2in.

16-20 H.P. WOLSELEY ENGINE.

other is a composite structure of steel and malleable iron. The pot is malleable, but the bearing surfaces are hard steel strips, secured by the pegs shown, and they can be removed readily. The back flange is steel, and the square nuts are steel, hardened and ground. It might be remarked fittingly here that all pieces secured to shafts are steel, throughout the whole range of Wolseley models.

The ⅜in. set screw at the foot of the pedal, of course, sets the position of the pedal without affecting the total travel.

There is but little with regard to the gear box that calls for comment. It is to be regretted that the speeds are only three in number, as there can be no doubt about the benefit of a fourth ratio, and the increased cost of manufacture would be quite insignificant. The bearings are perhaps somewhat more generous in their proportions than is usual in cars of this power, for it will be seen that there are four ball races and two thrusts on the primary shaft, and two phosphor bronze bearings with a total length of just under six inches on the secondary shaft.

The shafts are of Vickers nickel steel, and the gears of Vickers case-hardening nickel steel. The teeth are of five and six pitch, and the speed ratios are 3.75 to 1, 6.4 to 1, and 12.75 to 1. The two second speed gears are not exactly similar, as they differ in size by one tooth. The gear-box is a single aluminium casting, and the bearings are all housed in gunmetal.

The striking gear is controlled by a lever working in a plain gate sliding a tube on the outside of the brake shaft, as usual. However, instead of being connected directly to the tube, the lever is hinged at the bottom, and thereby all tendency for the tube to bind on the shaft is removed.

The interlocking gear is a casting sliding on two studs inside the gear box, the operating lever working in a slot, and moving the casting into engagement with one or other of the striking rods, both being held when in the neutral position.

The reverse pinion runs on a plain bearing, and is in perpetual movement. Hardened steel thrusts are used to prevent any endwise movement of the shafts, and where the ends of the latter are not capped they are provided with adjustable stuffing boxes. The box is, of course, slung from the under frame.

The foot brake is mounted on the gear box, the steel pin, which carries the shoes, being secured in a large bracket resting on a platform on the top of the box, and secured by bolts. This arrangement ensures the concentricity of the brake drum and shoes, and is therefore good, but it is open to attack on the ground that aluminium is too ductile a metal to withstand continued brake strains. We are assured that no trouble has arisen so far with any Wolseley foot brake on this account, and presume that this is due to the large size of the platform, to which the shoe-carrying bracket transmits pressure. Certainly if the shoes were hung from a stud screwed direct into aluminium nothing but trouble could be expected, but with this arrangement the stress on any one part of the box is not excessive. The diameter of the brake drum is 8¼in., and the width of the shoes 2in.

The operation of the brake is shown in the end view of the gear box, and it will be seen that it is adjusted easily by means of a spanner. The inclination of the pedal can be set separately.

From the gear box the transmission of power to the back axle is by means of a double-jointed propeller shaft, the joints being of exactly the same pattern as those used between the clutch and gears. The front, ring pattern joint, has a clever encasement, illustrated in the article on universal joints, which appears on another page. The torque of the axle is resisted by a triangulated tubular rod slung from a ball joint of customary pattern, situated a little behind the forward end of the shaft. This arrangement does not give complete immunity from velocity variations, but is very slightly better than the single-jointed shaft, as is pointed out in the article to which we have already referred.

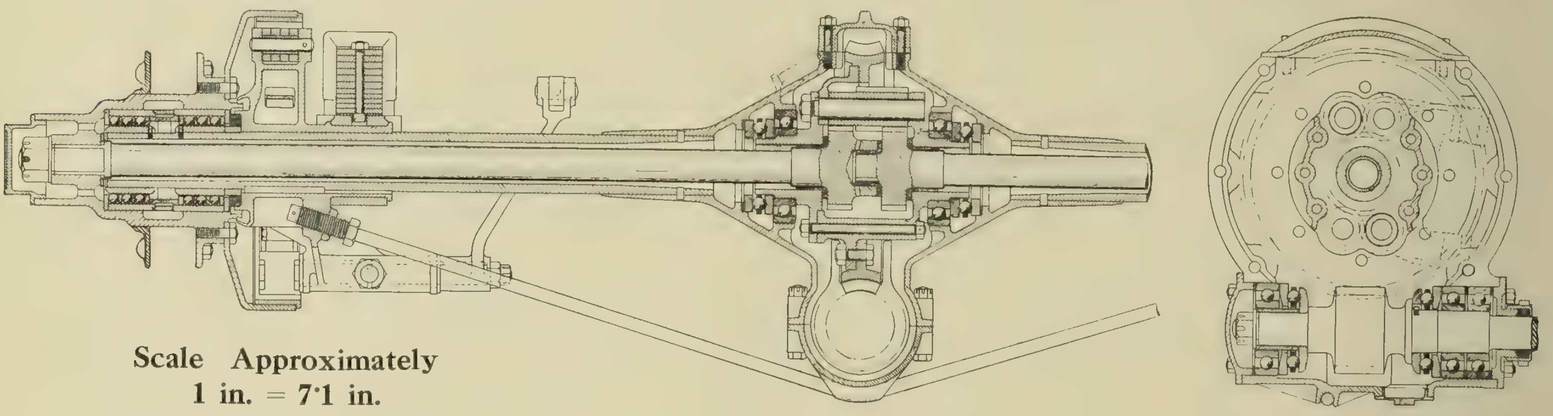
The prominent features of the back axle are the road wheel bearings and the ample size of the ball races used for the worm and worm wheel shafts. The road wheel bearings are made by the Wolseley Company themselves. The outer cylinders, the balls, and the separating rings all draw off the axle in one piece, and the balls are retained by enlarging the width of the separating rings on their inner circumference. The design lends itself to the fitting of detachable wheels, and has the advantage of being small in diameter.

The hub brakes are unusual in that the shoes are contracted by a horseshoe-shaped flat spring, instead of a helical one. This gives a powerful return, and also one that cannot be affected by mud or caked dirt, while rusting is less likely to weaken the springs. The diameter of the shoes is 11in., and the width 1½in.

The brackets carrying the brake shoes and spring pads are steel, are positioned by shoulders on the axle sleeves, and secured by keys. The oil trap formed at the inner end of the hubs should be noticed. The brakes are adjustable by self-locking thumbscrews situated at the rear ends of the pull rods, and they are accessible to a quite sufficient extent. The actual contact shoes are detachable, being cast iron attached by aluminium rivets, this metal being preferred to copper because it makes the renewal of the wearing surfaces a little more easy.

The lubrication of all parts of the axle is performed by the oil which must be supplied to the central casing at fairly frequent intervals. Lubricant can be inserted either by the removal of the inspection cover on the top of the differential case or through an orifice normally closed by a threaded brass cap. Grease has also to be supplied separately to the hubs and to the brake shafts, but the spring pad where it swivels on the axle is not provided with a lubricator, it being considered unnecessary.

We have now described the whole power and transmission system, and have next to consider the frame, springing, and steering gear. The first named is 4in. deep at the maximum, and is only very slightly narrowed from the dashboard forward. It is raised about four inches over the back axle, and, at the ends, it tapers to about half its maximum depth, this being equivalent to the average width.



Scale Approximately 1 in. = 7·1 in.

There are three cross members, one at each end and one at the rear extremity of the under frame. The side members end some ten inches behind the axle, and the springs connect to long dumb irons forged up with corner plates and riveted to the frame. The rear springs are four feet long, and the front springs three feet, while both are normally nearly flat, the difference in altitude between the ends and the centre being about a couple of inches.

The wheel base is 9ft. 10in. or 9ft., the track 4ft. 4½in., and the tyres are 820 mm. by 120 mm. or 815 mm. by 105 mm. respectively. All springs are shackled at the rear ends only, and every shackle and spring bolt is hollow, and provided with an end lubricator, which is supposed to be oiled with a syringe. We think that grease is to be preferred for this purpose, as it is cleaner and more lasting, and also because it is only one driver in a hundred who would take the trouble to use a syringe more than once a month, while most have sufficient care for their cars to keep greasers filled up, and many have a habit of turning all grease caps

with a frequency which is harmless if unnecessary. While on this subject we might remark that care has been taken on this chassis to provide a lubricator wherever lubricant is required, and that all but a very few are accessible. Among the parts which are liable to neglect are, as usual, the universal joints at each end of the propellor shaft, as their accessibility depends entirely upon the body builder, but both are easy to lubricate on the chassis. The compensation shaft for the hub brakes is also only disclosable by the removal of floor boards in the back of the body, as it is tucked away behind the middle cross member of the frame. No other parts are concealed, and such parts as drain cocks are uncovered where necessary by cutting the undershield.

The front axle is a forging of Vickers steel of H section, and it needs no comment. On the other hand, the arrangement of the front hubs is interesting, owing to the way in which the bearings are all carried in a housing, which slips on the stub axle, and has simply to be locked in place. The double-thrust washer is an excellent feature, and the neat way in which it is applied is equally commendable.

It is a well-known fact that Wolseley steering is exceptionally good, but there is nothing peculiar in the design to account for it. We understand that by calculation and experiment the exact inclination of the steering connecting rod has been discovered, which gives the least degree of steering movement (or flapping) due to spring flexion, and that the setting of the tie rod arms (which, when produced, do not meet at the centre of the back axle) has also been the subject of considerable research. No doubt the secret of steering lies in the correct setting of the usual parts just as much as in the improvement of the usual arrangements, and designers seeking a good steering gear will be likely to achieve more by experiments on the lines of those indicated than by the most accurate manufacture of wrongly-set parts.

The steering gear and column require no explanation, but are peculiar in that the control quadrant moves with the wheel. This seems to be a debatable feature, because on most cars the quadrant is fixed, and the moving levers make a Wolseley a little awkward to the novice. Taking the whole of the design of the 16-20 H.P. Wolseley, there are but few parts deserving condemnation and many which merit high praise. The design does not so much exhibit great ingenuity as extreme avoidance of extremes. Using the term "standard practice" in the usual loose sense, we think the chassis may be taken as an example of a first-class expression of the term. Certainly as far as workmanship and material go it would be hard to find a more conscientiously-constructed car, and there is also no doubt that every detail part of the design has been obtained in its present form after a series of, first, careful calculation, second, careful experimenting, and third, very accurate manufacture.

The debatable points are mainly the carburetting system, the trough lubrication, and the power of the engine for its size, as it is admittedly quite unsuited for competitions on the basis of customary rating formulæ. Leaving the first two counts as not coming within the scope of the present article, we might say while extreme power in comparison with size is, of course, a desirable feature of any engine, yet it has not so far been obtained without the disadvantage of "hard running," and there is a large class of car users who put comfort before anything else, and regard high speeds and violent accelleration as unnecessary. To these the makers of the Wolseley cars have always appealed, and it has undoubtedly proved a successful commercial policy Thus we recommend the careful study of the design we have just described to those who are interested in the production of a chassis of moderate brake horse-power to give all-round satisfactory service, as the mechanical portion of a carriage, for touring and for the general purposes to which a car of this type may be expected to be put.

CARBURETTORS FOR PETROLEUM SPIRIT.

Indicating the way in which future effort for improvement ought to be directed in order to obtain greater economy as well as greater power.

By Robert W. A. Brewer, A.M.I.C.E., M.I.M.E., M.I.A.E., F.S.E.

THE underlying principles of the modern high-speed internal combustion engine are practically the same to-day as they were fifteen years ago, several of the revolutionary features of design embodied in some of the most recent engines being really perfection in a practical form of ideas which had been tried without much success in gas engine practice.

One may say, without hesitation, that the greatest advances have been made in the direction of improved methods of producing the working fluid from liquid petrol and air.

Such results have only been attained by a careful study of the liquid fuel itself, its behaviour when treated mechanically, such as by a jet tube or deflectors, and the stability of the carburated air under the widely differing conditions of pressure and velocity to which it is subjected. This question of stability of an explosive mixture is a much more difficult one when the hydro-carbon is originally of a liquid than of a gaseous form, as sudden alterations of prevailing conditions of temperature or direction of flow may cause precipitation of the liquid, which in turn initiates a series of events possibly terminating disastrously.

Good practice in carburettor design should be such that uniformity of direction of motion of the carburated air is maintained under all conditions of throttle opening, and stream line form of entry to, and outflow from, the mixing chamber is adhered to. At the same time it may be necessary, under certain conditions, to produce turbulence in the vicinity of the jet orifice, when the throttle is so arranged, or the engine working at such a low speed, that the velocity of the air past the jet is insufficient to produce disintegration of the liquid.

It is quite conceivable that an engine may work at a low rate of revolutions with the throttle wide open, when the road resistance is high, and at such a time the maximum power output from the cylinders is desirable. The liquid petrol issuing from a jet orifice as arranged ordinarily, would be in the form of a fine stream, so presenting its minimum, surface area to volume, ratio for contact with the incoming air. Turbulence of either the air or of the liquid would be desirable under such conditions, and the means of attaining such an object should be studied.

Considering now the higher values of air and fuel flow, these may occur under two different sets of conditions, (1) the engine running at its maximum rate of revolutions, doing little work, and with the air passage round the jet restricted, say, by a throttle; and (2) the same rate of revolutions maintained with the choke tube round the jet fully open, allowing all the air possible to pass to the engine.

The above hypothesis holds good when the throttle is actually in the vicinity of the jet or mixing chamber, and its opening or closing directly affects the air velocity around the jet, and when no extra air is allowed to enter the induction pipe between the carburettor and the delivery to the inlet valves of the engine.

We may have, under the above conditions, the same air velocity prevailing, whether much or little work is being done by the engine, the weight of air being approximately proportional to the work done in each case, as it is governed by throttle opening. The amount of petrol flowing in case (1) will be excessive if no provision is made to counteract the ordinary tendency of the liquid to issue from the orifice.

A study of the relations between suction and air velocity through a tube, and of suction and petrol flow through a small orifice leads one to conclude that a certain mathematical variation holds good for all practical values.

Such being the case, it should be possible to design a mechanically correct carburettor, which would give an explosive mixture of constant quality at all engine speeds when working under the same conditions of road resistance.

This can be done either by working downwards from the design for maximum speed, or conversely. The writer prefers the former method of tackling carburettor problems in certain special cases, but finds that for ordinary touring work a basis of 20 miles per hour car speed is very convenient.

So far, the work is quite simple when the fundamenetal constants and variables are known, but the complex problems of the unknowns are difficult to tackle, and results are generally only arrived at by considerable experimental effort.

We may consider the following subjects as being the most difficult in carburettor problems :—

(1) Flow of the fuel under conditions of restricted throttle opening.

(2) Flow of fuel under rapid variation of throttle opening.

(3) Effect of inertia of the liquid in the jet and passage to the float chamber, causing lag of flow.

(4) Effect of sudden closure of throttle, the reverse of (3), causing continuance of flow.

(5) Shape and length of the induction pipe.

(6) Maintenance of correct temperature of carburated air from the mixing chamber to the engine.

(7) Prevention of precipitation of liquid petrol in the induction pipe.

(8) Most suitable sectional areas of choke tube, carburettor outlet, and induction pipe to suit a certain cylinder volume or piston area.

Considering the first conditions, the effect of throttle opening has been dealt with as regards its effect upon the flow of fuel when the throttle itself is situated in close proximity to the jet orifice, as when the throttle valve itself surrounds the jet. This is the case in some of the well-known carburettors, but in many others the throttle valve is at such a distance that its manipulation has the opposite effect, and closing the valve reduces the velocity of the air around the jet by restricting the flow of air at a distant point.

In such a case it will be possible to reduce the velocity of air round the jet to such a point that an insufficient quantity of petrol will flow from the orifice, to maintain an explosive mixture, or so to reduce the flow that it becomes erratic in its behaviour, and the engine stops.

Means must therefore be employed to increase the petrol feed abnormally at slow engine speeds, and this can either be accomplished by allowing a large jet area to be uncovered, or by increasing the local suction upon the jet orifice when the throttle is only slightly open. In the Claudel-Hobson type of carburettor with a shrouded jet, this is accomplished by shutting off all air supply outside the shroud when the throttle is nearly closed, and thus diverting the air stream into the vicinity of the jet. The effect is similar to what would be obtained were a variable choke tube employed. In other cases the carburettor is so designed that normally a high velocity of air is allowed around the jet orifice, and an extra supply of air is allowed to enter the induction pipe at high engine speeds.

Such arrangements are not entirely happy, as one cannot determine definitely the amount of air flow through a spring actuated or hand operated valve.

Some designers dispense with a valve or flap, substituting a slotted sleeve, having slots proportioned to allow the air opening to vary directly as the petrol flow does under the intensity of suction prevailing.

If such a design is correctly carried out it is admirable in every way, but the fact must not be lost sight of that as extra air is allowed to enter the induction pipe, or a portion of the carburettor away from the jet, the local intensity of suction at the jet itself must be affected.

Direct relation between jet area and air inlet area can be maintained by a mechanically connected air throttle and jet cover, such as is used on Napier cars. In this case the jet orifice is a narrow slot formed as an arc of a circle on the top of a jet tube of cylindrical form. A small pilot hole is pierced in the end plate of this tube, and a rotating sleeve cap, operated by the throttle, allows the petrol slot to become uncovered in proportion to throttle opening.

In such an arrangement we have two important functions working in conjunction, but by itself this arrangement makes no provision for the difference of pressure inside the carburettor with the same throttle opening at varying engine speeds. In other words, the same jet area is acted upon when the throttle is fully open at racing speed as when the engine is labouring on a hill.

The White and Poppe carburettor, which has been so successful, is in its

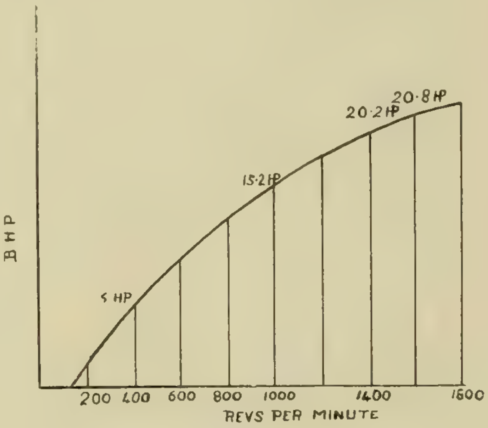


Fig. 1.

main principles a similar arrangement, the eccentricity of the two holes—that in the carburettor body and that in the rotating throttle—causing the petrol orifice to be varied in size as the throttle is operated. Also the air passes over the jet orifice at right angles to the direction of petrol flow, and by arranging the angle of the air orifice suitably there is no doubt that an excessive flow of petrol at high air velocities can be counteracted automatically.

Interconnection of air and fuel orifices is one step in the direction of perfect carburation, but it is evident that the third factor, already alluded to, must be reckoned with seriously.

The ideally correct control would be one in which both engine speed and throttle opening come into direct mechanical action automatically, so that, when the throttle was wide open and the engine speed slow, the volume of air passing through the carburettor could be concentrated round the jet by means of a governor actuating a variable choke tube, and, on the other hand, as the engine speed increased, due to a reduction of road resistance, the petrol flow could be maintained at the same proportional rate to air flow by a slight increase in sectional area of the choke tube, this being operated by a governor driven from the engine shaft.

Coming now to subject 2, we find that with ordinary carburettors a rapid variation of throttle opening affects the proper proportions of the explosive mixture, principally on account of the difference in the inertia of liquid petrol and gaseous air. The inertia of a petrol stream is considerable, and depends upon the mass of liquid in the stream and its velocity of flow. Where the stream is very short some of the objections disappear, and these can be further reduced by mechanical means. The issuing stream of petrol may be made to support a small weight or stopper, either by directly impinging upon it or by causing it to rise by passing the liquid around screw-shaped grooves in an extension of it. In some special circumstances, where fuel economy is specially sought, a damper is fixed under the direct control of the driver, or it may be operated by an air float riding in the mixing chamber itself, operated by the suction of the engine. (See fig. II.)

One of the earliest methods of checking petrol inertia and, incidentally, reducing the flow at high suctions was to insert a resisting medium, such as a spiral or bear's pole in the petrol passage itself. It is extremely difficult to design such a device and experiment will more easily determine the most suitable arrangement for any particular set of conditions.

There are two methods of determining whether any particular carburating apparatus is behaving properly when the machine itself is completed and working. The most familiar one is by an analysis of the exhaust gases, and this method has much to recommend it. When proper samples of the exhaust gas are taken, analyses of these will show whether the combustion has been complete or not, or whether an excess of air is present. Such a method takes considerable time, for after each analysis a change may be necessitated in the carburating apparatus and the whole process of testing has then to be repeated.

If, however, the design of the apparatus is an entirely new one, a preliminary series of tests will be required, viz., those of the liquid itself, and its property of flowing through the jet orifice in the proper proportion to the air entering the mixing chamber or the induction pipe. The writer has made numerous experiments with jets discharging in the same direction as the air stream when an unobstructed passage is provided, but a great deal remains to be investigated with regard to flow of liquid fuels under other conditions.

Knowing the proportions of liquid petrol required to be mixed with air to form a working fluid, calculation of both air ports and petrol orifice can be proceeded with.

If we take, for instance, a petrol consisting of 90% C and 10% H, such as is probably the constitution of the bulk of the fuel sold in this country to-day, we have :—

1 lb. H requires 34.8 lbs. air theoretically.

0.10 lb. requires 3.48 lbs. air for complete combustion.

1 lb. C requires 11.6 lbs. air theoretically.

0.9 lb. C requires $11.6 \times 0.9 = 10.4$ lbs. air.

$10.4 + 3.48 = 13.88$ lbs. air, per lb. petrol, theoretically.

At atmospheric temperature and pressure this represents $13.88 \times 13.4 = 186$ cubic feet air per lb. of petrol.

Having determined this quantity an additional amount must be allowed, depending for its magnitude upon the nature

of the carburating apparatus employed, the system of the induction pipe and the speed of the pistons. This means that when the time element is small there must be an excess of air to permit each molecule of hydrocarbon to associate with its necessary amount of oxygen, and unless the petrol liquid be reduced to an exceedingly fine mist, this association process will be incomplete, except when an excess of oxygen is present.

Supposing an initial excess of 50% is allowed, the total air admitted will be 280 cubic feet per lb. of petrol. The writer usually allows an air velocity of 200 feet per second past the jet orifice for normal engine speed, and taking this figure as a basis, the area of the air-way can be calculated for any known cylinder dimensions. This air velocity will produce a difference of pressure between the jet orifice and the float chamber of 9.5 inches of water head.

Calculating for normal engine speed, and this difference of pressure, the jet tube can now be designed so that it will discharge the correct volume of petrol equal in weight to 1 lb. for every 280 cubic feet of air passing.

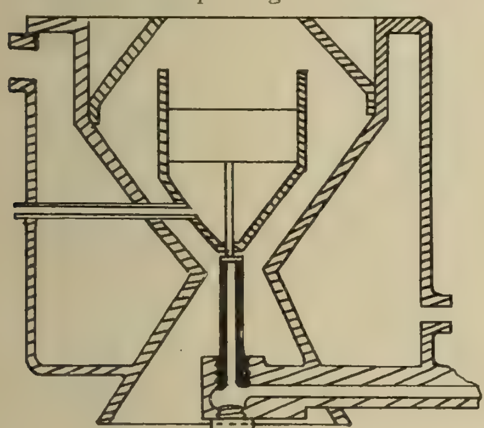


Fig. II.—Piston damper.

With direct suctions these orifices will usually discharge at the following rates, when the temperature of the liquid is 15° C. :—

Dia. of orifice	Discharge in gallons per hour
0.90 m.m.	1.2
1.00 ,,	1.35
1.05 ,,	1.54
1.10 ,,	1.70
1.15 ,,	1.90
1.20 ,,	2.14
1.25 ,,	2.35
1.30 ,,	2.70

The figures given in the preceding table are for steady suctions and take no account of variations of throttle opening or inertia of the liquid. Although in some cases this inertia effect is very marked, the mean flow of petrol over a given time will be almost constant, even when the suction varies from time to time, if the mean suction can be computed.

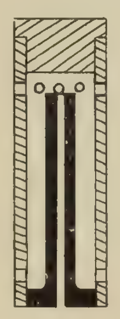
In other words, if a car travels, say, 20 miles in one hour, though its speed, from traffic interruptions and other causes, is varied, providing the same gearing, say direct, is employed all the time, the petrol consumption with a good type of carburettor will be approximately the same as though a constant and regular speed of 20 miles per hour had been maintained. The curves shown in fig. I. depicts the power developed by a certain engine at various rates of revolution, and it will be noticed that the consumption per horse power hour is practically constant at all powers for that particular engine. Naturally, if an engine be over-driven and its rate of revolution increased much above that for which it was designed, the petrol consumption will be abnormally high. It may be inferred that in such a case a great proportion of the fuel is ejected from the exhaust in an unburnt state.

The shape of the induction pipe has a large bearing upon efficient carburation. It is essential that the gas flow should be regular and conform to stream line configuration as far as possible. The size of the pipe should, therefore, be uniform or only slightly tapering, and the outlet from the mixing chamber should be preferably of Venturi formation, and thus assist in the complete atomization of the liquid. From this point the velocity of the gas should not decrease; it should be as far as possible uniform and free from great pulsation. The presence of an obstacle or sharp bend will result in the precipitation of liquid petrol, which, at this stage, is probably only held in suspension by the air, in the form of a fine spray.

In systems where a saturated vapour is admitted into a mixing chamber by a small pipe from the carburettor, any such precipitation becomes fatal. Saturation means essentially that the air cannot take up any more hydrocarbon, and that which may be deposited cannot be carried along by the following vapour, and only remains in the pipe or drains back into the carburettor.

One well-known type of carburettor which has been fitted successfully to the Wolseley Co.'s cars is a two-jet pattern of the saturated vapour type.

In this the petrol is injected into a tube of small bore with the object of producing a saturated vapour, which is conveyed thence some distance to a mixing chamber situated near the cylinder heads, which is also a throttle chamber. It will be gathered from the foregoing remarks that saturation of air depends on the temperature and humidity of the air



Claudel-Hobson shrouded jet.

Bears pole retarder.

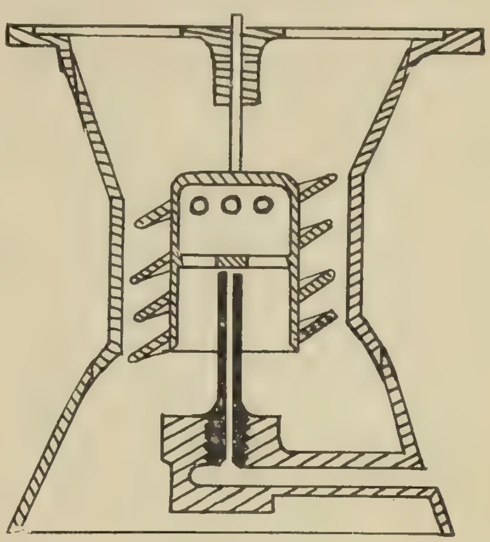
itself, and upon its pressure, and that when working near the saturation point difficulties are likely to occur due to instability caused by these varying factors.

The insertion of gauzes and baffle plates often produces the effect of arresting the liquid, and in many cases such obstructions have no good or useful effect in assisting the carburation process.

One of the most important features of the induction pipe is its temperature, and this should, in no case, be allowed to diminish between the carburettor and the inlet valves.

Any drop of temperature will aggravate any tendency to condensation, and it is preferable to jacket this pipe above the carburettor, particularly for fast-running engines.

One cannot conceive that the carburation process could be completed during the short fraction of a second that the



Whirling damper.

air takes to pass through the carburettor itself. The probability is that it goes on all the way along the induction pipe, to be completed in the cylinder itself during the compression stroke.

The latent heat of evaporation of the liquid must be borne in mind, and sufficient heat supplied from external sources to compensate for this. A certain drop of temperature is permissable and often desirable, but this should not be so great that the temperature of the mixture falls below its freezing point.

The lighter fractions of ordinary petrol may be considered to be heptane, and we find that, when mixed with its theoretical quantity of air, this fuel cannot exist as vapour at a temperature below 3.6° C.

If an excess of 20% of air above the theoretical amount is present, this critical temperature is reduced to 0.7° C., and by adding 40% of air above the theoretical amount we find that —2.0° C. is the critical temperature. If the normal temperature of air inlet be 15° C., and this drops to 0° C., we have a loss of 27 B.T.U. per pound of air passing.

Taking the latent heat of heptane as 240 B.T.U. per pound, and 20 pounds of air per pound of fuel, i.e., an excess of 50%, we have $\frac{240}{20} = 12$ thermal units required per lb. of air entering the carburettor.

Taking the specific heat of the air = 0.260, we have $\frac{12}{0\ 260} = 46°$ Fahr., i.e., 25.5° C. drop in temperature.

In order, therefore, to prevent freezing, all the incoming air should be at a temperature of at least 25° C. when it is the only source of heat supply.

However, carburation, as has been seen, must be carried out very rapidly, and the temperature drop becomes very much greater. Heat must therefore be added either to the liquid or to the air in the most effectual manner possible. There is no doubt that a great proportion

of the carburation is effected in the cylinder itself, and, where concentric inlet and exhaust valves are fitted, the temperature of the exhaust valve is only kept within reasonable limits by the heat given up to the incoming mixture.

In reviewing the numerous carburettors in practical use at the present time, I cannot pass lightly over two important types, viz., the multi-jet Trier and Martin and the multi-air-inlet G.A.

In the former the mechanical proportioning is carried out in a somewhat similar manner to types already considered, but is carried on into three independent stages with the extra important addition of an air port, whose aperture can be increased at will, after the throttle is fully open. The manipulation of such a port is a matter of individual approximation, whilst the earlier movements carry out mechanical proportioning of air and fuel opening. This extra air opening at the full throttle position overcomes the objections to mechanically-connected arrangements referred to previously.

The G.A. carburettor, with a series of steel balls resting upon circular air apertures, is very simple, though it has some undesirable features. Gravity acts against the engine suction, but the balls are liable to stick on their seats.

In this carburettor the throttle is quite independent, and has no direct effect upon either air or fuel orifice. The air stream is bound to vary and pulsate, owing to the unstable condition of the balls, and it is unlikely that a uniform mixture could be produced by such a device.

In conclusion, I am bound to admit that a really perfect carburettor is yet to be designed, and in spite of the great talent which has been devoted to the subject, the problem presents many complications, which are difficult to tackle in any one simple apparatus.

The essentials of a good commercial carburettor are that it should be :—

1. Free from springs.
2. Easy to fit.
3. Of small dimensions.
4. No complicated adjustments required. One operation should suffice to tune it up to any particular engine.
5. Effectively jacketed or supplied with the necessary heat.
6. Mechanically simple and free from moving parts.
7. Easy to clean.
8. Able to produce a uniformly proportioned mixture of hydrocarbon and air under all conditions of working.

Such apparatus as we now have at our disposal do not embody all the requirements, and can only be said to effect a compromise, some in one direction, and some in another.

Mechanical mixing carburettors have been tried, but the quantities of liquid fuel to be dealt with per cylinder charge are so small that the least derangement or wear soon renders them ineffective.

A wide field is open to the inventor of a really good petrol carburettor, and then comes the still more difficult solution of the paraffin carburettor problem.

THE 8 H.P. ROVER DESIGN.

A detailed consideration of the construction of a commercially successful small car of moderate cost.

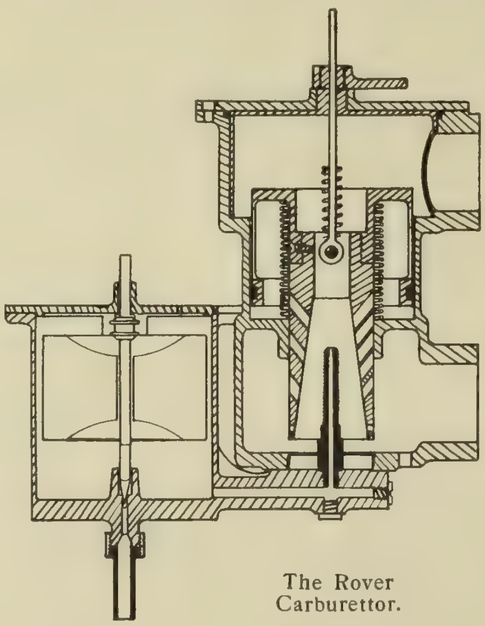
Although the 8 h.p. Rover is in many respects a peculiar car, and although a large number of the same type have been made and sold during the last few years, yet the design has been copied by other manufacturers to a remarkably small extent. In these days, when the sincerest form of flattery is so rampant in automobile design it is rather difficult to see why this is so, epecially as there is nothing in the whole car that is not cheap and easy to handle in both the foundry and the machine shop.

Possibly the explanation lies in the fact that the low compression slow speed engine with a single cylinder, 114mm. bore by 130mm. stroke does not appeal to the imagination of the modern school of voiturette designers whose efforts have lately been concentrated on the production of small four cylinder types.

The main characteristics of the chassis are the slow speed engine, the unit construction of the crank-case, clutch case, and gear box, and the suspension, which is three point, a transverse front spring being used. The frame is ash $1\frac{3}{4}$in. deep by $1\frac{3}{8}$in. wide, and the side members are strengthened by flitch plates mounted inside the ash. From the dashboard to the rear end these plates are $2\frac{1}{2}$in. deep, but the front ends are narrowed to $1\frac{3}{4}$in. or equal to the depth of the ash. The rear end of the frame is also ash of similar section to the sides and strengthened with a flitch plate $1\frac{3}{4}$in deep. The plates being all of the same thickness, namely 3-16in.

There are no cross members in the accepted sense of the word, except the back one, because from the point of attachment of the front end rear spring hangers the stresses usually resisted by a frame are borne instead by the aluminium engine and gear unit. This unit is bolted to a cast aluminium bridge of channel section of which the ends are attached to the side members of the frame alongside the spring hangers, and also to the end of the gear box.

The front end of the aluminium unit is bolted to a malleable iron channel section piece, called by the makers the nose piece, and shown in half sectional side elevation in the general arrangement of the unit. This nose piece carries a single large bolt, shown in the same view, and on this bolt it is possible for the pad of the transverse front spring to rock. Thus the engine



The Rover Carburettor.

and gear are directly supported at the centre of the front spring and at the forward ends of the rear springs.

The front part of the ash frame has, therefore, merely to support the body, the wings, and like fittings, and plays no part in sustaining the weight of either engine or transmission.

Such a form of construction would certainly be far from good for a heavy car, but in this instance it has stood the test of several years' use and its continuance is sufficient proof of the fact that the makers have had no trouble from it. It should be noticed that there are a couple of tie rods disposed close to the bottom of the clutch case, joining the crank case to the gear box (only one of these rods is seen in the elevation). We are informed that these rods were added because it was found that without them there was a tendency for the flange joints at each end of the clutch case to open, after a car had been in use for a few months, and that their addition entirely removed this trouble.

The way in which the bracket which supports the rear end of the unit forms an integral part of the gear box is made clear by the same figure, and the satisfactory nature of aluminium for the purpose is proved by the fact that in practice no trouble arises from it. This is no doubt due to the ample section of the arms, and to the continuation of the channel right across the casting, the gear-shaft coming through the centre of the channel at the point where it is deepest, namely, 116mm. outside measurement.

In addition to the rear brackets and the nose piece, the unit is also attached to the frame sides by a steel plate, the full frame width, about 114mm. wide, and about 6mm. thick, but this plate carries the dashboard, and owing to the light section of the frame, forward of this fitting, the plate cannot support much of the weight of the unit by the transference of stress to the side members.

The attachment of the rear springs to the frame is accomplished in the customary manner, the after ends being shackled and the forward ends secured in malleable brackets. Dumb irons carry the rear end shackles, the former being bolted to the frame by three bolts apiece. The corners of the frame being strengthened by corner plates it is not considered necessary to attach the irons to the back cross member, as well as to the side members, and as the overhang of the springs is only about 20mm. the twist on the sides of the frame is not particularly severe and the corner plates should be quite capable of resisting it.

The rear springs are about 900mm. in length, 45mm. in width, and there are five plates in each, the average thickness of the individual plates being between 6mm. and 7mm.

Details of the front spring are shown in the view of the front axle arrangements, and we need therefore say no more about it here, except to add that the width of the spring is about 45mm.

The front axle has an interesting and distinctly bold type of steering swivel, which appears in section in the view of the end of the axle. The whole of the load is supported and the side stresses are also resisted by the steel balls, which are half an inch in diameter. It will be noticed that each ball lies between two conical cups, each cup having annular contact with the ball. The surfaces of the cups are all case-hardened to a good depth, and adjustment of the bearing can be made by rotation of the upper cap, the top end of the swivel fork being split and provided with a clamping screw for locking the cap.

It will also be seen that the stub axles and the swivel pins are separate pieces instead of being a single stamping, as is frequent, and this is a design which has a good deal to recommend it. In the first place there is no difficulty attendant on the use of very tough steel for the stub axle, which is a part that requires the maximum of strength. Then the swivel pin is simple and cheap to make, while (of importance in this particular instance) the ends of the pin can easily be hardened to receive the ball without any risk of affecting the properties of the stub axle steel. Finally, the steering arms also being separate, can, owing to their simple form, be made of a material having somewhat greater strength than usual. It might be urged that the built-up construction is less advantageous practice than the solid one, because of the ever present possibility of built-up structures coming apart, but against this must be set the manufacturing advantages of the three-piece design, and the liability to breakage of the latter would be slightly smaller than that of the single piece, other things being equal, because the material is subjected to less trying manufacturing conditions.

On the tubular front axle two collars are shown, spaced 700mm. apart, and at equal distance from the centre. These collars are brazed in position, and each carries the half of a pin joint. The other halves are the ends of a pair of tubes 16mm. in diameter, which converge and terminate in ball joints mounted on opposite sides of the clutch case. The purpose of these rods is, of course, to radius the ends of the front axle, which is necessary owing to the transverse spring. The behaviour of this spring when stressed from only one end can be seen easily if it is remembered that the pad is quite free to rock on the big supporting bolt, so that the whole length of the spring is available for the absorption of a one-sided shock, as well as for a shock affecting both wheels equally. This rocking motion is constantly taking place, and for lubrication a small brass casing is used to carry a screw-down grease cap in a convenient position just beneath the radiator.

The radiator is supported by the forward ends of the side-members of the frame, and the fan-carrying bracket is mounted on top of the front end of the nose piece. The fan itself is an aluminium casting, the insertion of copper rivets in one blade serving to secure balance.

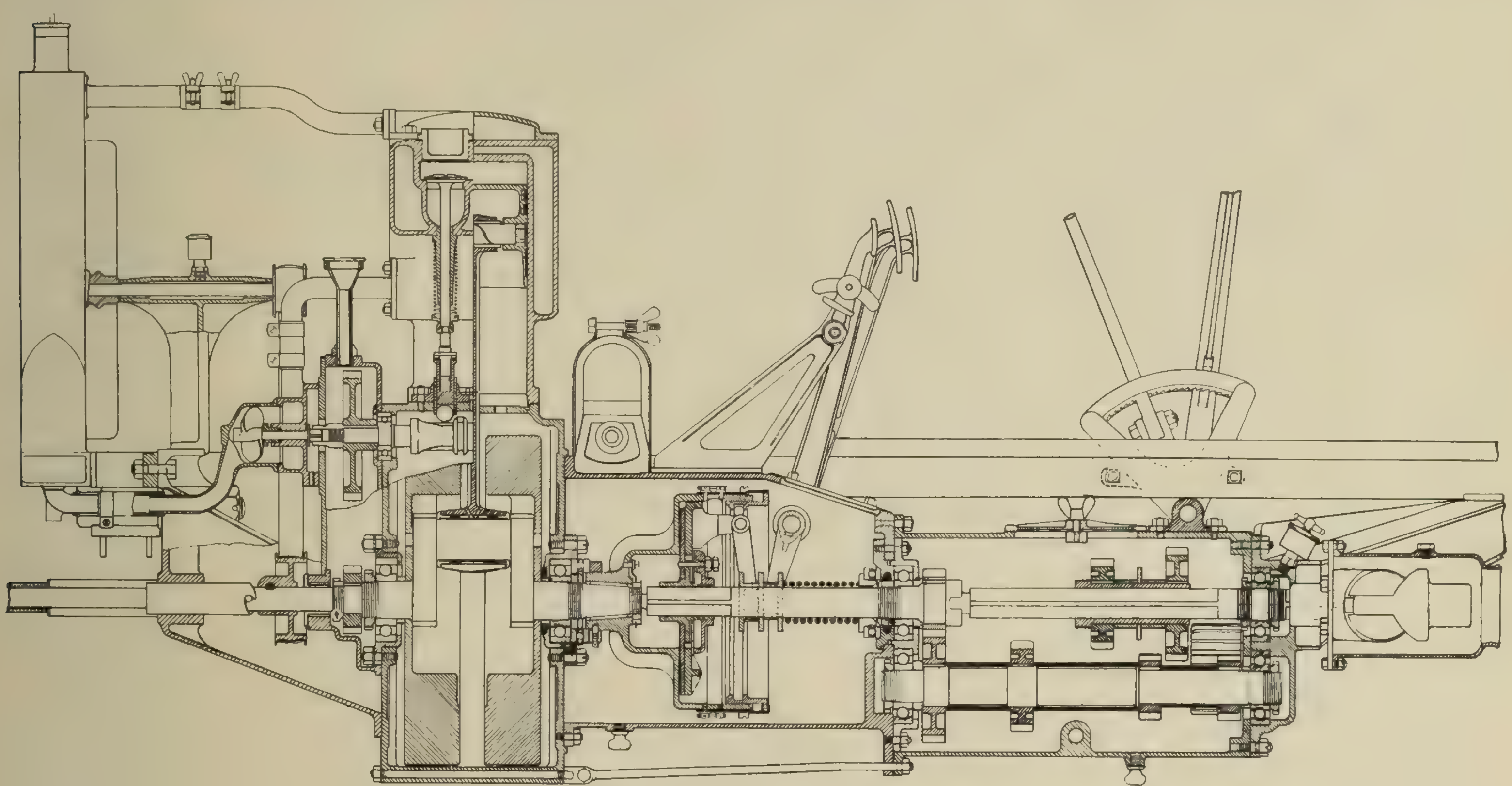
Having thus described the disposition of the frame and springing we can proceed to consider in detail the engine and gear unit, which is, perhaps, the most interesting part of the chassis. There is nothing of particular or peculiar interest regarding any detail, but the unit as a whole is instructive, because it is almost entirely dependent on a 98% aluminium alloy for its strength, and yet is certainly satisfactory in use, though, at first glance it would not appear to be well calculated to withstand the bending stresses which it is required to resist, and which must be fairly severe.

The splitting up of the case into three separate compartments probably has no bearing on the strength, and, while it increases the number of machine-shop operations necessary, it also makes for simplicity in machining, as there is no part that cannot be faced by the simplest of tools; besides, each part being small, there is a slightly smaller chance of bad castings occurring, and there is not the same temptation to pass a casting that is neither wholly bad nor wholly good.

Two thoroughly good points are the way in which the ball races are carried in housings of malleable iron, and the careful manner in which the inner races are secured to the shafts. It seems a pity that the iron housings were omitted in the case of the bearings at the end of the gearbox, though, of course, there is not the same need for them as these races are almost free from shock.

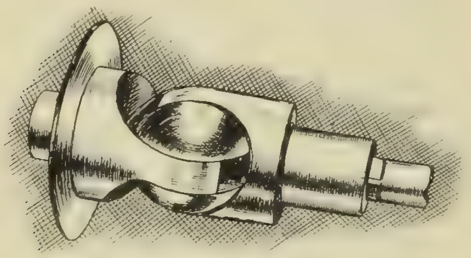
The engine itself is admittedly modelled on the old eight horse power De Dion Bouton. It has a stroke of 130mm. and a bore of 114mm. and is rated at a nominal 8 H.P. The compression is designed to be about 66 lbs. to the square inch, at its maximum, so it is easy to obtain slow running and the maximum speed is not very high. The normal speed is given as being 900 r.p.m., and the maximum as 1,600 r.p.m., while the minimum speed obtainable with the usual carburetter and ignition, is under 150 r.p.m., a very desirable feature in a car with so large a single cylinder.

The cylinder is held down in the usual way by studs, and calls for no special comment, except that it should be capable of improvement by enlargement of the water intake and exit. The area of valve opening is just under a thousand square cms., when at its maximum, and, as this represents close on one-tenth of the area of the piston, it is in a proportion in keeping with first-class practice. The valve operation is one of the features of the car.



Engine Clutch and Gear Unit.

and is easily followed by reference to the general arrangement. The exhaust cam proper and the inlet cam are of the same dimensions as regards total lift, but the cams can be slid by depressing a pedal, and when longitudinal movement takes place the inlet valve is allowed to remain shut continuously while the exhaust valve engages with a secondary cam, causing it to open at the top of every piston stroke, to remain open during the down stroke, and to close at the bottom. This action converts the engine into an air compressor and a smooth, but powerful braking action is the result. Needless to say, the sleeve and cams are one solid piece, and



Unique pattern of universal joint.

we consider that it is much better to arrange the parts in this manner rather than to use the alternative method of sliding the whole camshaft.

The construction of the tappets should be noticed, as it illustrates an excellent way of introducing a fibre pad for deadening noise. The small spring in the centre of the tappet keeps the top piece in perpetual contact with the valve, and this top piece is separated from the lower half by a fibre ring through which the pressure applied to the ball end is transmitted. The ball is $\frac{3}{8}$ in. in diameter, and its use is doubtless due to the presence of sideway slide. The extremely narrow area of its contact with the cams is not exactly good, but if the latter are sufficiently hard there is no reason why the durability should not be satisfactory.

There are only a few other points with regard to the engine which require further elucidation than is made obvious in the drawing. The attachment of the cast-iron flywheels to the crankshaft should be noticed, and we may add that the crank, connecting rod and piston are balanced by the simple expedient of drilling out the opposite rim of the wheels.

The piston is peculiar on account of the ring arrangement, there being two rings in each groove. This is a practice which the Rover Company inform us has been found satisfactory, and has been in use for some time. If the fittings of the rings is carried out with care there is no obvious reason why they should not behave as well as when separated, but on the other hand, there is no particular object in the design, especially as it has been demonstrated conclusively that there is no necessity for more rings than three, the fourth merely causing extra friction.

The gudgeon pin fixing is good and easy to follow from the drawing. The bush and pin being both steel, while the big end bush is gun-metal lined with babbitt.

It should be noticed that oil excluding or separating washers are used to prevent the entrance of oil from the clutch case to the crank chamber, and that there is a gland round the shaft where it emerges from the distribution gear case. There is, however, a free way for oil through the front main bearing for feeding the gears.

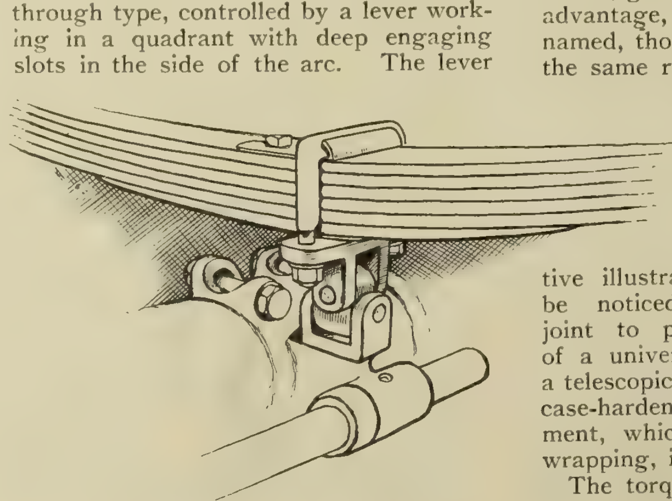
One small point which calls for adverse criticism is the position of the pump gland. The pump is a simple two-bladed propeller and is enclosed by an aluminium cap held in place by a dog mounted on two long studs projecting from the front of the distribution gear box. In order to tighten the gland it is necessary to remove this cap, which means letting out all the water from the cooling system, and although any leakage along the shaft is trapped in a chamber before it reaches the crankcase, the placing of the gland adjustment in a more accessible position would be a valuable improvement to the car as a whole.

Another small criticism which might be made regarding not only the engine, but the whole unit, is that there is a free use of studs depending for their security upon a comparatively short thread in the aluminium. Failures on this account are no doubt rare, but it is a form of construction better avoided whenever possible, and usually easy to avoid.

The clutch is a pattern which has been used for Rover cars for some years, and it is satisfactory in action when the plate separating device is set so as to clear the surfaces beyond the point where oil can cause them to stick together. The main body is, of course, cast iron, and the inner part of the clutch is made of the same material, while the central plate is steel. The action of the engaging levers is fully explained by the drawings and we need only add that adjustment is performed by turning the ring, the groove in which forms the fulcrum for the levers, of which there are three. The advantages of this clutch are that it is not costly to make, and it is durable as well as being sweet in action. Its disadvantages, that it is a little difficult to handle if the oil is of too stiff a consistency and, that when worn it is not so easy to repair as are clutches where the main body itself is not used for a working surface. Also the striking gear is liable to develop a good deal of slack. However, as the normal life is so long, the advantages should more than counterbalance the drawbacks.

The casing of the clutch is neat, and the large inspection door allows all adjustments to be made without dismantling, though to break the unit is easy when the latter is detached from the car.

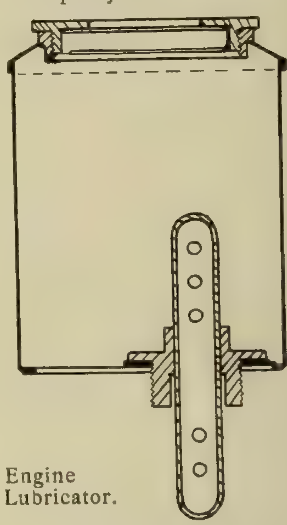
The gears also present no special point for comment, being of the simple straight through type, controlled by a lever working in a quadrant with deep engaging slots in the side of the arc. The lever rocks on a pivot carried on the outer end of the shaft, and it is forced against the quadrant by a spring which is compressed in a socket as the lever is pressed outwards.



Swivel mounting of rear springs.

The cross tube (which turns on the brake shaft as usual) carries a lever connected by a link to another fixed to a shaft running across the bottom of the box. This shaft carries an internal arm and fork, by means of which the slide is operated.

The method of securing the gear wheels to the sliding sleeve and secondary shaft is shown in the drawing, and both sleeve and shaft are, of course, circular. It will be seen that the assembling is simple, and the necessary fitting equally so, while the replacement of secondary shaft gears is a simple job.



Engine Lubricator.

The first drive between the primary and secondary shafts gives a ratio of 1.8 to 1, the second speed wheels are of equal size, and the first speed wheels are verse train has the same ratio as the permanent drive only, of course, in inverse ratio, namely, 1 to 1.8.

The bevels in the axle have a ratio of 4.125 to 1.

Notice should be taken of the grease chamber and screw-down cap behind the rear end bearing of the upper gear shaft. This is considered necessary, because if the oil level drops in the box there is a small chance of the bearing running dry, though the corresponding race at the front end is fed with oil thrown up by the permanent gears. This arrangement is a refinement which is probably not essential, and might be omitted without much danger of injury to the bearing.

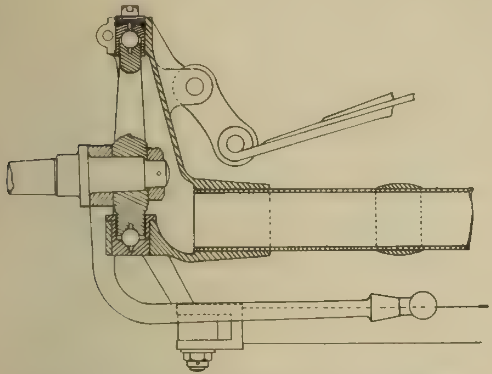
There are no other details of the unit which require explanation, except that the side of the gear box not shown is a thin steel plate, secured by studs and nuts. This enables the box to be cleaned out thoroughly, and is cheaper than an aluminium plate would be, while being equally effective.

The drain plugs in the bottom of the clutch, gear, and crank cases might, with advantage, be a little larger, and the last-named, though not shown, is situated in the same relative position as the former pair.

The next point for consideration is the universal joint, which is a unique design, said to be very satisfactory in use. We show a special perspective illustration of this, and it should be noticed that not only has this joint to perform the usual functions of a universal coupling, but it is also a telescopic slide. All the parts are steel, case-hardened and ground, and the encasement, which is completed by a leather wrapping, is fairly complete.

The torque of the drive is resisted by the universal joint, the stress being trans-

mitted by the casing of the propellor shaft (which is in one piece with the back axle) to the shaft itself, through the front end bearing. The spring pads are mounted



Front axle end and steering pivot showing built up stub axle and front spring shackle.

on the aluminium axle casing, and are secured by clamping screws, but the torque is not thus transmitted to the springs, because of the peculiar mounting of the latter on the pads. In order to relieve the springs of all twisting stresses due to the elevation of one road wheel at a time, the pads and axle are linked together by small Horkes couplings. This gives a sufficiently secure attachment and also permits universal movement. It is a refinement which is probably not necessary, but it must assist the springs to act in the freest possible manner, and should afford an interesting and worthy source of experiment for other designers.

The axle itself, together with the propellor shaft casing, is cast aluminium, of an almost pure quality. Aluminium has been used by the Rover Company for axle work for some years, and as there is no immediate prospect of its employment being discontinued, we assume that it has not given trouble. It is comparatively cheap, and it is light, but even if strong enough for the small cars made by the Rover Company we should prefer steel, and consider that steel can be made stronger, weight for weight, and just as cheaply.

No doubt the machining of the Rover axle is not an expensive performance, and the possibility of the ductility of the metal affecting the security of the housing of the bearings is overcome by the ample size of the ball races. In regard to the latter the way in which the inner races are secured is commendable. It will be noticed that the weight of the chassis is supported by the driving shafts, but we do not think there is any objection to this in so small a car. The arrangement of the brakes is open to criticism on two points, firstly, that the shoes are not as wide as they might be, considering the width of the drum; and secondly, that the socketing of the operating shafts into the aluminium of the differential case must result in wear and finally in rattle. It would be easy to bush these bearings, and not at all costly.

The internal expanding shoes are separated by means of pins set eccentrically on the end of the operating shaft instead of by means of the usual cam, and this is a good point, in that it gives a positive motion in both directions. The advantage of this is negatived in this instance by the use of wire ropes to actuate the brake shafts, but with a positive form of rod control the need for springs would be removed. The external contracting brake is controlled in a manner similar to that of the internal one, and is cleared from the drum by external springs, which draw the upper half out of engagement, too vigorous an action being guarded against by a stop, preventing the movement of the band to an extent more than sufficient than to just clear.

We have now described to a sufficient extent the main portions of the chassis, and have only to consider such parts as the carburettor, silencer, lubrication system, etc.

The carburettor is the standard Rover pattern, which is already well known. Its action is made clear by the drawing, and we need only add that the holes in the rising and falling cone and the strength of the spring are, by means of experiment, determined for each particular power of engine, once and for all, and that subsequently each car is tuned up by means of enlarging the size of the jet. Test drivers are provided with a small reamer, and each car is tested on a particular hill, the jet having to be enlarged till the car will climb the hill on the direct drive under normal road conditions.

The inlet and exhaust pipes are both copper, but the latter is connected to a steel expansion box, and subsequent connections are all steel. There are two expansion chambers joined by a short length of pipe, and the final exhaust is not unduly noisy, though, of course, it is next to impossible to completely silence a single cylinder engine, without great loss of power.

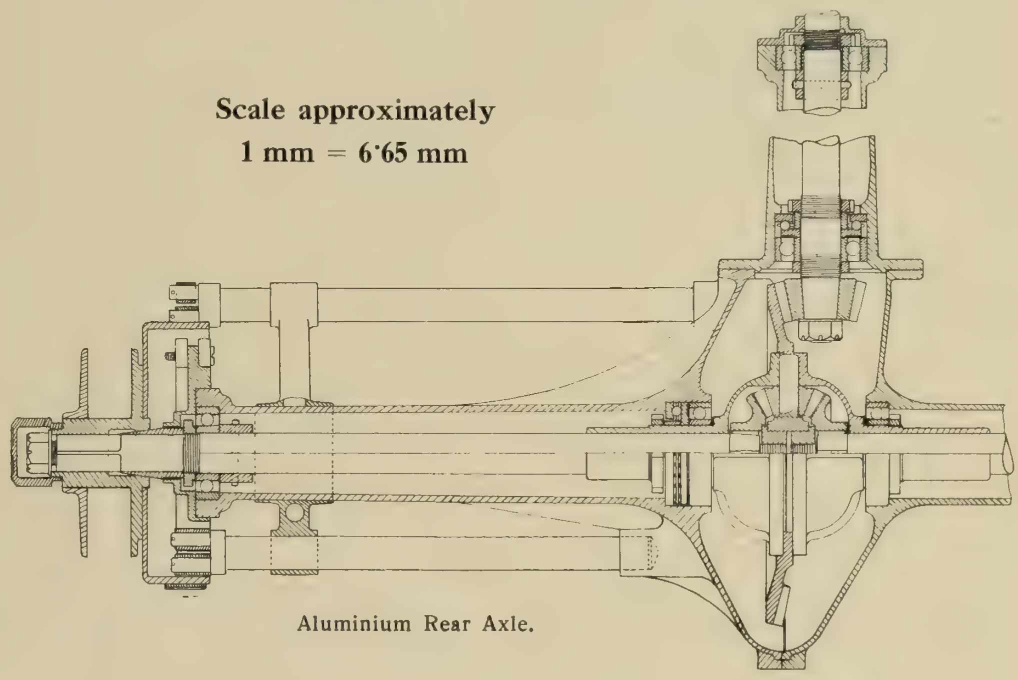
Lubrication has been reduced to the simplest possible system for the engine, by the use of the brass pot, mounted on the top of the crank case, and designed to contain just sufficient oil for a definite number of miles of running. This pot has to be filled with oil, which runs slowly into the crank chamber, taking several minutes to do so, and when empty the pot also acts as an air vent for the crank case. The clutch and gear box require to be filled up in the usual way as does the back axle. Practically all other points where ball bearings are present are provided with a grease feed, and the spring shackles and brake connections have, most of them, small cycle pattern oil cups, which are mostly arranged in accessible positions.

In general, while the design is peculiar in many parts, and is open to severe criticism on various theoretical grounds, it has proved to perform satisfactorily on the road, as is evidenced by the ability of the makers to obtain a fair price for the car for several years in succession. The principal part of the design with which we feel inclined to find fault is the considerable difficulty of getting at any part of the engine, clutch, and gear unit without dismantling the whole. Thus while it is possible to adjust the clutch from above, no part of it can be withdrawn without breaking the unit. Similarly, a new part in the gear box could not be fitted without several hours of work, and again, to take up the big end not only means opening the unit, but necessitates the complete removal of the crankshaft and detachment of one of the flywheels.

In detail matters, the car is well-thought out, and except in regard to dismantling, the parts are as accessible as they are in most designs. The lubrication of the engine puts rather too high a premium upon the memory and intelligence of the driver, and we think an automatic system would be an improvement, though the pot, which has to be filled at regular intervals, is no doubt less trouble than a gravity drip feed, and less likely to be abused.

Water pipes and waterways generally are all small, except the actual cylinder jacket, and the pump would have less work to do were they larger. Similarly, the altitude of the radiator should be increased, so as to allow natural flow to assist the pump.

However, it must not be forgotten that this car has been very successful commercially, and possesses a reputation for extreme reliability. The eliminating process of trial and error has been largely responsible for many parts of the design, and if these parts are satisfactory in use, their theoretical shortcomings are not of vital importance.



Scale approximately 1 mm = 6·65 mm

Aluminium Rear Axle.

AUTOMATIC ENGINE STARTING.

A discussion of different systems for the easy starting of internal combustion engines with which experiment has been made.

By J. Dalrymple Bell.

THE problem of providing means by which the engines of motor cars may be started from the driver's seat, is one which has received much attention. Such a device is, however, essentially a luxury—perhaps in the same sense that a paid driver is—and on this account it may be that the attention given to the solution of the problem has not been so serious as that given to other problems, the solution of which involves increased efficiency or economy. Still, in view of the fact that those manufacturers who fit seat-starters, or self-starters to their cars, find that they meet with enthusiastic approval on the part of the purchasers,

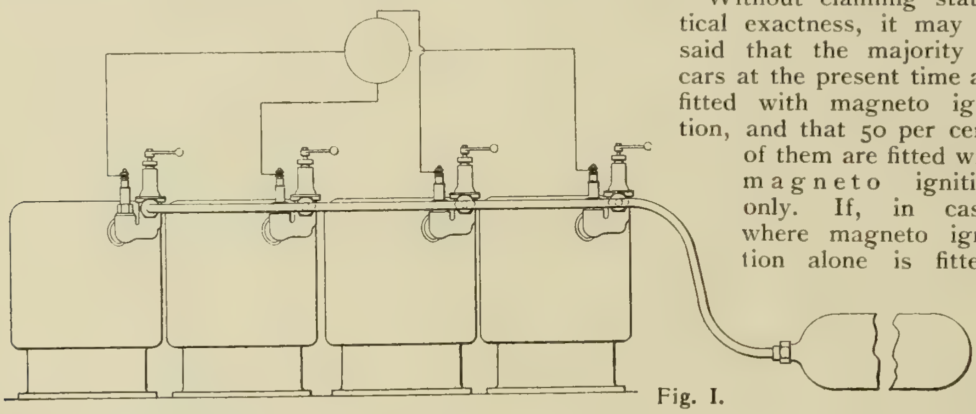


Fig. I.

there can be little doubt that for a reliable moderately-priced article of the kind there would be considerable demand.

Seat-starters may be classed as in the following table, and in this article they will be dealt with in the order as set down :—

(1) Electrical Ignition.
(2) Compressed Combustible Mixture.
(3) Compressed Air or Inert Gas stored in Reservoirs :
  (a) Admitted to the engine cylinders.
  (b) Admitted to a separate cylinder or motor.
(4) Mechanical.
(5) Electrical Power.

Electrical Ignition.

Although the least reliable, the switch starter is the most common of all self-starters. Unfortunately, it hardly merits the name of "starter" without the qualifying adjective "occasional." As, however, it is the *vis viva* of those starters which come next on the list, some little attention must be given to it. As a rule, cars fitted with accumulator ignition have a switch starter already to hand. An exception is in those cases where the system of ignition is that known as the single make and break, or non-trembler coil. In these cases a switch starter may be added, in a simple manner, by putting a trembler and switch in shunt circuit with the contact breaker. In this connection it should be pointed out that there is a very marked difference between the trembler and the single spark systems when used for the purpose of starting on the switch. The trembler spark is very much more effective, and a mixture which is too poor to ignite with a single spark will often ignite with the trembler spark, though, undoubtedly, the single spark, by reason of the exactitude with which it is timed to occur, is the better system of ignition under running conditions. Two possible explanations of the efficacy of the trembler spark for starting may be advanced. It may be that the phenomenon of the generation of ozone by an electrical discharge in air causes an uninflammable mixture to become an inflammable one or, the effect may be due to a process of convection which causes an inflammable portion of the mixture to replace an uninflammable portion which surrounded the sparking plug.

Without claiming statistical exactness, it may be said that the majority of cars at the present time are fitted with magneto ignition, and that 50 per cent. of them are fitted with magneto ignition only. If, in cases where magneto ignition alone is fitted, it is desired to fit some system whereby a spark may be generated while the engine is stationary, there are several systems open to adoption. Accumulator or battery ignition may be fitted in complete duplication to the magneto ignition, or, if it is not considered necessary to carry the duplication to such an extent, the well-known dual system, in which the distributor and sparking plugs and their connections are common to both systems, may be adopted. If it is desired to still further cut down the duplication, a system has recently been put on the market which, by making use of the armature of a high tension magneto as an induction coil, only entails the addition of a battery, switch, trembler, and some slight alterations to the magneto. The last is, of course, for the purpose of starting only. A fourth system of providing for the generation of the initial or starting spark is that fitted to the Thomson-Bennett magneto. A gear coupling is provided between the engine and magneto. This serves the double purpose of advancing or retarding the ignition, and of turning the magneto, independently of the engine, through an arc of a circle sufficient to cause a spark to take place in the cylinder whose piston is on the firing stroke.

A four-cylinder engine, as is well known, generally stops with its cranks in a horizontal position. In order to draw a mental diagram of the relative movement required between engine and magneto, divide the crank circle as the face of a clock or watch is divided. Then the crank of the piston which is on its firing stroke will be in a position corresponding to a quarter past the hour. Suppose that the magneto is advanced 24 degrees, its position when the engine stops will be 19 minutes past the hour, that is to say, 114 degrees past its firing point for that cylinder. It follows, therefore, that the advancing and retarding gear must be specially constructed to give a range, for a four-cylinder engine, of at least 114 degrees in a retarding direction, and even with such a range the possibility of generating the required spark would be by no means certain, for it would allow of no movement in a forward or advance direction in which to accelerate the armature. As there is no obstacle to prevent it, the range should, in practice, be one of at least 180 degs., that is, half a revolution. Of course, the objection to a single spark only—unless a trembler is put in the earthing circuit of the magneto, and even then the time of sparking is short—applies, but, as will be shown later, in some starting systems where a rich inflammable mixture in the cylinder is ensured, a single spark may suffice.

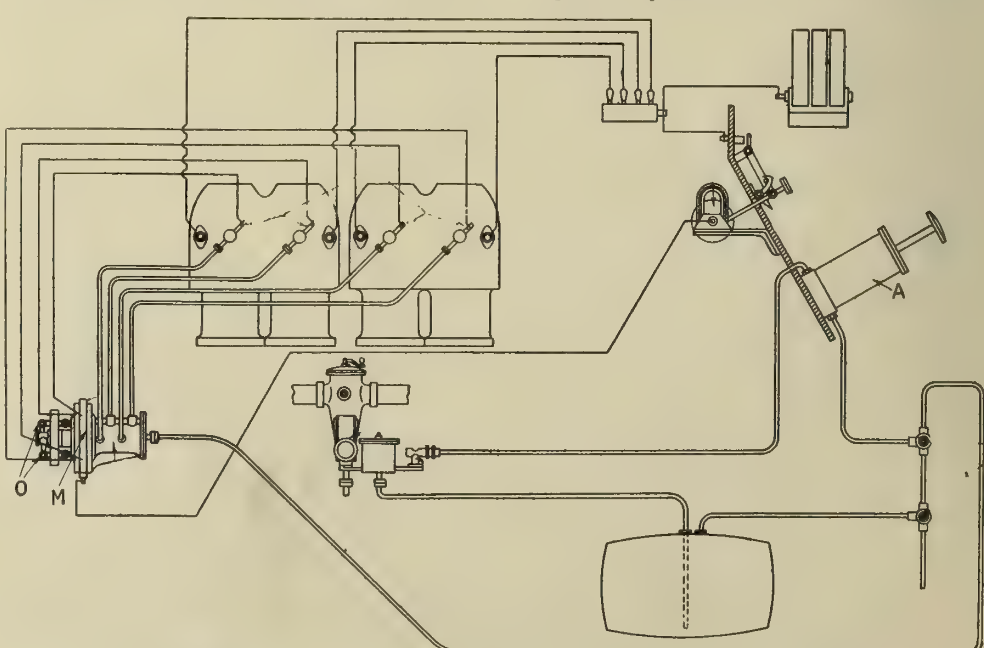


Fig. II.

Compressed Explosive Mixture.

In this system the combustible mixture may either be stored in a reservoir, or it may be supplied by a pump or pumps. If a reservoir of compressed mixture is used, the obvious source of supply to the reservoir is one or more of the engine cylinders in which the ignition is temporally cut off, the remaining cylinders driving the engine. What is not quite so obvious, however, is the necessary arrangement of valves for the purpose, and, as an example, the drawings of a patent specification are reproduced.

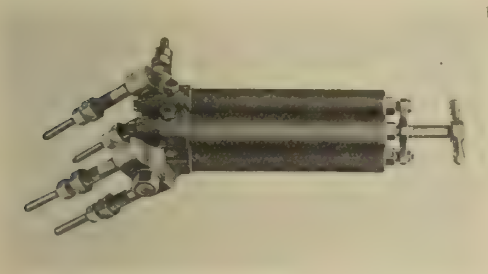


Fig. III.

In fig. I. the four cylinders of an engine are shown connected to the reservoir, through the special valve boxes, which also carry the sparking plugs. The valve boxes carry a special form of non-return valve, and these valves may either be held open, or kept closed on their seatings, or allowed to operate automatically by placing the levers in different positions. If the ordinary inlet valves of the engine are mechanically operated, a means for putting the operating mechanism out of gear must be provided. If, however, they are automatic in action, no such provision is required. When it is desired to fill the reservoir the ignition of one or two cylinders is put out of action, and the levers of these two cylinders are so positioned that the special valves act automatically as non-return valves, and the reservoir is charged with combustible mixture to a pressure nearly equal to that of the pressure of compression. When it is desired to start the engine, both the inlet and exhaust valves of the engine are closed, and all the special valves are opened, with the result that not only are all the cylinders charged with a combustible mixture, but the engine is brought into a position of balance. After this the valve operating mechanism is restored and the engine started on the spark.

Another system in which a combustible mixture is supplied to the cylinder when its piston is on the firing stroke, is illustrated in Fig. II. As shown, the mixture is supplied to the cylinder whose piston is on the firing stroke, by the foot pump A, through a distributing valve M, which puts the pump in communication with the proper cylinder. This distributing valve is driven by and timed with the engine, and non-return valves are fitted to the cylinders. When the engine is running, the plug of the distributing valve is, by a centrifugal arrangement, lifted off its seat to prevent wear. As in the previous system described, the engine is started on the spark after the combustible mixture has been caused to enter one of the cylinders. The pump may also be used for pumping up the pressure in the fuel tank, and for starting a separate ignition is provided, the electrical distributor of which is combined with the distributor valve, as shown at O.

A remaining example of supplying a combustible mixture to the engine is illustrated in Fig. III. In this device a separate pump is provided for each cylinder of the engine, but the pumps are combined in one unit, are operated together, and draw the mixture from a common carburettor. As will be understood, there is no necessity for any alteration to the engine, nor for anything in the nature of an engine-driven distributing valve. At the same time, this commendable simplification has been attained without any sacrifice of efficiency; in fact, the efficiency has been increased, and for this reason. If a combustible mixture is pumped through a distributing valve into one cylinder only, given that there is no combustible mixture in the others, the ensuing explosion would have to be sufficiently energetic to carry the engine round through half the compression stroke of the next cylinder, in the order of firing, and through half the suction stroke and the whole of the compression stroke of the next again, before the engine could possibly get another impulse, and it is doubtful if it would even get it then, as only half the suction stroke is available. With a four-barrelled pump, the cylinder on the firing stroke receives a charge, as does the next cylinder in the order of firing whose piston will be on the compression stroke, and by these two cylinders, both valves of which are closed, the engine, if it stops out of position for starting on the spark, is brought into it. The next cylinder again would have its piston on the suction stroke, the inlet valve being open, and the mixture in this case fills the cylinder and overflows into the induction pipe, ready to be drawn in as soon as the engine started. In the case of the fourth cylinder, the exhaust valve of which is open, it is scavenged by a combustible mixture before its suction stroke.

The pump is mounted on trunnions, which form plug valves, and by moving it about the axis of its trunnions the connection to the cylinders may be opened or closed. By retaining the principle of having a separate pump for each cylinder, and altering the design, a small and neat electrically-driven apparatus may be made. In starting systems of this type it has been found of advantage to have the inlet as close as possible to the sparking plug, and Fig. IV. illustrates how, by the use of special sparking plugs such connections may be made in cases where either the engine is provided with one sparking plug only, or where the sparking plug is fitted to the cap over the inlet valve, while the exhaust valve and its cap are at the opposite side of the cylinder, and therefore in a position too remote for the self-starter inlet.

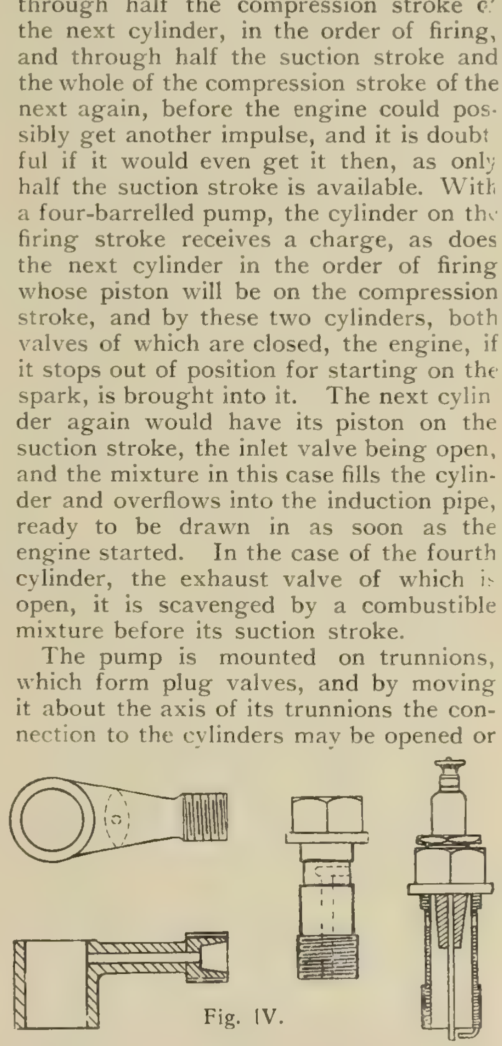


Fig. IV.

Showing special plug or sparking plug for supplying mixture.

Compressed Air or Inert Gas Stored in Reservoirs.

(a) Admitted to the Engine Cylinders.

This system has been more developed than any other, and one firm, Messrs. Newton and Bennett, fit it as a standard to all their S.C.A.T. cars. As will be seen from the accompanying illustration of parts, Fig. V., these consist of a small eccentrically-driven air pump, a reservoir, a bevel gear connection to the engine, a hand-operated valve and pressure gauge, a cylindrical reservoir, and a casting which carries the housing for a rotary distributing valve, and forms the connection to each of the cylinders. This casting is bolted down so as to form the exhaust valve covers, and at the same time the compressed air inlets, which are fitted with non-return valves.

Fig. V.

Compressed air from the small pump, the maximum pressure from which is limited by a clearance space at the end of its cylinder, is stored in the reservoir, and when it is desired to start the engine, the hand valve is opened, and the air passes through a pipe to the distributing valve. This valve is so timed with the engine that the single port which is cut in it registers with a particular channel in the casting which communicates with the cylinder whose piston is on the firing stroke. As both inlet and exhaust valves of this cylinder are necessarily shut, the pressure acts on the piston, and causes the engine to move. At the end of the stroke the air is cut off by the distributing valve from this cylinder and "switched" on to the next, the air in the first cylinder being exhausted through the ordinary exhaust outlet. In short, to start the engine, it is caused to operate as a compressed air engine. While doing so, however, its action as an internal combustion engine is not interfered with. At first sight it might appear that at best, on account of the compression space, an internal combustion engine must be a very inefficient compressed air engine, but it must be taken into account that the compression space is to some extent filled up for the reason that before the compressed air from the reservoir is admitted to a cylinder—with perhaps the exception of the first cylinder—it already contains a compressed charge.

A point of interest in connection with this particular device is that during normal running of the engine, the valve is kept off its seat, so that wear is avoided. It will be understood that owing to the

non-return valves fitted to the air pressure inlets, the suction strokes of the engine cause a partial vacuum to be formed

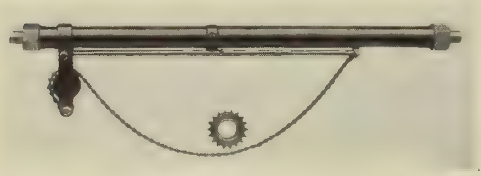


Fig. VI.

in the air passages. The stem of the valve, however, is in the atmosphere, and the atmospheric pressure acting on the stem of the valve lifts it off its seat and presses it lightly up against its cover. To allow of this, the valve and its spindle are connected together by a telescopic joint, the spindle being provided with a solid key for driving purposes. One point in favour of this system of starting is that there are incidental advantages in having a source of compressed air at hand. For instance, tyres may be inflated and pressure may be applied to the fuel tank and to the lubricating system.

A self-starter working on the same principle is fitted when required to Renault and various other cars.

Various modifications have been suggested and patented in connection with this system of starting, such as the substitution of the distributing valve by poppet valves, worked by separate cams, these being slid into or out of engagement with the valve stems, either by the pressure or by hand lever; and various means have been adopted for keeping the rotary valve off its seat to prevent wear.

As a source of pressure for starters it has often been suggested to use some of the explosion pressure from the cylinders, and there are well-known names in the patent lists in this connection, but such a system would appear to have one very grave objection, and that is, the possibility of the compressed charge entering the reservoir before it is exploded. Of course, the pressure of compression and that of explosion are very different, and a valve sufficiently loaded would prevent the compressed mixture entering the reservoir before it was fired, but a spring-loaded valve is liable to break down. The introduction of gauze, so long as it is kept cold, would prevent an explosive mixture being fired in the reservoir if it were present, still, the fact remains that such a system introduces an element of risk which is undesirable.

(b) Admitted to a separate cylinder or motor—

Under this sub-heading there are two distinct propositions. The first is to use a small compressed air engine complete as such in itself, and the second is to use a simple piston and cylinder capable of turning the engine through, from a half to a few revolutions. To employ a complete air engine of a three-cylinder type would seem to be somewhat elaborate, unless there were some compensating advantage, such as the engine acting as an inflator. But there is the difficulty that its consumption as an engine would have to be very much larger than its output as a compressor. To start the main engine the effort of the starting engine would have to be sufficiently great to overcome the load due to compression. In starting an engine by hand, the inertia of the flywheel is to a certain extent taken advantage of, and the load to be overcome is thereby reduced from a maximum to a mean, but with the starting engine the flywheel, at any rate at first, impedes. It follows, therefore, that for a short time the demand on the starting engine is considerable, and consequently the amount of compressed air used is relatively great. If then the same engine be used as a compressor, its output as such will be correspondingly great, and to store that output in a reservoir would necessitate the adoption of elaborate cooling arrangements. The obvious solution of the difficulty would be the employment of gearing, the ratio of which could be altered, but at best the system would be an elaborate one.

Fig. VI. shows a starter of the single cylinder type. The piston and cylinder take up half only of the tube, and the whole is extended, as shown, in order that it may be fitted across the car, either in front of the engine or between the clutch and gear box. The two telescopic tubes contain springs which bring the piston back when the pressure is relieved. To the engine shaft a free wheel chain sprocket is fitted. Normally the chain hangs clear of the sproket, but the first motion of the piston draws it into engagement. Further movement causes the engine to be turned through half a revolution. When, with this starter the engine is ignited by a magneto, the variation of the load due to the compression has a distinct advantage. The load at first a maximum, gradually diminishes until as

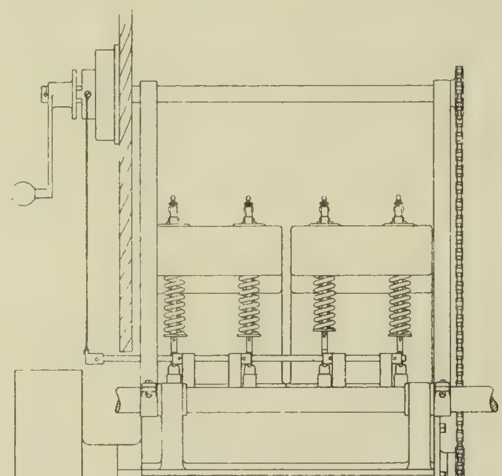


Fig. VII.

the firing point is reached, the resistance the starter has to overcome is of small value, with the result that there is an increase of speed at the particular time when such increase, as conditioned by the magneto, is most desirable. Although the particular type of this starter illustrated shows a chain and sprocket wheel, it is evident that if more than half a revolution of the engine per stroke of the starting cylinder were desired, an ordinary form of rack and pinion gear could be used.

In view of what has been previously said as to the disadvantages of storing the pressure from the explosions in the engine cylinders, and as a source of pressure is essential for these two starters just described, a pressure pump driven by the engine is the obvious alternative. The desirability of such a pump for other purposes than starting is evident, but such a pump is in the nature of an accessory provision, for which must be made by the engine manufacturer.

Mechanical.

Broadly speaking, starters which come under this heading provide for the engine being turned from the driver's seat. When an engine is turned for starting, apart from the speed required if the ignition is by magneto, a certain critical speed is conditioned by the carburetter, and it

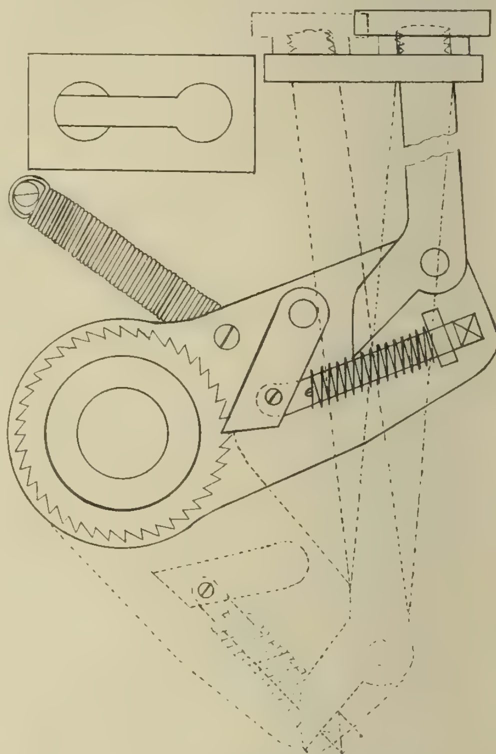


Fig. VIII.

would appear that in order to ensure this speed being attained, the best ratio for the starting gear is one to one. This being so, the difficulty arises of providing any hand-operated mechanism, which may be so placed that the driver, from his seat, will have the same purchase as he has when standing or stooping in front of the car. In the arrangement shown in Fig. VII. this difficulty is met by cancelling the load, that is to say, doing away with the compression. The inlet valves are kept open, and the engine turned with the ignition "off," until the cylinders are full of mixture. The inlet valves are then allowed to assume their normal positions, and the engine is started on the switch.

Another form of mechanical starter is shown in Fig. VIII. This consists of a ratchet arrangement, which is fitted to the shaft between the gear box and clutch. This mechanism is worked by foot, and allows of the inertia of the flywheel being taken advantage of, the engine being "swung" until it gets over the compression stroke.

Electrical.

In view of the possible adoption to a much greater extent of electric light for cars with a contingent dynamo driven by the engine, for charging the accumulators, this article would hardly be complete without some mention of electrical starters. As was pointed out in connection with a system employing a small com-

pressed air engine, the maximum load to be overcome is the highest point of the load curve when turning the engine against its compression, and in this case the maximum has to be taken as the mean. This alone would necessitate the use of a battery of accumulators of much greater out-put than that used for ignition purposes, but to this again must be added 30 per cent. or 40 per cent. of battery power to make up for loss in its conversion into mechanical power by a small electric motor. An electrical drawback is also encountered. A dynamo, unless it is of the permanent magnet type, depends upon the residual magnetism of its field magnets for the initial current generated. It is evident that a current coming from a dynamo, and one going into it, are opposite in direction; the result is that if a dynamo is used alternately as a dynamo and motor, the residual magnetism will be alternately different in sense, and, of course, where charging of accumulators is concerned, this reversal is inadmissible.

Within the limits of an article dealing with self-starters generally it is, of course, impossible to do more than briefly indicate the action of each system, but the fact that so much attention has been paid to the matter naturally raises the question why cars fitted with self-starters are so few in number. Possibly the answer may be found in considering the commercial aspect. Self-starters would, in classification, come under the heading of motor accessories, but of those systems which have been mentioned in this article, it will be observed that there is only one which does not necessitate special provision being made for it by the manufacturer of the engine, or the fitting of parts with a degree of accuracy which entails the employment of skilled labour. The exception is, that system in which a pump is provided for each of the engine cylinders. With this exception it will be seen that complete standardisation is impossible. Not only does this result in increased cost of production, but to a large extent it proscribes them from the class of motor accessories, a class to which they would naturally belong, more especially as they are a luxury, and one that every motor car owner might not require, nor would be willing to pay extra for.

TESTING SYSTEMS FOR ENGINES AND CHASSIS.

By an Erstwhile Final Tester.

NOTWITHSTANDING some of the reported assertions of a few large makers of cheap automobiles, there is no doubt that the satisfactory behaviour of either an engine or a complete vehicle cannot be guaranteed unless it is subjected to a test, more or less severe, under actual working conditions. Considering the extremely fine accuracy of first-class automobile parts and their careful inspection at all stages of production, which is customary, it can easily be argued that tests ought not to be necessary, but the fact remains that they are absolutely indispensable, and the only aspect of the matter of practical interest is the discovery of the cheapest means of arriving at the best possible results.

The days have now gone by when the purchaser of a good car might be sure that it would take a month to tune it up into its proper form, for most of the better-known cars can now be relied upon to be in good order at the completion of their road test. If improperly carried out testing may be a very serious item in the cost of production, and considering the way in which selling prices have lately fallen, we would draw makers' attention to the testing department as being one where money could usually be saved without any loss of efficiency.

One of the chief reasons of the costliness of testing, as usually carried out, is that it is essentially a time job. It is far from easy to devise a piecework testing system which will not result in the work being scamped because it is intricate in nature and, to a great extent, a matter of personal equation.

Engine testing is a straightforward bench job, and can be treated as such, providing the payment arrangements are not an incentive to shortening of the standard test run or hurrying of the power reading, but chassis tests are quite a different matter, because whether the car is satisfactory must either depend entirely upon the opinion of its driver or upon a series of timed track and hill tests, in themselves expensive to make, even if the necessary apparatus was available.

A very usual system is to test the engine separately till it gives a fixed power at a definite speed, this being, generally, the maximum, or nearly the maximum, power. Then to test the chassis for a distance of about twenty miles to make sure that nothing is absolutely wrong, and finally for one of a few picked men to drive the car a short distance, he being empowered to say whether it is up to standard or not.

In many small works this system works extremely well, because some one person holding a responsible position is able to make the final test personally. Thus in several small shops the works manager or the chief designer has to pass every car before it leaves the place, but in a large business the system is not nearly so efficient, because the final testing is in the hands of men whose only interest in the business is a wage of a few pence per hour, and while one man may be a trustworthy critic, another may be very erratic in his judgments, and it is difficult to find out whether work of this kind is being done as well as possible, providing that it is done passably. That is to say, if cars are actually giving trouble, their owners will inform the makers at once, but they will often not do so if their cars are simply not running quite as well as they ought to do.

The ideal system of testing should ensure every car of a particular model only differing from another of the same type by a very small amount. The maximum and minimum speeds should only show variations of a few per cent., and the hill-climbing power of any two cars should be equal within similar small limits. Such matters as silence and petrol consumption should likewise be observed and adjusted if necessary. It is extraordinary how many firms neglect to check consumption of fuel, and as a result it is quite usual to find two cars supposed to be exactly similar differing by twenty per cent., or even more, while it is still common for owners to discover that a small amount of adjusting will give them miles more to the gallon without perceptible loss of power.

It has too long been a platitude to say that the fuel bill is only a small item in car upkeep, and at the present time the statement does not possess even the advantage of truth, because repairs and replacements have now become an insignificant figure, and the tyre bill is not alone so great as to make the cost of petrol of no account.

Low fuel consumption is likely to become a potent argument for salesmen in the near future, and the first step towards obtaining it is to test thoroughly every chassis before delivery.

In seeking to find the best combination of efficient testing and economical testing it should be remembered that the first purpose of a test is to make sure that the car will go, and the second, that its running is up to "standard test." The first is satisfied best by a short run in the hands of a not too highly skilled driver, and can be dismissed without further consideration, but the second can only be determined by a more or less laborious process of individual operations.

It is more economical to test engines thoroughly on the bench, and to refrain from putting them in the chassis till they have given a satisfactory power reading at all speeds than to rough test the engine by making it give a maximum power within certain limits, and to leave all the fine adjustments, for slow running and quick acceleration, to be done after erection in the chassis. It is easy to determine a standard power curve for a standard type of engine, and neither difficult nor costly to adjust each engine till it conforms very closely to it, but this adjustment should only be attempted after each engine has been given several hours run under load, and to obtain the best results the engine should be thoroughly flushed with paraffin before the final settings are made. Of all the different kinds of dynamometer, probably the most satisfactory is the electric, and the least accurate the fan, but there is really a good field for the enterprising inventor in this matter, as there is a very real need for a cheap, accurate automatic recording instrument that will give a power and speed curve direct. The evolution of such an apparatus ought not to be very difficult, and in shops where large numbers of engines are tested it would save a good half of the wages bill, both because it would save a certain amount of time directly, and because it would enable a bonus system to be used, whereby it would be to the advantage of the men to take as little time as possible in getting a perfect curve from an engine which had completed its rough test or long run.

The duration of the long run necessary is a matter which can only be decided

after experiment, because it, to a considerable extent, depends upon the degree of accuracy of the finish of the parts and the care taken in erection, but it can safely be said that eight hours is the least time that can be expected to give good results, and the longer the run the better the engine is likely to be when finally tuned up. Undoubtedly a great deal of time is wasted on rough testing by the use of inconvenient accessories. It is quite common to find that the petrol and water feeds have to be "rigged" for each engine, and proper pipe couplings, and even proper ignition arrangements are far more the exception than the rule. In many test shops it is customary for two or three men to take half a day or more to get an engine set up and got ready for the rough test. Taking fifteen hours at sevenpence, which is a fair allowance, this is equivalent to 8s. 9d., and with proper test-shop fittings this can be reduced easily to, say, three hours at the same rate, or 1s. 9d., a difference of seven shillings. It may be added that these two instances are both very close approximations to the actual figures of two equally well-known firms, the larger concern making the worse showing.

The drawing office can assist the reduction of testing costs in the shop by making the engines as much self-contained as possible, so that there is no trouble in mounting engines complete with their own carburetters and ignition gear. Of course, magnetos are almost always carried on the crankcase now, but where dual ignition is used, and it is desired to test on both systems, it is advisable to arrange for the loose parts to be provided for by a rack or table adjacent to the test stand, and there are many advantages in the Daimler system, where everything except control connections is carried directly by the base chamber.

Assuming that an engine is tested carefully at all speeds, and that it is properly cleaned after that test, it should but rarely require any after attention. The cleaning may seem a small matter, but is a thing that should never be neglected, for if the engine stands some time in the erecting shop with its base chamber full of dirty oil and its valves and pistons steadily gumming up, it is not likely to give the best results when first tried on the road. This same matter of engine cleaning leads to a digression, which is that it is a job too often neglected altogether. Only a few weeks ago the writer came across a case of a new car, from the crankcase of which the owner took out several ounces of fine particles of metal, mainly aluminium, which had obviously been bolted up in the crankcase in the engine-erecting shop. It proved beyond all possible doubt that the makers had not got an efficient system for ensuring the car reaching the buyer in proper condition, and yet they make one of the best-known cars of the day.

Like the engine, the chassis requires a rough test to make sure that it has been correctly erected, and to disclose any bad faults in material, and the practice of one or two large manufacturers in employing cheap drivers for this rough test has a good deal to recommend it. Firstly, most cars are driven badly by their owners, or their owners' paid drivers, and if a car will not stand fairly rough usage when brand new it is never likely to be able to do so; secondly, as the rough test is only made to see that the car is "roughly" right, no delicate discernment is required of the driver; and, lastly, satisfactory rough testers can be obtained in most districts for 4½d. to 6d. an hour, while a good final tester is worth never less than 9d., and often as much as 1s. an hour.

The rough test is, of course, necessary to give faults a chance to develop. A new chassis should be driven a distance of at least twenty miles before any tuning up is attempted, as every part has to undergo a "settling-down" process, and loose nuts or badly-made joints will usually not show up in less than an hour's run on the road, in fact, most makers of good cars insist on a much longer rough test than the distance mentioned, and probably most first-class cars run a hundred miles at least before they are delivered.

It is fairly obvious that testers, especially the cheaper men, will be prone to waste time on the road, and there are many advantages in the possession of a private track, but as those owned by Clement-Talbot, Ltd., the Napier, and the Wolseley Co. are only small in extent, and as no other English firm use a track at all in testing it is difficult to suggest a complete system of practical track tests. It would also be very costly to build a satisfactory track, so there is nothing to be gained by discussing the matter at length.

The common way of controlling men on the road is to give them a definite run to make and to discharge them if it is found that they are constantly longer out than they should be. Such a plan has two obvious disadvantages, one that it is impossible to be sure that the car is taken to the place named, and the other that it is an encouragement to fast driving—a scurry out, a long wait, and a scurry back again. In the opinion of the writer it would be worth while, where a large number of cars have to be dealt with, to obtain a sufficient number of combined distance and maximum speed recorders which could be attached to the chassis while on test. It would not require much ingenuity to devise a simple way of fitting the drive, and it would provide considerable check on the driving, and, with fuel measurement, a direct measure of consumption. The apparatus should not be very costly in the first instance, and there is no reason why it should not be reliable.

It is a sound saying that lookers-on see most of the game, and if some of the heads of testing departments were to spend their time roaming the bye-ways of the neighbourhood they could not fail to observe that, taking all makes of car into consideration, there must be an incredible number of hours wasted every week. These hours spent by the roadside have all to be paid for, though they serve no useful purpose, and merely add to the total cost of production of the finished article.

The most difficult part of testing is certainly the final part, because with a careful engine test and careful erection it should be only rarely that any trouble arises during the rough test, and it is beyond question that it is for final critical observation at the hands of a few picked men that a private track of dimensions within practical possibility is most useful. Passage with car after car over the same surface, round the same corners, and past the same walls gives the best possible assistance to comparison, especially as regards noise. On such a track acceleration, ease of gear changing, correctness of control lever adjustment, setting of clutch, etc., can all be observed in the minimum of time and, moreover, as it is all carried out under the eyes of the staff, careless judgments are less likely to be formed than on the open road.

In the absence of a track it is an excellent thing to insist on every car making a particular short round trip, if possible in the hands of the same man. A hill test is usually easy to arrange for either on a track or a road circuit, but speed tests present greater difficulties. If the engine tuning is all that it should be there is less need for high speed testing than for any other test, and as but very few firms include an actual run against time in their series, it may be taken that no ill-effects are found to result from its omission.

In conclusion, it is hardly disputable that testing has very often been handed over to incompetent men because the works' managers regarded production of the erected chassis as their prime business, and had a feeling that testing was something quite outside their legitimate scope.

On the contrary, those responsible for the design and for the production of parts should be kept in continual close touch with the testing department, for it is by the intelligent observation of tests that real improvements can most easily be discovered, while it is absolutely certain that, given the right man in charge, many and many a testing shop could be run with double its present efficiency at but little more than a half of its present cost. There is often a tendency for the various sections of a large works to be kept apart as much as possible by their respective managers who have each their own section to look after, and too often are terribly afraid of doing anything to help each other. It is on account of this well-known fact that the "resident engineer's department," or, as it is better known, "the interference department," has become part of so many factories. This department often does much useful work, but there are several cases where it would be easy to show that it did not consider that the testing shop was its concern to anything like the same extent as the machine shops.

It ought to result in time and money saving if there was a regular system in every works by which a monthly report had to be rendered by the testing shop to the other departments, showing just in what particulars the work of each had recently been found wanting. It would assist the discovery of bad processes in manufacture and in erection, and would certainly place an additional check upon the detail inspectors in the assembling and erecting departments. Such reports to be of any real value would require to be prepared carefully from ample data, but if each engine test and each chassis test was booked up, as it ought to be, it would not be difficult to pick out the notes of interest to each department.

DESIGNING PISTONS.

Some general hints on the construction of pistons and piston rings, for students of automobile work.

By E. J. D. Buckney.

THERE is hardly a part in a petrol motor which varies more in its proportions than the piston, in spite of the fact that it works under practically the same conditions in all engines. At first sight one is struck by the fact that no two manufacturers place the gudgeon pin in the same relative position; some have it right high up inside the piston head, others a little lower down; but the favourite position seems to be as high up as possible. That this is not the best position, from the point of view of wear on the piston itself, can be readily seen.

It must be borne in mind that the piston in a petrol motor not only performs the function of a piston pure and simple, but that of the piston-rod slide also. This is, of course, done to save length, the result being that the side thrust, due to the angularity of the connecting rod, has to be borne by the piston itself, and in order to keep this thrust down to a reasonable amount, the piston has to be made longer, or in other words, is extended below the rings.

Now the piston can be regarded as being pressed against the cylinder wall by the connecting rod, and treated as a bearing supported at one point by the gudgeon pin. In order to equalize the pressure along the length of the piston, the correct position for this support is in the centre of the bearing surface; the piston rings themselves cannot be regarded as forming part of this surface, as they do not support the piston in the cylinder, but are merely for the purpose of making a gas-tight packing.

The centre of the bearing surface, that is, the centre of the gudgeon pin, can be found as follows:—

For example, with a piston of the diameter and length shown in Fig. I., which are normal proportions, take x as the required distance from the piston head, and then, referring to Fig. I., we have—

$$\cdot1875\,(x - \cdot0938\text{in.}) + \cdot1875\,(x - \cdot5156\text{in.}) + \cdot1875\,(x - \cdot9688\text{in.}) + (x - 1\cdot3125)\,\frac{x - 1\cdot3125\text{in.}}{2}$$

$$= (6 - x)\,\frac{6 - x\,\text{in.}}{2}$$

or approximately—

$$\cdot1875\,x - \cdot0177\text{in.} + \cdot1875\,x - \cdot0966\text{in.} + \cdot1875\,x - \cdot1816\text{in.} + \tfrac{1}{2}\,(x - 1\cdot3125\text{in.})^2 = \tfrac{1}{2}\,(6 - x\,\text{in.})^2$$

$$\cdot75\,x - \cdot5625\text{in.} = 6\,x - 18\text{in.}$$

$$\text{whence } x = 3\cdot313\text{in.} = 3\tfrac{5}{16}\text{in.}$$

So we have the gudgeon pin centre is 3/5-16in. from the top of the piston, or 5-16in. below the centre line. By locating the gudgeon pin here, the height, and consequently the weight, is greater than that of a similar engine with the pin close up to the piston head, *a* and *b*, Fig. II. There are two remedies for this first, to shorten the connecting rod, which is objectionable also, as it increases the side thrust; second, to offset the crankshaft slightly so that the cylinder is not directly over the crankshaft, and by this means the connecting rod can be shortened with no change in its angle on the down stroke, hence no increase in the maximum side thrust, and the engine is then shorter and more compact than the original *c*, Fig. II.

By offsetting the crankshaft the angle. and likewise the side thrust of the connecting rod, is lessened on the explosion stroke, and increased on the compression stroke. The greatest thrust occurring on the explosion stroke, it is seen that the offset is an advantage for this reason, too

The gudgeon pin should be of steel, case hardened and ground; the size depends on the necessary bearing surface required for the connecting rod end; if

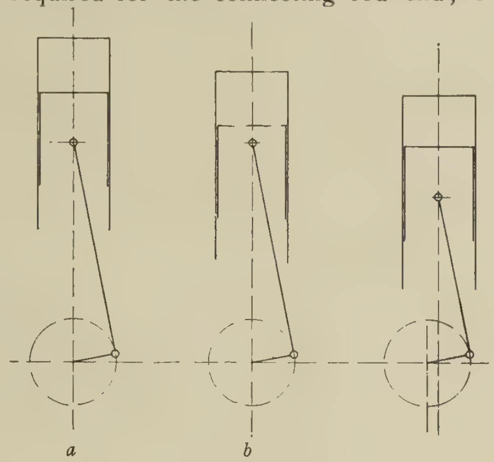


Fig. II.

this be sufficient it will be found that the pin is of more than sufficient strength to resist bending, and it can be made hollow for lightness, and drilled through for lubrication The area of surface should be such tnat the maximum thrust of the connecting rod, on the explosion stroke, does not exert a pressure of more than 1,800 lbs. per square inch on the projected area of that portion of the pin which is covered by the connecting rod bush.

The practice of fixing the gudgeon pin in the piston and leaving the connecting rod free upon it, is almost universal now, though there are one or two makers who fix it in the connecting rod end and allow it to oscillate in the piston bosses.

There are a great many ways of locking the pin in position, so that it shall not slide and groove the cylinder walls; of these, two are shown, Fig. III.: *a* consists of a set screw having a square head, which is tapped into one, or both of the piston bosses with a fine thread, the diameter of the screw being reduced where it enters a hole drilled in the gudgeon pin; this screw is prevented from rotation by a thin, shaped collar of steel, with a square hole in it, which is slipped over the screw head and secured by a split cotter pin. Another method, which needs no no explanation, is shown in *b*, Fig. III.

Either of these methods are quite satisfactory, but of the two the latter is considerably the neater.

Pistons are sometimes reduced in diameter in the central part for lubrication; this recess only reduces the wearing surface, and so is objectionable, and if an oil-way is required, it should be in the cylinder wall, low down, so that it is covered by the piston towards the bottom

As the piston head gets hotter than the walls, the clearance should be greater at the top than at the mouth; also, the greater the diameter the greater the clearance must be. This varying clearance is obtained either by turning the piston to a very slight taper, the diameter of the head being less than that of the mouth; or, by turning it in a series of steps so that the diameter is increased as each ring is passed; this latter method is found to be the better in use, though it does not give a really good bearing surface at first.

The next points for consideration are the thicknesses of the head and walls. The head of a piston of 4 inches diameter need be no more than 5-32in. thick in the centre, which may taper off to 7-32in. where it joins the walls, and no webs inside will be necessary. Behind the rings, and for the walls, $\frac{1}{8}$in. is sufficient, tapering off from below the gudgeon pin bosses to 3-32in. at the mouth; these dimensions apply to a piston in good cast iron; if made of steel they can be considerably reduced. It is possible to make the piston head slightly thinner if it is domed, or stiffening ribs are used; the total weight will still remain practically the same as that of a flat-topped piston, but ribs are objectionable as being likely to cause hard spots in the metal of the walls, and they also render it a more difficult matter to get all the pistons in a multi-cylinder engine of exactly the same weight, which is essential if the engine is to be well balanced. For this reason oil grooves on the piston, which necessitate the walls being thickened up behind them, are liable to cause trouble.

It is a good plan when setting out a piston to provide a short, fine thread, for

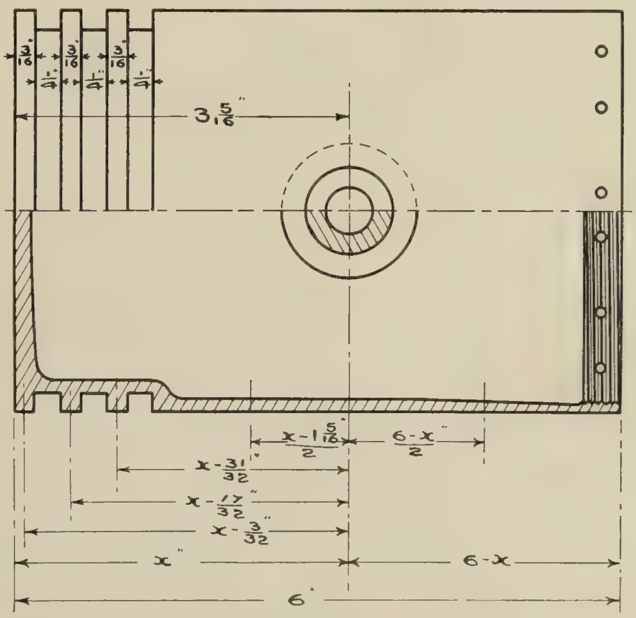


Fig I.

a distance of say, ⅜in. inside the mouth, which may be very slightly thickened up for the purpose. This thread forms a ready means of attachment to a threaded mandrel while the piston is being turned up outside and the head faced, and avoids the necessity of having a boss cast on the head for chucking; these bosses being comparatively large masses of solid metal on a thin section, are frequently the cause of defective metal in the head, especially if the boss is used as a riser from the mould when the castings are made. This thread inside the mouth should not be machined off, as it will collect the oil thrown on the inside of the piston, and if the walls are drilled through with a few small holes, the oil will be carried through to the cylinder walls, and a lip need not be cast on the piston mouth for the purpose, see Fig. I.

The next point for consideration is that of the piston-rings, the chief variations in which are in the number used, and their method of manufacture. The majority of makers use either three or four for each piston, evenly spaced near the head. Provided the rings are well fitted, and made in such a way as to take a good, even bearing all round the cylinder; it is found in practice that three are quite sufficient to make a gas-tight joint, and that a fourth only serves to increase the friction.

Pistons rings should be made of good, sound, grey cast iron, which is better than steel for the purpose, as it retains its spring longer in use, and is not so liable to score the cylinder.

A great many rings fail in that they do not bear evenly all round; this can be easily seen in an engine that has been run for a short time, by dark patches on the ring faces, due to the charge blowing past, resulting in loss of compression, reduction in power, and uneven wear in the cylinder.

Below is given a method of making piston rings, which, though rather more expensive than others, gives such good results as to warrant the extra cost.

First, have a tube of good grey iron cast (of sufficient size to machine up inside and out), with three or four lugs at one end; these lugs are to be faced and mounted on a face plate, and they are necessary, as if held in a jaw chuck the casting would not be round when taken out after turning, as the jaws would spring it. When mounted on the face

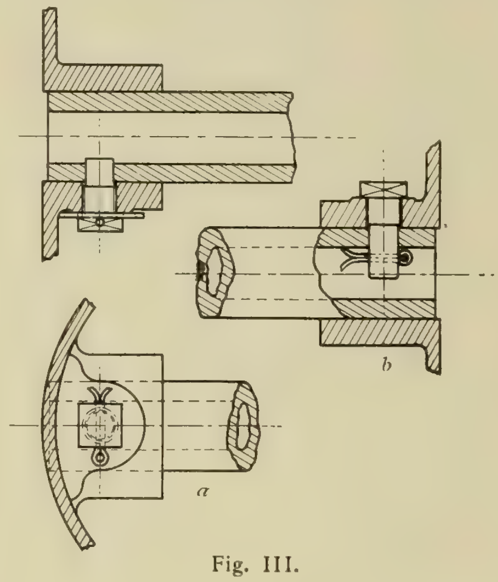


Fig. III.

plate the casting should be turned up outside, and the end faced, the inside being turned eccentrically the desired amount by shifting slightly on the face plate.

The finished end is then parted off about 1/100in. wider than the finished ring; the new end is then faced up, and the next parted off, and so on, each new end being faced up after a ring is cut off, as the parting tool does not leave a good surface.

The rings should then be mounted on the grinder in a light spring chuck, or if possible a magnetic chuck should be used, the faced side being next the chuck. The 1/100in. should then be ground off, leaving the ring the exact width of the groove in which it is to fit; the necessary clearance should then be ground off the side which was previously next the chuck.

The rings are then split at the thinnest part, and the slit closed up by tying binding wire round each ring; they are then clamped between plates, which, if made slightly smaller than the ring, readily allow for setting true; the ring, when closed with binding wire is, of course, not perfectly round, so must be carefully set in the plates, in order that the thick and thin parts come out correct; it is then turned or ground up outside to fit the cylinder bore, and is ready for use.

The various sizes of a ring of, say, 4in. outside diameter, and 5-32in. to 3-32in. thickness in its various stages, are given below, to serve as an example.

The finished ring to be 4/1-16in. diameter before closing the slit, and to allow it to close in to 4in., a cutter 5-16in. wide should be used for slitting, at an angle of 45° to the axis of the ring, in which case the ends will just clear when in the cylinder.

The outside diameter of the casting should be reduced to 4⅛in. at the first operation, which leaves 1-32in. to be taken off all round after slitting. The casting should then be bored eccentric to 4/1-16in. —(5-32in.+3-32in.) or 3/13-16in., and the rings parted off and their sides ground as above. Next split the rings with a 5-16in. cutter at their thinnest point, now ⅛in. thick, and close up with binding wire, which will reduce the outside diameter to 4/1-16in. as nearly as possible, and the final reduction to 4in. may then be made, as described already.

The rings should be closed with binding wire, because if they are forced into a tube for this purpose, they do not take their natural form, and in consequence are not so satisfactory.

If pinned in the piston grooves to prevent rotation, the pin should be put on the side of the groove next the piston head, as the lower joint is the one most required to be close.

CORRESPONDENCE.

In our introductory leader we have mentioned that we shall be glad to give space in our pages for technical argument likely to assist the advancement of the science of automobile engineering, and we wish it to be understood clearly that we shall, at all times, be glad to publish a letter, providing its contents are, in our opinion, likely to be of interest to a reasonable number of our readers.

There are many subjects in connection with automobile work concerning which there is much still unknown, and it is by the collection of numerous individual experiences that the solutions of the different problems are most likely to be found, with the minimum of time expenditure. For this reason we shall always be glad to receive particulars of any new inventions, to hear of any strange experience, or to give publicity to any new theory affecting any process of manufacture of any description of automobile.

There is doubtless little need for us to suggest particular examples of the nature we have indicated, as many of our readers will be able to name a dozen such in an equivalent number of minutes, or in even less time than this. Most men who hold a position of responsibility, in a creative section of any business, have interesting experiences to relate, or are possessed of certain theories differing from accepted ideas. We offer such men the opportunity of comparing their ideas with those of others, to their own benefit, and the benefit of the whole industry.

We shall also, at all times, be pleased to assist our readers by answering any queries that may be put to us, or to do our best to obtain an answer by publishing the questions. Similarly, we shall be glad to receive suggestions of subjects for articles. In this connection it is obviously impossible for us to deal with all the controversial matters affecting automobile construction, at the rate of more than quite a few in each issue, especially as it is our intention to deal thoroughly with all subjects with which we deal at all. The opinions of readers would, however, be of great value to us, as they would help us to decide just what subjects were the most important at the moment.

In thus inviting correspondence we would lay particular stress on the fact that we shall be especially interested in the opinions of those engaged in automobile work in other parts of the world. There is still so much to be learnt concerning the small internal combustion engine, and the other highly stressed mechanisms of small self-propelling vehicles, that no section of the world-wide industry can afford to neglect what is being done elsewhere.

British designers have ceased to be extensive copyists of Continental constructors, and are now making cars as good as, or even better than, any other people in the world, but they cannot afford, for this reason, to neglect to keep in touch with other nations. Far the best way to maintain a lead is to improve on competitors, which can only be done by studying their work. This argument, of course, applies with equal force to American or Continental makers, and the best all-round results are undoubtedly to be obtained by a free interchange of ideas the whole world over. We shall be only too glad to assist such interchange to the utmost of our power.

THE

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Correspondence on interesting subjects, which come within the scope of this journal is cordially invited. All letters must contain the name and address of the writer, but not necessarily for publication. If a reply is desired by post enquiries must be accompanied by a stamped addressed envelope.

CONTENTS.

THE CURE FOR HARD RUNNING.

ALTHOUGH there are still to be found many who will assert dogmatically that racing and other forms of competition have recently exercised but small effect upon the design of cars, in a beneficial direction, and although there are also many good arguments which can be advanced in support of this contention, it must not be forgotten that almost all forms of competitive trial have, during the last five or six years, encouraged the production of lighter and ever lighter chassis, with engines of decreasing size and increasing power. The term "high efficiency," which has been applied to such chassis by their constructors, is not inapt, but as "high" is merely relative, it decreases in force as the average improves, and there is need for some expression which will imply an internal combustion engine with a compression as high as is practicable, and developing its maximum power at a piston speed considerably in excess of a thousand feet per minute.

However, nomenclature apart, the high efficiency engine has marked a definite stage in the history of the automobile, and as the universal replacement of the vertical cylinder type by an internal combustion turbine of any kind appears to be an event of the distant future, it is well to investigate the possibilities for improvement of the existing standard design.

The two outstanding features of the majority of the engines under discussion is their silent operation, and their high horsepower as compared with their weight. The task of their designers has been none the less severe since these two qualities are, to an extent, contradictory. The higher the pressures between the parts of a machine, and the higher the speed of motion of those parts, the greater the difficulty of preventing noise, and as noise is an effect of which shock is the cause, the greater the difficulty of eliminating vibration, which is also caused by shock.

Another much-sought quality is that which is somewhat inadequately termed "smooth running," and this is less easy to obtain as speeds of revolution and violence of reactions increase. It is not too much to say that, up to the present, no engine has excelled in all the directions mentioned above. In some cases silent operation is to be found combined with light weight and high horse-power, but then usually the running is not particularly smooth, considering the full range of speed possible. In other cases smooth running goes hand in hand with high power, but then the power relative to weight (that is to dimensions) is not conspicuously great.

When examined, these facts resolve themselves into the statement that if a definite power is required it may be obtained from a large, low-speed "smooth-running" engine, or from a small, high-speed "hard-running" engine. Or likewise, from a large low pressure engine, or a small high pressure engine.

These two alternatives are not a duplication of the same meaning, because it is possible to obtain different degrees of explosion pressure by alterations of the pressure of compression of the gas previous to ignition, but in practice high pressure is usually made use of in conjunction with high speed of piston travel, and, of course, it is piston speed and not speed of revolution to which we refer above. An investigation of present-day engines leads to the conclusion that "hard-running" is always present when the piston speed is much above a thousand feet per minute, but it is not safe to assume, therefore, that the two are really inter-connected, because they may both arise from the same cause by different means. Before attacking the problem of the cure for the negative quality in question, it is necessary to have a more clear idea of the meaning of the phrase, "hard running."

Careful analysis of physical sensations, when a passenger in cars of many different types, leads to a very broad division of cars into two classes. In one class, when running fast, especially when running fast up hill, the uppermost sensation is one of being pushed by a gentle but irresistible force. In the other class there is an obviousness regarding the work being done; a passenger cannot forget that he is being propelled by an engine, and an engine which is working hard. It is not a matter of silence, for often cars of the first class make more noise than those of the second class; the feeling is the same even if the ears are firmly stopped up or deafened by the rush of the air. We have said that in the second class it is obvious that much work is being done, and this can only mean that there is some bodily feeling which announces that parts are being highly stressed, or are in rapid motion, and this must mean—can only mean—that in the second class of car, vibrations are present of a character not existing in the first class. It is, of course, assumed that comparison is only made between cars equally well made—of the same class of merit as machines —and there are many cars which belong neither to one class nor the other, but hover between the two.

Now if "hard running" is caused by vibration, and assuming for a moment that it is so caused, the next step is to ascertain whence could arise vibrations peculiar to a type of engine

not so very dissimilar to the type in which they do not occur, and this leads directly to consideration of the much-discussed conditions for balance of a vertical multiple cylinder engine. It is accepted that with single or double cylinders only, there are always vibrations perceptible as such, unless elaborate balancing apparatus is fitted, but as the four cylinder vertical engine is in mathematically perfect primary balance, its possibilities for vibration have been under-estimated. To digress for a moment, and to hark back to the time when the controversy between the exponents of four and six cylinders was at its height, it will be remembered that the conclusion arrived at amounted to this:—That while six cylinders were in better mathematical balance than four cylinders, and while six cylinders also had the advantage, in that they gave a more even torque, still practically there was but little difference in comfort between a six-cylinder and a four-cylinder car if they were of equal constructional quality.

This is still true as a generalisation, and four-cylinder cars have been so greatly improved with regard to exhaust silencing, since the time to which reference has just been made, that the difference between four cylinders and six is even less marked, in some respects, than it was then. It is, however, a fact that a six-cylinder is hardly ever "hard running" (that is, if the cranks are set so as to obtain the full advantage of the six-cylinder type), and this may be regarded as more evidence in favour of the theory that "hardness" is really vibration. Of course, a six-cylinder car is often such that even minus two cylinders it would still belong to the smooth-running class, but this does not entirely discount the value of the argument.

Turning again to the theory of balancing, while the four-cylinder vertical engine is in perfect primary balance, the six-cylinder is in perfect secondary balance also, and the four-cylinder is secondarily unbalanced. The secondary and higher orders of forces which affect the problem of engine balancing are usually disregarded as being too small to be worthy of notice, but this view must have arisen from the practice of years now gone by, since in present-day engines they may be quite considerable. It must not be forgotten that these secondary forces increase, not in proportion to the speed of crankshaft revolution, but to the square of that speed, and they are also increased by increased piston speed as distinct from crankshaft speed, that is, they increase with the stroke, if the speed of revolution remains the same. Thus it is easily possible for a small engine running at fifteen hundred revolutions per minute to produce larger unbalanced forces than are present in a larger engine running at a thousand revolutions. A little calculation will show that in the case of two engines, one giving twenty horse power at eight hundred revolutions, and the other giving the same power at sixteen hundred, the latter might be much the worse balanced of the two, even allowing for the increased weight of the pistons of the larger engine.

To give an actual example, if we take two engines both of a bore of 100 mm. (which enables us to assume that the pistons are of equal weight), if the smaller has a stroke of 110 mm. and develops, say, thirty horse power at fifteen hundred revolutions, and if the larger has a stroke of 130 mm. and develops the same power at eleven hundred revolutions, then the largest unbalanced force, when the two are giving the same power, is in the smaller engine and is equivalent to 1.6 times the corresponding force in the larger engine. Numerically, so much depends upon the weight of the piston that it is difficult to give an estimate, but such an unbalanced force might, in the higher of the two cases just quoted, be as much as three hundred pounds when at a maximum. It must be remembered that unbalanced forces can only be counteracted by a force applied to the engine externally. They are forces tending to move the whole engine, and as such they can only be absorbed by the frame in which the engine is mounted, whence in a car they result finally in a slight movement of the springs. It is a matter of common knowledge that when engines are being tested on rigid stands they frequently rupture their holding-down bolts, if they are run fast for a long period of time. This can only be caused by the unbalanced forces in the engine which, if they are large enough to break bolts, are sufficient to vibrate a chassis.

In the light of the example we have just given, it requires but a moment's thought to see that with such crankshaft speeds as are now common, *i.e.*, two thousand revolutions per minute, or even more, the secondary forces, unbalanced in a four-cylinder vertical engine, may easily assume very considerable proportions indeed. It is, therefore, reasonable to assume that the actual vibration of a high speed engine may also be quite considerable. It is, of course, only when the period of the unbalanced force harmonises with the natural period of vibration of the chassis that any large motion of the latter can take place, but high speed alternating stresses between the engine and the frame must result in a tremor, and it is almost certainly this tremor which accounts for the sensation called "hard running."

If this be assumed to be only a partial cause there are other peculiarities of the type of engine, the shortcomings of which are under discussion, and these also would tend to produce chassis vibration. It has been stated so often that the force of explosion in an internal combustion engine is completely balanced, whatever the number of cylinders, that it is apt to be forgotten that this is not strictly accurate. The force of the expanding gas drives the piston down, and there is an equal reaction tending to lift the cylinder upwards, that is to say, the force on the piston acts through the connecting rod and so causes a downward pressure on the crankshaft; the reaction on the cylinder head acts through the cylinder and the crankcase, so causing an upward pull on the crankshaft bearings. This push on the shaft and pull on the bearings are equal, and so balance each other, but the torque of the crankshaft, which is the outcome of pressure on the piston, is not so balanced, owing to the obliquity of the connecting rod. The greater the resistance to turning of the crankshaft the greater the side pressure, which means the greater the effort of the crankshaft to remain stationary while the cylinders revolve backwards upon it. At every power stroke the engine, as a whole, tends to revolve upon the crankshaft, and this tendency is counteracted by the stiffness of the frame in which the engine is secured. Of course, the fewer the number of cylinders the less steady the torque and so the greater the ability of the frame to move, for supposing it gives a little on each power stroke, if the gap between each stroke is sufficient, the cylinders will swing back before they are again pushed over by the next power stroke.

An excellent illustration of this recoil movement of the cylinders is to be found with a two-cylinder engine with both connecting rods on the same crankpin and the cylinders at right angles. This type of engine is extremely well balanced, and the vibration from unbalanced forces in it is not worth considering, but the explosions occur at irregular intervals. If one of these engines is mounted in a chassis with easy springs it will be found that at low speeds of revolution the whole engine "jumps" at each explosion, as the speed is increased and the explosions become separated by smaller and smaller intervals of time the vibration decreases almost to vanishing point. For a moment it might seem that it would follow from this argument that the higher the speed of an engine the less the effect of the explosions on the chassis, but while this is true, it is also true that the more violent the explosions the more pronounced the kick will be, and the lighter the cylinders the less their inertia, and, therefore, the greater the effect on the frame to which those cylinders are secured. This means that the higher the compression pressure the greater will be the throw-back of the cylinders, if the speed of piston movement remains constant. And, if the force of explosion is constant also, then the lighter the cylinder the greater its movement.

Although this recoil is not so serious a matter as the secondary unbalanced forces, yet it is reasonable to suppose that a small high compression engine would cause more chassis vibration at low speeds than would a large low compression engine. In the six-cylinder, of course, owing to the overlapping of the power strokes, the tendency of the cylinders to turn backwards away from the piston resolves itself into a steady pressure, which does not cause vibration, and this again is quite in accordance with ordinary practical observations.

We have already said that the immediate future of the road automobile is undoubtedly wrapped up with the development of the "high efficiency" engine, and, therefore, the problem of the cure of "hard running" is an extremely important one. There are many who, if they do not go so far as to deny its existence, at all events declare that the difference in comfort between the modern type and the older pattern of slow speed engine is entirely discounted by the greater "life" of the former. This is largely a matter of personal preference, but the advantage of a combination of both the utmost liveliness with the utmost smoothness is too obvious to admit of denial. It is well known that the balancing of the secondary forces in a reciprocating engine is a matter of considerable practical difficulty, but it is not any greater than many other problems which have been attacked and overcome by the newest branch of engineering, and at present it would appear, not merely to merit, but almost to compel the attention of the creative genius of the profession.

THE GEOMETRY OF STEERING GEAR.

By James Langmuir Napier.

THE title of this article is comprehensive but must be understood to imply merely that the author is concerned more with centre lines than with dimensions of parts. The information offered is intended to assist the designer by directing attention to principles rather than by dictating practice, and, where an opinion is offered, it is given with deference. The subject is treated quantitatively and with reasonable exactness, but long unnecessary proofs of obvious propositions have been avoided.

The first point in automobile steering gear, which seems to call for geometric treatment, is the ordinary system of arms and links connecting the steering pivots of an articulated axle. This system is outlined at *a, b, c, d, e*, Fig. I., where the steering arms produced are shown meeting at the centre of the back axle, as recommended by the originator of the idea of inclining the steering arms. The recommendation had at least the merit of accurate definition which was probably a relief to the overburdened designer of the period, but the system seems to have been adopted without examination and to have enjoyed a reputation for accuracy by no means deserved. I find, for instance, in a work published eight years ago, the statement that, when the steering arms are arranged as at *a, b, e*, Fig. I., the locus of the intersection of the steering arms produced "strictly is a curve to which the line" (indicated in Fig. I. by *f, d*), "is a tangent, and with which it may practically be confounded"; but when I find the same writer speaking airily of an error of steering of "only about 3 degrees," I begin to suspect that he has liberal notions as to "strict" accuracy.

The subject of that particular section of the steering connections now under consideration is one which does not lend itself to the construction of general synthetic formulæ; in other words, the most convenient method of designing these parts is one of trial and error. In discussing the subject I am therefore obliged to assume certain dimensions, or at least the proportions which they bear to each other. The governing features of the mechanism are *a*, the length of each steering arm; *b*, the distance between the centres of the steering pivots; and *w*, the wheelbase; and I assume as reasonably probable proportions that *a* is 1, that *b* is 5, and that *w* is 12. I shall adhere to these proportions throughout, except where it is otherwise expressly stated.

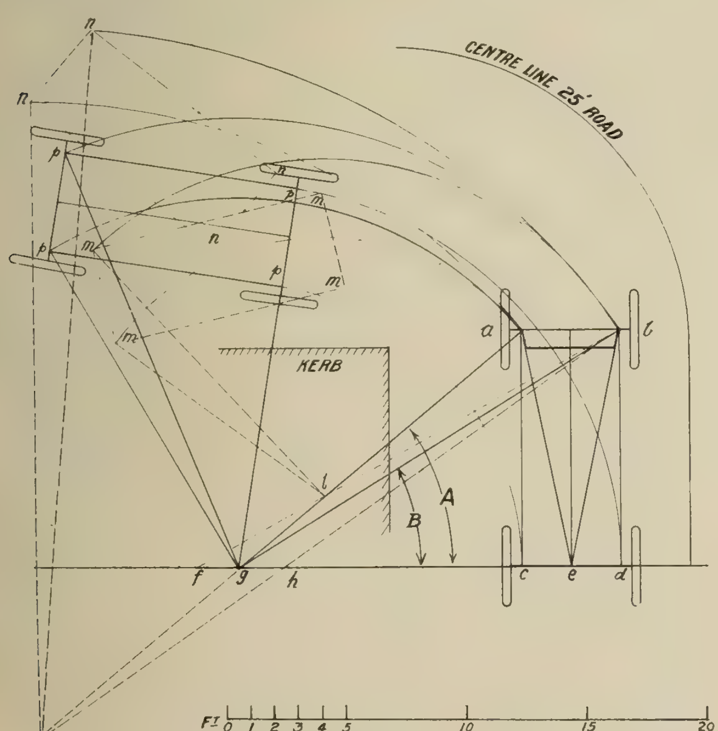


Fig. I.

In Fig. I., which is approximately drawn to scale, I assume that the unit *a* is ten inches. The length of wheelbase is therefore 10 feet, and the distance between steering pivots 50 inches. The letters *a* and *b* in Fig. I. indicate the centres of the steering pivots, and $a\,b = c\,d$. *A* is the angle of inside lock, and *B* the angle of outside lock, in which circumstances the stub axles produced meet in *g*, a point in *c d* produced, so that all four wheels roll on concentric circles and the points *a, b, c, d*, will be carried to the points *p, p, p, p*, without skidding. The angle *A* as drawn in Fig. I. measures 40°, and it is obvious from the indications of road width in the figure that it is not necessary to consider a greater angle of lock.

With a given wheelbase and distance between centres of steering pivots it will be necessary for further investigation to know the relative values of the angles *A* and *B* at various angles of lock, and I have accordingly prepared the following table, in which :—

$W = ac =$ wheelbase.

$L_1 = gc = \dfrac{w}{\tan A}$

$L_2 = gd = \dfrac{w}{\tan B}$

$L_3 = gb = \dfrac{w}{\sin B}$

The last two columns (L_2+L_3) give the width on the ground necessary for a complete turn without reversing, assuming the dimensions given for Fig. I. An allowance of one foot should be added in each case to cover the width over wheels in excess of the dimension *c d*. Where the space is bounded by vertical walls a further allowance equal to sin *B* × radius of wheel will be necessary.

A	B	L_1	L_2	L_3	L_2+L_3 inches.	L_2+L_3 feet.
5°	4°49½′	137·20	142·20	142·70	2849	237
10°	9°20′	68·06	73·06	73·99	1471	123
15°	13°33′	44·78	49·78	51·10	1009	84
20°	17°32′	32·97	37·97	39·84	778	65
25°	21°20′	25·73	30·73	32·99	637	53
30°	24°57′	20·79	25·79	28·45	542	45
35°	28°27′	17·14	22·14	25·19	473	40
40°	31°52′	14·30	19·30	22.73	420	35

In any steering gear of the type under discussion, the condition of perfect rolling can only occur at three points in the whole range of lock. One point is, of course, in the middle position of the gear, that is, at zero of lock, and the other two are symmetrical on either side of the middle position, and may, therefore, be considered as one point only. This point may be arranged as in Fig. I. to occur at 40° of lock, but as this is not necessary we shall now abandon the scale of Fig. 1, and consider *A* and *B* as any angles whatever, maintaining however the relative values attached to them by reason of the proportion between the wheelbase and the distance between steering centres.

It may happen therefore that at any angle of inside lock, *A*, the stub axles produced do not intersect in *c d* produced, but in some point outside of that line, such as *j* or *l*, and for purposes of definition, I call the steering error positive when the intersection occurs in some such point as *j*, to the rear of the back axle, and negative when the intersection occurs as at *l*. The positive form of error is more common than the negative, and generally larger. Assuming that the intersection is at *j*, the error of steering is the angle *g b h*, which I call "error α" when we use the angle *A* as a basis of calculation. Similarly, if the error were negative and the point of intersection at *l* "error α" would be the angle *fbg*. I apply the description "error β" to the angles similarly found when *B* is made the basis of calculation.

If all the wheels do not roll freely, one or more of them must skid, and we may first assume that the back wheels do not skid at all. We may also assume that all the skidding is done by one front wheel only. Let the inside front wheel roll and the outside one skid; then, whether the intersection of the stub axles produced be at *j* or *l*, the virtual centre of the rolling circles of the system is still at *g*, and the frame *a, b, c, d* will be carried to *p, p, p, p* precisely as if the rolling were perfect, but with some detriment to the outside wheel. But if the skidding

be confined to the inside wheel and the point of intersection be at j, then h will be the virtual centre of the rolling circles of the system, and similarly if the axles intersect at l the virtual centre will be at f and the car will be deflected, in either direction, to an extent proportional to the angle of error, and accordingly as the error is positive or negative. Under equal conditions of tyres and road surface the tendency is for the wheels to skid equally and divide the error between them.

All the skidding is not necessarily confined to the front wheels. Under certain conditions of loading and road surface the adhesion of the front wheels may be superior to that of the back wheels, and the latter will then tend to slip sideways in rounding a curve. I have indicated this condition in a very pronounced form in Fig. I., where *n n* and *m m* are assumed to be positions possible to the frame *a, b, c, d* when the steering error is respectively positive or negative. In the case of positive error, whatever force is necessary to cause sideslip of the driving wheels is a force tending to move them away from the centre of the turning circle j; in fact, the driving wheels are endeavouring to set the frame as nearly as possible parallel to the front wheels, a condition of affairs which is precisely what the driver, by means of the steering wheel, is endeavouring to avoid. Consequently a combination of relatively good adhesion of the front wheels with considerable positive steering error is liable to make the steering heavy. The opposite effect will be found when the steering error is negative, in which case the tendency of the drag of the driving wheels is to increase the angle of lock.

Fig. II. is a diagram of an ordinary steering system with an internal connecting rod, the steering arms being directed backwards. Fig. III. is a similar diagram of an externally connected system.

In Figs. II. and III., O and Q are the centres of the steering pivots; OC and QD are the steering arms in their central positions, each arm being inclined at the angle θ to the centre line of the figure. OE and OF are the corresponding positions of the steering arms after the inside wheel has turned through the angle of inside lock A. In both diagrams the angle FQD is made equal to the angle A, and the condition of no steering error is that the angle EOC shall be

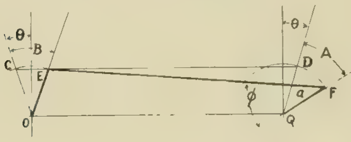


Fig. III.

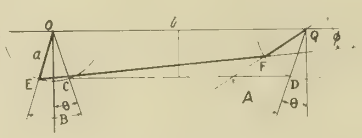


Fig. II.

equal to the angle B, as previously calculated from the assumed proportions of the system. In order, therefore, to find error α, we have to calculate the value of the angle EOC and compare it with B, when if EOC be the greater the error is positive, as previously defined.

In Fig. II the angle $EOQ = EOC - \theta + 90°$

$$\therefore \quad EOC = EOQ + \theta - 90°$$

and error $\alpha = EOC - B = EOQ + \theta - (B + 90°)$

Join OF. Then

$$EOQ = EOF + FOQ$$

Put a for EO, the length of steering arms.

Put b for OQ, the distance between steering pivots.

Then $EF = CD = b - 2a. \sin\theta$

and since angle $OQF = 90° - (A+\theta)$

$$OF = \sqrt{a^2 + b^2 - 2ab.\cos\{90° - (A+\theta)\}}$$

$$= \sqrt{a^2 + b^2 - 2ab.\sin(A+\theta)}$$

$$\cos EOF = \frac{a^2 + (OF)^2 - (EF)^2}{2a.\ OF}$$

$$= \frac{2a^2 + b^2 - 2ab\sin(A+\theta) - b^2 + 4ab.\sin\theta - 4a^2\sin^2\theta}{2a\sqrt{a^2+b^2-2ab.\sin(A+\theta)}}$$

$$= \frac{a - b.\sin(A+\theta) + 2b.\sin\theta - 2a\sin^2\theta}{\sqrt{a^2+b^2-2ab.\sin(A+\theta)}}$$

$$= \frac{a(1 - 2\sin^2\theta) + b\{2\sin\theta - \sin(A+\theta)\}}{\sqrt{a^2+b^2-2ab.\sin(A+\theta)}}$$

Similarly:

$$\cos FOQ = \frac{b^2 + (OF)^2 - a^2}{2b.\ OF}$$

$$= \frac{b - a.\sin(A+\theta)}{\sqrt{a^2+b^2-2ab.\sin(A+\theta)}}$$

The other errors are calculated in the same manner and the complete formulæ are tabulated here, in addition to those for the calculation of the value of the angle ϕ, which has some influence on the stresses of the system. Note that M and N (equal to EOF and EOQ) are not always the same angles, and that A and B must be given their calculated values.

For internal coupling rods:

$$\text{Error } \alpha = M + N + \theta - (B + 90°)$$

$$\cos M = \frac{a(1 - 2\sin^2\theta) + b\{2\sin\theta - \sin(A+\theta)\}}{\sqrt{a^2+b^2-2ab.\sin(A+\theta)}}$$

$$\cos N = \frac{b - a.\sin(A+\theta)}{\sqrt{a^2+b^2-2ab.\sin(A+\theta)}}$$

$$\text{Error } \beta = M + N + A + \theta - 90°$$

$$\cos M = \frac{a(1 - 2\sin^2\theta) + b\{2\sin\theta - \sin(\theta - B)\}}{\sqrt{a^2+b^2-2ab.\sin(\theta - B)}}$$

$$\cos N = \frac{b - a.\sin(\theta - B)}{\sqrt{a^2+b^2-2ab.\sin(\theta - B)}}$$

$$\cos\phi = \frac{b\{2\sin\theta - \sin(\theta - B)\} - 2a\sin^2\theta}{b - 2a.\sin\theta}$$

For external coupling rods:

$$\text{Error } \alpha = 90° + \theta - (M + N + B)$$

$$\cos M = \frac{a(1 - 2\sin^2\theta) + b\{\sin(A+\theta) - 2\sin\theta\}}{\sqrt{a^2+b^2+2ab.\sin(A+\theta)}}$$

$$\cos N = \frac{b + a.\sin(A+\theta)}{\sqrt{a^2+b^2+2ab.\sin(A+\theta)}}$$

$$\text{Error } \beta = A + \theta + 90° - (M + N)$$

$$\cos M = \frac{a(1 - 2\sin^2\theta) + b\{\sin(\theta - B) - 2\sin\theta\}}{\sqrt{a^2+b^2+2ab.\sin(\theta - B)}}$$

$$\cos N = \frac{b + a.\sin(\theta - B)}{\sqrt{a^2+b^2+2ab.\sin(\theta - B)}}$$

$$\cos\phi = \frac{b\{2\sin\theta - \sin(\theta - B)\} + 2a.\sin^2\theta}{b + 2a.\sin\theta.}$$

These formulæ have at first sight a somewhat formidable appearance, which is lessened on closer examination. It will also be found that after slight experience of their use one is able to estimate very closely the quantities which will give the best results. Before plotting a curve of errors throughout a whole range of values of A or B, it is advisable to make the

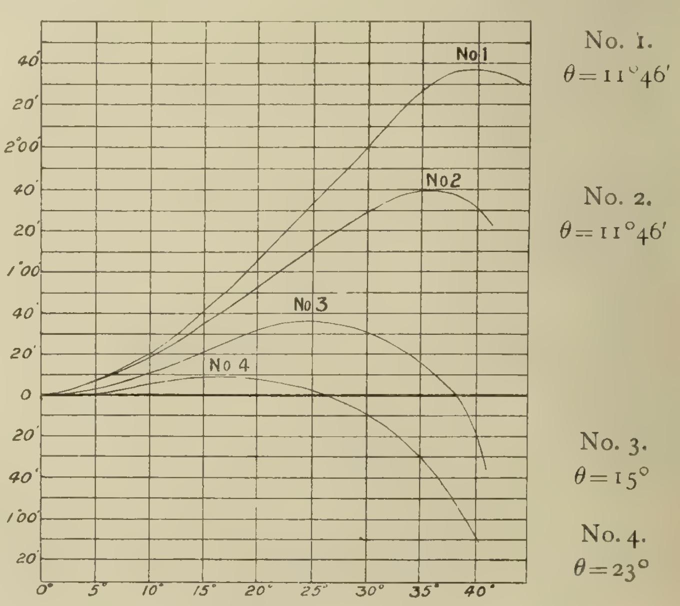


Fig. IV.

first test at $A = 30°$ or at the corresponding value of B, which, in our instance, is $24° 57'$. For ordinary purposes a four-figure

table of logarithms is sufficient, bearing in mind that it is not strictly accurate when dealing with the cosines of small angles.

In Fig. IV. I have plotted the extent of error α for two values of θ both for an internally and an externally connected system, curves Nos. 2 and 3 relating to the first and curves Nos. 1 and 4 to the second system. When θ is $11^\circ\ 46'$ the steering arms produced meet in the centre of the back axle at zero of lock.

Some general tendencies of these curves may be noted :—

(1) Error β is somewhat greater than error α.

(2) A long wheelbase reduces the steering error.

(3) For equal errors, an external system requires a larger value of θ than an internal system.

(4) In both internal and external systems the curve of error is raised by reducing the value of θ, and lowered by increasing it.

(5) In an internal system the curve of error is raised by reducing the length of the steering arms.

(6) In an external system the curve of error is lowered by reducing the length of the steering arms.

(7) In an internal system the value of ϕ is increased by reducing the length of the steering arms.

(8) In an external system the value of ϕ is diminished by reducing the length of the steering arms.

The value of ϕ should always be as great as possible, and should not be allowed to fall below 30°, especially in an internally connected system. I have calculated the value of ϕ corresponding to 40° inside lock in the internal system, of which the steering error is shown by curve No. 3, Fig. IV., and I find that when $a=1$, $\phi=29^\circ\ 24'$. When $a=0{\cdot}5$, $\phi=33^\circ\ 18'$. This indicates that the angle $\theta=15^\circ$ is, if anything, somewhat too large for safety if $a=1$, and therefore that either θ or a must be diminished. But either of these expedients would have the effect of raising the curve of error, that is, of increasing the positive error, and it is evident therefore that with an internal system of the assumed proportions we can only improve curve No. 3 by diminishing the range of lock.

The value of ϕ corresponding to 40° inside lock in the external system of which curve No. 4 shows the error, is $30^\circ\ 39'$ when $a=1$ and $26^\circ\ 22'$ when $a=0{\cdot}5$. Here the case is different from that of the internal system ; if the curve of error be unduly lowered by reducing the value of a, it can be raised by reducing the value of θ, and both of these operations facilitate the design of an external system. Generally, the external system will be found to involve smaller mean and maximum errors, and should be adopted whenever the design of the steering pivots, stub axles and wheels admit of it.

In the days of short wheelbases steering errors were unavoidably greater. To take a case in point, if we assume $a=1$, $b=5$, and wheelbase $=9.311$, and if the arms of an internally connected system produced meet at the centre of the back axle the value of θ would be 15°. Error α would be positive throughout, and at about 35° of inside lock would amount to $1^\circ\ 46'$, which cannot be its maximum value, and this we have seen cannot be conveniently reduced without reducing the range of lock.

Some such considerations probably prompted the modifications of the ordinary connections, which took forms approaching those indicated in Figs. V. and VI. The error of steering always begins by being positive and may remain so throughout the whole range of lock. In the case of an internally con-

Fig. VI.

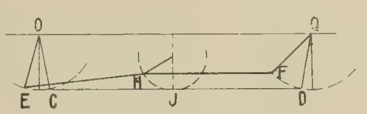


Fig. V.

nected system positive error means that the connecting rod CD, which was originally the correct length, virtually becomes too long. Hence some arrangement such as is shown in Fig. V. for shortening the direct distance between its extremities. It is clear that by such means we can reduce the error of such a curve as No. 2, Fig. IV. For if we make the angle FQD, Fig. V., say, 40°, and EOC equal to the corresponding calculated value of B, and from E and F, with radius equal to CJ or JD describe circles intersecting in H, and if we support the point H as indicated in the figure, then we have arranged mechanically that there shall be no error of steering at 0° and at 40° of lock. Intermediate errors will still exist, and it will be found that the improvement to be effected on such a curve as No. 3, Fig. IV., by any construction similar to that of Fig. V. is too small to compensate for the undesirable complication.

Fig. VI. shows an analogous arrangement applied to an externally connected system. In such a system the connecting rod, originally of correct length, becomes virtually too short, and the construction requires accordingly that the process of Fig. V. should be reversed and the rod originally bent should be gradually straightened. This construction, attributed to Lavenir, is remarkable chiefly for the extravagant claim made for it that it gives perfect rolling at all angles of lock. It may be that I have not hit on the precise proportion of parts necessary to achieve such a result, but I have not been able to find valid support for the statement. In my view the system has precisely the faults and the virtues of that shown in Fig. V., with the disadvantage of greater liability to accidental injury.

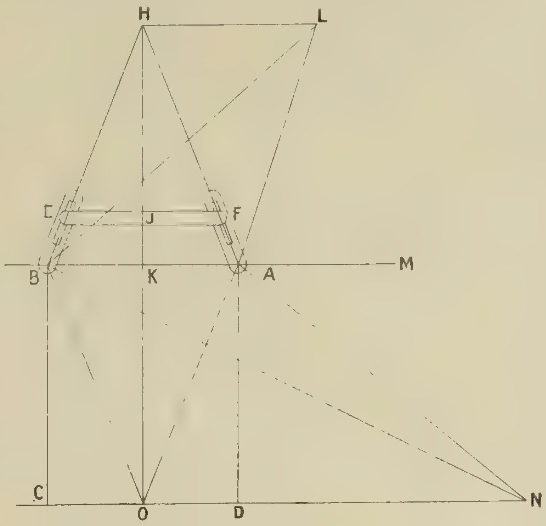


Fig. VII.

Fig. VII. shows diagrammatically the construction of Bourlet's sliding system of steering connections. A and B are the centres of the steering pivots, and BC is the wheelbase. Make $HK=KO$ and join HB and HA. BE and AF are slotted steering arms engaging with the ends of the rod EF, which is constrained by guides to remain parallel to AB at a constant distance JK. Then if the arms BE and AF directed towards H take up new positions, such as BL, AL, it can be shown that HL is parallel to AB since EF is parallel to HB, and JK is constant. Join OB and OA. Then the angle $HAM=OAM$, and if HAM be turned about the point A so that HA coincides with the direction of AM, AM will coincide with AO. Similarly if the angle HBM be turned about the point B so that BH coincides with BM, BM will coincide with BO. Then if the angle LAN be made equal to HAM and CN is parallel to BM, it can easily be shown by the method of *reductio ad absurdum* that the angle LBN is equal to the angle HBM ; that is to say, that when the steering arms lie in the directions BC and OL, so that HK is equal to KO and HL is parallel to AB, the stub axles produced will intersect in the production of the back axle, and HAL and HBL will be the angles of inside and outside lock respectively. The system is geometrically exact, but involves certain difficulties of mechanical design, notably the avoidance of excessive bending stresses in the sliding bar EF, and in the adequate protection of the sliding surfaces. The difficulties do not strike me as insuperable, and the advantage of exact steering is considerable.

In Fig. VIII. let the circle EDC represent the area of contact between a tyre and the road. The wheel is constrained to move in the direction OA, but can only roll freely in the direction OB, deviating from OA by the angle θ. If the wheel were permitted to follow the path OB the point C, being the foremost point of contact with the road surface, would remain stationary until it became the hindmost point of another area of contact, CBG. It is, however, constrained to move to H through the distance CH at right angles to OB. H bisects the chord of a circle touching the circle EDC and having its centre, K, in OA. HF is obviously parallel to OB, and it can easily be shown that CH is equal to $CD=CE$ sin θ ; and this is true for every point on the circumference of the wheel in the same plane as EC. If

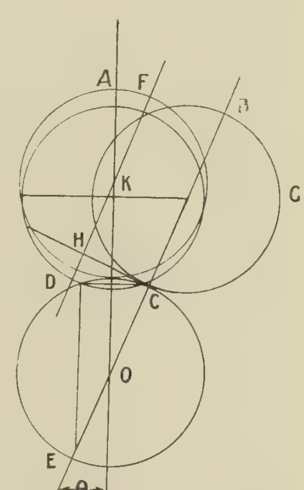


Fig. VIII.

the circumferential velocity of the wheel be V feet per minute, p the air pressure in the tyre, f the co-efficient of friction between the tyre and the road, and b the width of a narrow strip of tyre containing EC, then the work done per minute in moving

the wheel, as far as that narrow strip is concerned, is

$$V \times f \times p \times b \times CE \sin \theta$$

But $p \times b \times CE$ is the load on that portion of the strip of tyre which is in contact with the road; and since the reasoning applied to the point C may be applied to any other point on the same half of the boundary of the area of contact, of whatever form that area may be, it follows that the whole work done in moving the wheel is (in foot lbs. per minute),

$$V \times \text{Load on wheel in lbs} \times f \times \sin \theta$$

and the horse-power wasted in destroying the tyre and the road surface is

$$\frac{V \times \text{Load} \times f \times \sin \theta}{33{,}000}$$

To realise the numerical value of this quantity in a moderate case, we may assume that f is 0·5; that θ is 0°30′, of which the sine is ·0087; that the load on one wheel is 500 lbs., and that V is 1760, corresponding to 20 miles an hour.

Then

$$\text{HP.} = \frac{1{,}760 \times 500 \times 0{\cdot}5 \times {\cdot}0087}{33{,}000} = 0{\cdot}116$$

Considered as a waste of power occurring only when rounding a curve, this does not seem serious, but, when it is remembered that the wasted power is exercised in the destruction of rubber, it appears desirable to make the steering error as small as possible. The worst offenders, of course, are those wheels which are permanently out of parallel.

This consideration applies not only to wheels which are out of parallel in plan, but also to those which are out of parallel in elevation, that is, to splayed wheels. Since splayed steering wheels are already common, and we are threatened with splayed driving wheels, it may be worth while to consider their effect on tyres.

A splayed wheel is virtually the base of a cone whose apex is at the point where the axle produced would meet the road surface, as indicated in Fig. IX. The tendency of such a cone is to roll in a circle, of which the apex is the centre and the side of the cone the radius; and the effect of restraining the wheel from so rolling is the same as if the wheel were compelled to make a complete revolution about a vertical axis, passing through the centre of the area of contact between the wheel and the road surface, once during such length of straightforward progression as is equal to the circumference of the circle in which the wheel tends to roll. Unlike the case of the wheels in vertical planes, but badly aligned as seen in plan, the friction caused by splaying the wheels is affected both by the form and extent of the supporting surface. A long narrow area of contact causes more friction than one more nearly circular, and a large area of contact more than a small one.

I shall assume, principally for the sake of having something easy to integrate, that the area of contact is circular. Then, at any radius r inches, p being the air pressure per square inch, the load on an annular strip of infinitesimal width dr is $p\ (2\pi r.dr.)$ and in one revolution of the wheel about a vertical axis that load is moved through $\frac{2\pi r.}{12}$ feet, and the work done in respect of the annular strip is

$$\frac{p \times f \times (2\pi r)^2 . dr}{12}$$

where f is the coefficient of friction. If r be the radius of the area of contact, the work done in respect of the whole area for each revolution about a vertical axis is

$$p \times f \times \frac{4\pi^2 r^3}{36} \text{ft. lbs.}$$

Then if V be the velocity of the car in feet per minute, and R be the length in feet of the side of the imaginary cone, of which the wheel forms the base, the horse-power wasted in friction by each wheel is

$$\frac{\frac{V}{2\pi R} \times p \times f \times \frac{4\pi^2 r^3}{36}}{33{,}000}$$

$$= \frac{V \times p \times f \times 2\pi r^3}{R \times 33{,}000 \times 36}$$

and it is obvious that this quantity varies inversely as the square root of p.

For a numerical example we may assume that the tyre is inflated to a pressure of 70lbs. per square inch, and that the load is 500lbs. on each wheel. The radius of a circular area of contact will then be approximately 1.5 inches. For an average speed of 25 miles per hour V will be 2,200, and f as formerly, we may assume at 0·5. If we assume that each wheel is inclined to the vertical at an angle of 1 in 20, and that the diameter of the wheel is 3 feet R will be 30, and the horse-power wasted on each wheel will be

$$\frac{2{,}200 \times 70 \times 0{\cdot}5 \times 6{\cdot}2832 \times 3{\cdot}375}{30 \times 33{,}000 \times 36} = {\cdot}0458 \text{ H.P.}$$

or for two wheels 0·0916 H.P.

To look at the question from another point of view, the torque on each steering pivot is

$$p \times f \times \frac{2\pi r^3}{36} \text{ in ft. lbs.}$$

which in the example we have taken amounts to

$$\frac{70 \times 0{\cdot}5 \times 6{\cdot}2832 \times 3{\cdot}375}{36} = 20{\cdot}62 \text{ lbs. at 1 foot radius.}$$

which is the torque opposed by the steering arms. In opposing the forward progress of the car the torque acts with a virtual radius of 30 feet, and the force opposing the motion of the car is therefore (since it acts at each wheel)

$$\frac{2 \times 20{\cdot}62}{30} = 1{\cdot}378 \text{ lbs.}$$

and the horse-power wasted is therefore, as before:—

$$\frac{1{\cdot}378 \times 2{,}200}{33{,}000} = {\cdot}092 \text{ H.P.}$$

These figures have a trifling appearance. It is perhaps more commensurate with the importance of the subject to state that in the example under consideration the work devoted to the destruction of the tyres during the whole time the car is running amounts to 3,036 foot pounds per minute.

When, as in Fig. IX, the steering pivots are vertical and (as shown there by dotted lines) the wheels are vertical, it is quite evident that if one or both steering whels meet an obstruction there is a tendency to deflect the axle or axles backwards and to impose some stress on the steering arms and connecting rod. It appears to have occurred to some inventor that, since the smaller and more common class of obstructions lie on the ground, if the lowest point of a wheel were prevented

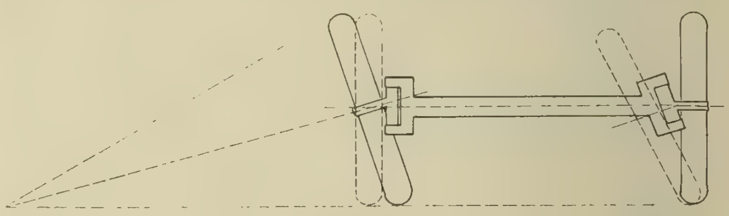


Fig. IX. Fig. X.

from moving with reference to the steering pivot, no torque would be imposed on the pivot in surmounting an obstacle. Hence the construction shown by full lines in Fig. IX., and its later development shown by full lines in Fig. X. This latter avoids the grinding of the tyres which I have shown to be inherent in the splayed wheel, but at the expense of avoiding all practical usefulness.

In both figures the centre line of the steering pivot passes through the centre of the area of contact between the wheel and the road, and it is quite clear that no amount of horizontal force applied at that centre will tend to turn the steering pivot about its axis. Both constructions involve the fallacy of assuming that the resistance to the rolling of a wheel is a horizontal force acting in the plane of the road surface. This is not the case, and although the resistance has a horizontal component, that component is not in the plane of the steering pivot.

In Fig. XI. a vertical wheel moving in the direction from A to E is shown at the instant of leaving the ground to pass over an obstacle. The wheel is then in equilibrium under the action of three forces: AO, a horizontal force driving the car; BO, a vertical force due to gravity; and CO, the reaction at the point of contact with the obstacle. CO is equal and opposite to FO, the resultant of AO and BO, and can be resolved into two components DO and EO, of which DO is equal and opposite to BO; and EO, the resistance to rolling, is equal and opposite to AO, and is equal to W. tan θ where W is the load BO on the wheel.

Since the force CO is applied in the plane of the wheel, so is EO, and it is directed towards the centre of the wheel. Therefore, if in Fig. XIII, A be the intersection of the axle and the

steering pivot, and the pivot be inclined to the vertical at the angle ϕ, the couple tending to turn the steering pivot is

$$AB.\cos\phi.W.\tan\theta$$

Since ϕ is always a small angle this expression does not differ greatly from $AB.W.\tan\theta$, which is the couple tending to turn the steering pivot when both wheel and pivot are vertical. It appears therefore that the construction indicated in Fig. X is not justified. It has the further disadvantage that the wheel

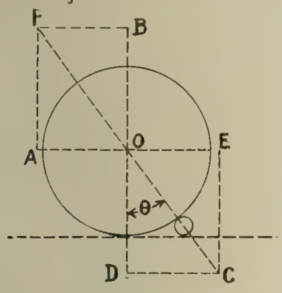


Fig. XI.

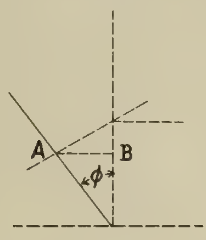


Fig. XII.

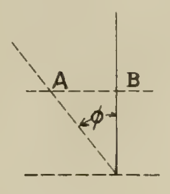


Fig. XIII.

is in unstable equilibrium when vertical. Any disturbance would tend to cause it to fall into the position shown by dotted lines in Fig. X., in which position it is statically in stable equilibrium.

The splayed wheel, Fig. XII., is more efficient in diminishing the effect of road shocks. Here the couple tending to turn the steering pivot is approximately

$$AB\,(1-\cos\theta)\,W.\tan\theta$$

which, even when θ is so much as 30°, is not much more than one-eighth of the couple tending to turn the steering pivot when both wheel and pivot are vertical. Strictly, the couple in the case of the splayed wheel is :—

$$AB\,(1-\cos M)\,W.\tan M$$

$$\text{where } 1-\cos M=\frac{1-\cos\theta}{\cos\phi}$$

Either formula is, however, of purely academic interest, since the steering connections must always be designed, not in accordance with the theoretical mean, but with the practical maximum stresses. For this reason I am not greatly impressed by the economy of any system of mounting steering wheels designed solely to prevent straightforward road shocks from exerting turning moment on the steering pivots. From this point of view the Renouf system, in which the wheel is vertical and the steering pivot is in the plane of the centre of the wheel, is geometrically perfect, but even in such a case it is necessary to provide for such a contingency as a glancing collision with a kerbstone.

The Renouf system unfortunately offers to the designer a temptation to reduce dimensions for the sake of neatness. It is also somewhat difficult to make the wheel detachable and at the same time conveniently applicable to a driving axle. If these difficulties were overcome, as they might be at the cost of some slight sacrifice to appearance, the system would lend itself peculiarly to the benefits to be derived from raking the steering pivot in the plane of the wheel. The effect of this arrangement, properly designed, would be to put the wheels in static stable equilibrium when parallel to the frame and to cause them to incline inwards when rounding a curve, thus reducing rolling resistance and the bending stresses on the spokes.

The geometry of the subject is not difficult, but undeniably complicated, and I do not propose to add it to this already lengthy article. The complete quantitative investigation for any particular case, while absolutely essential, if the best result is to be attained, will be found a somewhat laborious undertaking, but first approximations may be found conveniently by graphic construction.

I should add that it is not safe to assume that the raked steering pivot will automatically supply a "righting lever," to borrow a nautical term, which will ensure that the steering wheels will always tend towards their middle position in the event of failure of any portion of the gear. The torque which I have described, when dealing with the effect of splayed wheels, opposes the "righting lever," and is as likely as not to be the more powerful. Breakdown of steering connections should always be insured against by the provision of ample strength and bearing surfaces.

THE LUBRICATION OF AUTOMOBILE ENGINES.

A consideration of the general principles of lubrication and the systems employed by various makers, with comparison of the advantages and disadvantages of the different arrangements.

DURING the present year there have been two interesting discussions on the subject of the lubrication of automobile engines. The first was on the occasion of the paper read by Mr. G. H. Baillie before the R.A.C., and the second at the reading of Mr. R. K. Morcom's paper before the Institution of Automobile Engineers. The former paper was a general sketch of the systems in use and ended with a conclusion in favour of trough, or other constant-level forms of splash lubrication. The latter dealt only with forced lubrication, for which the author declared his strong preference.

It is possible that Mr. Morcom is unduly prejudiced in favour of forced lubrication owing to his long experience with it on his own engines, but it is also possible that Mr. Baille was influenced in coming to his decision by having seen some very badly designed forced systems. One thing, however, to which neither of them gave much attention, was the question of the material and dimensions of the crankshaft journals.

It is well known that there may be a great difference in the behaviour of similar bearings under similar conditions if the materials used are different. There is a surprising difference between one steel and another, and one white metal and another in this respect, and it is usually strongly advisable to experiment with different bearing metals till the one is found which is best suited to the particular steel of the crankshaft.

Size, however, has a still greater effect on durability, and though it would be thought that every man capable of making an engine that can run would take due notice of this fact, we make no apology for pointing out that it is better to make an engine of such a length that the crankshaft is properly supported and the big ends get sufficient bearing surface to prevent the oil being squeezed out on each explosion stroke, than to save three or four inches on the length of the bonnet.

It may be said, without fear of contradiction, that if there were no gudgeon pin half the troubles of automobile engine lubrication would be over, because the majority of these troubles arise from the contradictory necessities of carrying sufficient oil to the small end, while keeping too much from the cylinder walls. In fact, one of the most important reasons why new systems of lubrication are now being tried in place of the plain splash system, which has been in common use so long, is that, with the high speed of the present day engines, if sufficient oil is supplied to splash-feed all the bearings, the cylinders are smothered and it is impossible to prevent excess of oil reaching the combustion chamber, with the usual evil effects.

An engine ought to be capable of driving a car for at least fifteen thousand miles before it requires cleaning internally, and it ought not to require bearing renewal under double this distance, always providing that it receives fair treatment. Yet there are very many engines now being turned out by makers with good reputations which require the combustion chambers to be cleaned out every five thousand miles, and also need bearing attention within the same period of time. It should give no offence to emphasise the fact that the progress of the automobile is all the more extraordinary by reason of the lack of engineering knowledge on the part of many early designers who, none the less, were able to turn out good cars, and it is obvious from some of the lubrication systems of the present day that this lack of knowledge was not confined entirely to years gone by. First of all it appears to be forgotten very often that, if a bearing is loaded above a certain critical point oil cannot be retained in it, and the shaft and brasses must come into contact with each other. This is compensated for to some extent, if the load is alternating, as it is in the case we are considering, but if the pressure on, say, a big end, is so great at the moment of each explosion that the oil is

squeezed out for an instant, during that instant part of the crank pin and bush must be rubbed away. If, on the other hand, the oil can be retained all the time, then the life of the bearing will be satisfactory. The pressure at which oil is fed to a journal has an effect upon the critical pressure between shaft and bush, of course, and it is because forced oil raises the critical pressure that its use

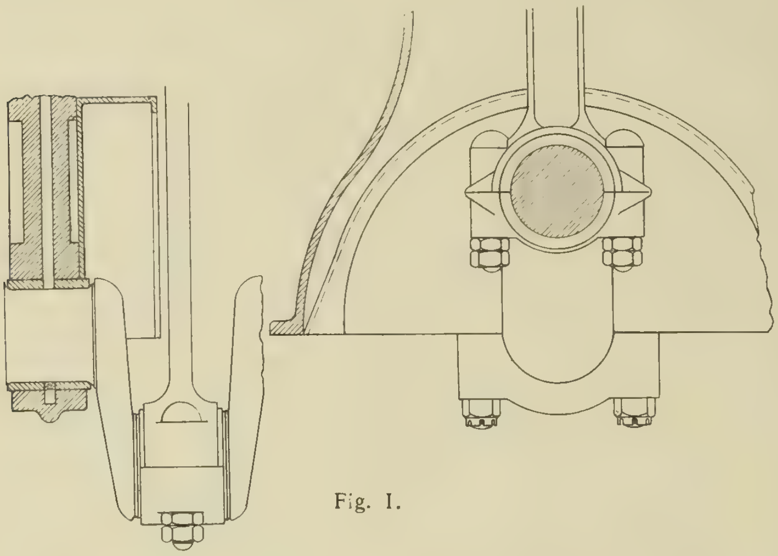


Fig. I.

increases the life of a bearing so greatly.

Still, even with the highest pressure of oil, the bigger the bearing the longer its life, and this is particularly true of the big end, because even with forced lubrication it is difficult to make sure that the pressure shall always be the same, for if one bearing is slack it at once causes a drop of pressure at the others, unless each is fed separately.

Speaking broadly, forced lubrication is only necessary when bearings have to sustain heavy loads or continued high speeds, and an engine which will run well with splash lubrication for ordinary touring work will often seize at some part if it is run at high speed continuously. This is quite in accordance with laboratory observations made many years ago, and has been proved again and again on Brooklands racing track, where engines are run at full power for long periods. Habituées of Brooklands will remember certain cars which were capable of bursts of etremely high speed, but could do nothing in long races, and in almost every case the reason was that the lubrication was faulty. Some of those cars which have lately carried off many prizes owe their success, not only to the high power which their engines can develop, but to their ability to maintain that power for many minutes consecutively, which means that the bearings do not run hot when run at very high speed with heavy loads.

The success of a forced system, however, depends entirely upon its detail construction. It is not enough to force quantities of oil to the crankshaft bearings, because, if this is done, the cylinders will be over-oiled by spray if the engine speed is high. The cylinders need very little oil, comparatively speaking, and if the crankshaft bearings are streamed with high pressure lubricant it is almost impossible to prevent quantities of it reaching the combustion chamber. Thus for racing purposes it is necessary to force as much oil as possible to the main bearings and big ends, consistently with a fairly clean combustion space.

The devices which have been tried to keep oil (when fed under high pressure) away from the cylinders are extremely numerous and varied in character. A favourite one which has been used for many years is to fit a baffle plate of thin sheet metal at the bottom of the cylinder, the plate being slotted to permit the connecting rod to move. This arrangement is effective with splash lubrication, whether of the plain or controlled type, but its effect is not always sufficient with forced oil. The makers of the Vauxhall car get over the difficulty by drilling a number of small holes in their pistons just below the rings, and it is claimed that excess of oil on the cylinder walls is squeezed between the walls and the lower part of the piston, so that when it reaches the holes it is at once forced through them and falls down inside the piston. Other makers have found that a scraper ring (an additional piston ring at the bottom of the piston) is effective, and others use both scraper rings and baffle plates.

It has been practically proved that the over-lubrication of cylinders arises mainly from the oil which gets on the crank webs, and is thence whirled off in a sheet of spray which impinges upon the cylinder walls with considerable force. It would appear to be better to tackle this trouble at its root by keeping the oil away from the cylinders than by arranging mechanism to remove it as fast as it is put on them, and it seems curious that (so far as we are aware) no one except the De Dion Company has yet tried the device shown in Fig. I. It has been found that quite a small thrower ring will prevent the exudation of oil from a bearing along the shaft, and, as the oil on a crank web must leave it at the outer end, if the latter is not finished square, as in the case of the De Dion, but is brought to a fairly sharp edge, as shown in Fig. I., it ought to be easily possible to catch all the oil in the hood surrounding it. Then if the small end was fed by pump through a pipe outside the connecting rod, or through a hollow connecting rod, the exudation therefrom should still be sufficient for the cylinder, and not too much.

One thing that is apt to be forgotten is that it is not sufficient to make a lubrication system which is good from the point of view of bearing life, unless it is also economical of oil, and the advisability for obtaining a low consumption is another potent reason for keeping oil away from the combustion space. In the discussion which followed Mr. Baillie's paper at the R.A.C. some figures were given relating to the performance of a fleet of 'buses in which trough was substituted for forced lubrication, and it was announced that the consumption per engine had been altered from about a hundred miles to the gallon of oil to as much as eight times the distance. There are only two things to account for this, one that the old system was sending quantities of oil to be burnt up in the cylinder, and the other that owing to inadequate stuffing boxes, much oil was being pumped through the end bearings of the crankshaft to fall upon the road. It is not fair to either system to take a bad example of one and a good example of the other for purposes of comparison.

It is also unwise to have too much regard for the present practice of well-known firms, because everyone is now experimenting, and it is most probable that some years will pass before there are many completely satisfactory systems. Not long since a census of the leading manufacturers would have shown that an enormous majority were using plain splash, but it would obviously have been wrong to have argued that this proved plain splash to be the best, and similarly, because many makers are getting good results from a particular trough system, or a particular forced system, it does not follow that either is the most satisfactory system obtainable. In the case of the Wolseley, the Daimler, the Sheffield Simplex, the Belsize, and a number of other touring cars, the trough system is at present working well, and certainly it is more economical and less troublesome than plain splash, but it has yet to be proved that forced lubrication properly applied will not result in better bearing durability, with equal economy and lack of trouble-giving. Amongst the principal exponents of entirely forced systems are the De Dion, the Lanchester, the Sunbeam, and the Thornycroft. So it is obvious that opinion amongst leading manufacturers is fairly evenly divided, when it is possible to pick out such examples as we have just cited.

Similar examples are to be found in the heavy vehicle industry. For instance, Commercial Cars, Ltd., obtain excellent results from their trough system, and the Albion vans do equally well with their forced oil.

It is noticeable that the protagonists of the trough system all state that the disadvantage of forced systems is the length and small diameter of the oil passages, but this is not a really sound argument, because there is no need for the oil-ways to be either small or tortuous, and whilst oil under pressure requires a firmly fixed obstacle to bar its progress effectually; oil which has to reach a bearing through a small hole, even though it be a very short one, by gravity alone, can be kept out by quite a soft piece of dirt. On the other hand there is much truth in the statement if it is applied only to the forced systems now in use and not to forced lubrication in general, because most makers use pipes of a needlessly small size, and are apt to make any holes in the crankshaft of very small bore, to avoid weakening the shafts. As regards cost of manufacture the trough system would be in a position of slight advantage, but the difference is not great, as the pump and its fittings are needed in either case.

Seeing that this is so it would surely be well to make use of the power of the pump to feed the main bearings at least,

even if it was considered inadvisable to drill the crankshaft for big end leads. Ideally there ought to be a pump to each bearing to prevent one or more bearings being starved because one of the others was slack, and so was passing more than its fair share of oil, but in practice it is not found necessary, and it is only essential to see that the leads to each bearing from the main supply pipe are of approximately the same length and size. Systems in which oil is fed to each bearing in turn by a rotary distributer would not seem to be so good at those in which pressure is maintained in every bearing all the time the engine is running. This leads naturally to consideration of the point at which oil is best fed to a bearing, and here we find that convenience has been studied much more than anything else by practically all car manufacturers. Probably the fact that oil ought always

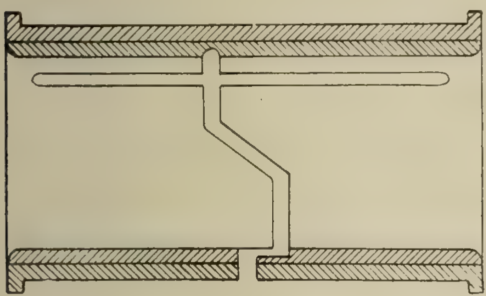


Fig. II.

to be fed along the line of minimum pressure has not forced itself upon the notice of internal combustion engine makers, because the position of minimum pressure is always shifting with the varying stresses to which the crankshaft is subjected. Mr. Morcom suggested that the best possible way to feed oil to a bearing with alternating loads of this character was to keep it under pressure in a central groove of the form shown in Fig. II., when the oil must always be present at the position of lowest surface pressure, wherever it may be. The staggering of the groove was suggested as an easy way of avoiding the formation of a ridge on the crankshaft, which might be produced by a simple circular groove. It is thus easy to supply oil under pressure to main crankshaft journals, and easy to prevent it causing mischief when it escapes, by one of the methods mentioned previously; the chief difficulty is to secure an adequate supply to the big ends and gudgeon pins, without an undue supply to the main bearings. It might be well to emphasize the fact that the stresses, and therefore the bearing pressures on these parts are very high, for in the case of a four-inch engine the total explosion pressure is about 3,800 lbs., and this has often to be distributed over a bearing with an effective area of less than three square inches. Though this pressure lasts only for an instant, it is probably during that instant that eighty per cent. of the wear takes place.

With the usual form of crankpin drilling the orifice is nearly at the top the instant before explosion takes place, and as the piston is travelling upwards fast, and is stopped suddenly by the crank at the top of the stroke, then the top of the bearing is probably the place of lowest surface pressure for the instant before explosion, even when the resistance of the charge to compression is considered. This means that the oil spurts into the bearing in the best possible place at the best possible moment. The lubrication of the gudgeon pin is less easy on account of its reciprocating movement, but in order to give it a charge of oil just before the explosion takes place, it would be advisable to take a lead from the big end bush at a point well in front of the top of the cap, so that while the piston was on the upstroke the port should register with the hole in the crank pin just before the top of the stroke is reached, to ensure the charge of oil reaching the pin at the moment when it can get between the surfaces where it is needed an instant later.

It would be an undoubted advantage if it were possible to feed the big ends by other means than by a drilled shaft, since the steels most used for automobile crankshaft work are liable to fracture if any sharp corners are left in the drilling. It is easy to countersink the outside ends of the holes, but it is not easy to prevent sharp angles inside, save in cases like the Lanchester, where the stroke is so short that a straight hole can be taken diagonally from the centre of a main bearing to that of an adjacent big end.

There do not appear to be any insurmountable difficulties to prevent the use of separate direct feeds to each big end, and it is a matter quite worth the attention of designers of racing engines.

Another point on which there appears to be considerable difference of opinion is as to the size and nature of the pump. That is to say, the amount of pressure desirable, as this varies from nothing up to as much as 40 lbs. per square inch. This ought always to depend upon the maximum bearing pressure, and upon the speed of rubbing of the bearings. Mr. Morcom gave the equation that if P equals the peripheries of every bearing all added together, then the pump ought to deliver 8P cubic inches of oil when running at its maximum speed. It is obvious that any such equation must be empirical, but Mr. Morcom ought not to be far wrong, considering his wide experience. This equation has the advantage of being very simple to use, and it may be interesting to examine its effect in a particular case. Suppose a crankshaft of a four-cylinder engine with five main bearings, and with a diameter of 1½ins., the big ends being the same size.

Then $P = 9 \times 1.5 \times \pi = 42.5$ and $8P = 340$ very nearly. Now if a plunger pump is used, and the maximum speed of the engine is 1,360 r.p.m., the pump will have to deliver a quarter of a cubic inch of oil per revolution, or half a cubic inch per camshaft revolution. Thus, if the pump is driven from the half speed shaft it would deliver the right quantity, according to Mr. Morcom, if it had a diameter of from ¾ to 13-16in., and a stroke of 1in., and so it will be seen the result is more or less in accordance with current practice.

Mr. Morcom also gives the best dimensions for the bore of the delivery pipe as $\frac{P}{480}$, and in the example used above this comes out equal to just over 11-16in., which is much larger than is usual, being more than double the size of the majority in actual use. Probably the idea is that the bore of the delivery pipe should be about equal to that of the cylinder of the pump, if a plunger type is used, but while it would be well for the total sectional area of a number of small delivery pipes to equal that of the cylinder, probably .75 of the cylinder area would for all practical purposes be enough for a single pipe.

To sum up the arguments on both sides it appears:—

That plain splash lubrication is quite satisfactory for engines which have large bearings (*i.e.*, a low bearing pressure), and do not run at high speeds, except, perhaps, for quite short periods of time.

That the trough system of splash lubrication is more economical of oil than plain splash, and less troublesome to the driver of the car, as it requires much less attention.

That where bearings are loaded heavily, or are required to run at high speeds for long periods, forced lubrication will increase the life of the bearing surfaces and remove the risk of seizure.

That forced lubrication has the disadvantage that it is liable to smother the cylinder walls and so waste oil, and gives rise to other troubles, but that this can be got over by fairly simple means.

That in point of cost the forced system is the most expensive to install, the trough system being a little cheaper, and the plain splash much cheaper than either.

And finally that, generally speaking, whatever the nature of the bearings and their rubbing speed, the higher the pressure under which oil is fed to them, the greater is their durability certain to be.

This summary is certainly in favour of forcing oil to all bearings, but it also leads one to suppose that the trough system has its uses, and for many purposes is very little inferior to the former.

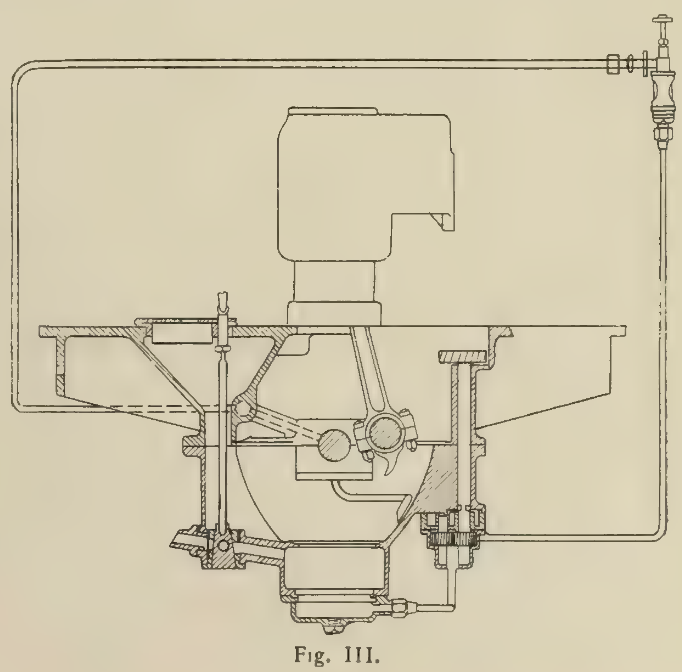


Fig. III.

In either case a great deal depends upon the details of the system, and it is therefore instructive to examine some examples of both types as at present constructed.

Taking trough systems first, the details of the Wolseley were given in our last issue, and the Daimler system is described on another page in this number. Both are purely splash systems, as the main crankshaft journals are lubricated by oil splashed from the troughs. The pumps used are of moderate capacity and supply the troughs only, a point of difference being that the Wolseley pump supplies all the troughs at once, and the Daimler each in succession, through a distributer. There is probably but little to choose between the two on the score of efficiency.

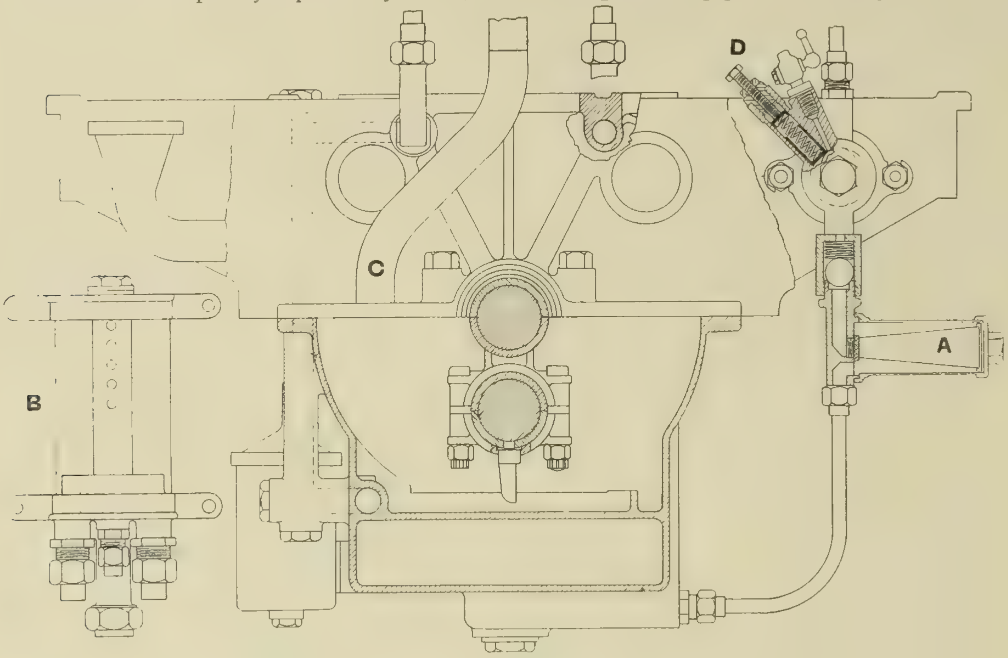


Fig. IV.
A. Gauze filter. B. Dashboard reservoir. C. Overflow pipe from B.
D. Adjusting screw for pump.

The Sheffield-Simplex arrangement, Fig. III., is a combination of the two systems, having trough-supplied oil for the big ends and gudgeons, with forced oil to the main journals. The same gear pump is used to supply the troughs and the main bearings, and, with an output of 1.2 gallons per 1,000 revolutions, it is said to supply the main bearings with oil at a pressure of about 4 lbs. per square inch. When experimenting with this system great difficulty was found in correcting over-lubrication of the cylinders, and finally both baffle plates and scraper rings were adopted to prevent this. It appears that the trouble was attributed to the splash from the troughs, but it was probably greatly aggravated by the oil escaping from the main bearings. This appears still more likely when it is remembered that neither the Wolseley nor the Daimler have any special protection for the cylinders, though the Wolseley has special ducts to carry any excess of oil fed to the main journals away from the crank webs. The Sheffield-Simplex dippers are peculiar in that they are scoop-shaped, while the Daimler dippers are blunt-ended, and the Wolseley brought to quite a fine point. The oil in the Sheffield-Simplex system is supplied to the main bearings only, and the overflow from them is taken to the troughs. Before reaching the distributer box, from which the main bearing leads pass, the oil is led to the tell-tale of the piston type on the dashboard, so the latter shows if the pump fails. This means a rather long passage for every drop of oil, and for this reason we think it might be an improvement to make the tell-tale a bye-pass connection, and keep the main bearing leads as short as possible, though with a pump capable of giving a fair pressure, choking of the pipes is unlikely.

The Belsize system, Fig. IV., is also trough, combined with separate gravity feeds to the main bearings. The pump is of the rotary type, and has the advantage that it is mounted high, and can be taken down without letting the oil out of the crankcase. It feeds to a tank on the dashboard, whence the oil runs to the main bearings, and to the troughs. Overflow from the dashboard container is returned to the sump, so there is never any pressure in the system, and we believe the makers have not experienced any great over-lubrication troubles. This strengthens the opinion that it is the oil forced from the main bearings which has caused the trouble in perfecting the Sheffield Simplex system. The amount of oil passed by the Belsize pump, Fig. V., can be adjusted by the set screw and lock-nut, and the non-return valve beneath the pump is to obviate the necessity of filling up the system from above the pump whenever the engine is started, there being always some oil in the container.

It is noticeable that none of the four engines which we have chosen as typical of the trough systems are of the ultra high efficiency type. That is to say, they are not engines from which every effort has been made to obtain the last ounce of power. This remark, however, applies with equal force to the examples of forced systems mentioned, of which the first, and one of the oldest, is the Lanchester. This is illustrated in Fig. V., and is a completely forced system. The crankshaft is supplied with oil under a pressure of about 40 lbs. per square inch, according to the makers. The pump feeds to a pipe running under the crankshaft having leads to each main-journal, whence diagonal passages take oil to the big ends, and from them it passes up inside the hollow cylindrical connecting rods to the small ends. The gudgeon pins are also hollow, and oil can pass through them to the cylinder walls. Here, again, there is no special device to prevent over-lubrication of the cylinders, notwithstanding the high pressure, and this is perhaps partly due to the small crank radius, which would very considerably reduce the amount flung off from the webs. Also the pistons are rather long, and none of the oil-ways are large, so the initial high pressure cannot force an excess of oil to any bearing.

In the De Dion arrangement, Fig. VI., all bearings, except the gudgeons, are supplied with oil under a nominal pressure of about 10 lbs. per square inch. Oil from the gear pump (which is of patented design) is taken to a pipe of large diameter in the bottom of the crankcase, and led thence to each main bearing. From the main bearings it passes to the big ends through drilled holes in the crankshaft. The gudgeon pins and cylinders depend upon oil spray from the other bearings. To prevent over-lubrication of the cylinders, hoods of sheet metal used to be placed over every crank web, somewhat as shown in Fig. I., but at present only baffle plates of the customary pattern are employed. The degree of oil pressure having been reduced to slightly less than it was formerly.

The Thornycroft system is interesting, because the makers of Thornycroft cars have had considerable experience in the construction of large petrol engines for marine work. For some of these big slow-running engines they have used oil at as high a pressure as 100 lbs. per square inch, without ill results to the combustion chamber, and without any protective devices for the cylinder walls. The car system is conspicuous by the neat leads to the main bearings, and these we illustrate in Fig. VII. The gudgeon pins obtain their oil by splash from the cranks, and the pressure at which the oil is fed to the main bearings is adjustable within limits by a setting arrangement, which adjusts a relief valve passing oil back to the sump. This feature is shared by the Sunbeam, which is a similar system, with three main bearing leads feeding the big ends through drilled passages in the shaft. The Sunbeam, like the Belsize, has one very good point, and that is that the pump is high up, and can be taken off without losing any oil. To lift the pump thus above the height which it can suck when dry, means that priming will be necessary if the non-return valve beneath it should stick up, but it seems bad practice to do as the majority of makers and arrange, not only the pump, but the filter also, so that it cannot

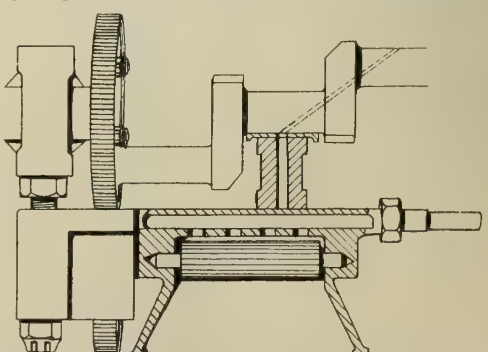


Fig. V.

be inspected without emptying the system. It is conceivable that it might be necessary to handle either of these important parts of the lubricating plant by the roadside, and to lose a gallon of oil in doing so would be annoying, and perhaps impracticable in the absence of sufficient spare lubricant or of a vessel to contain the old oil.

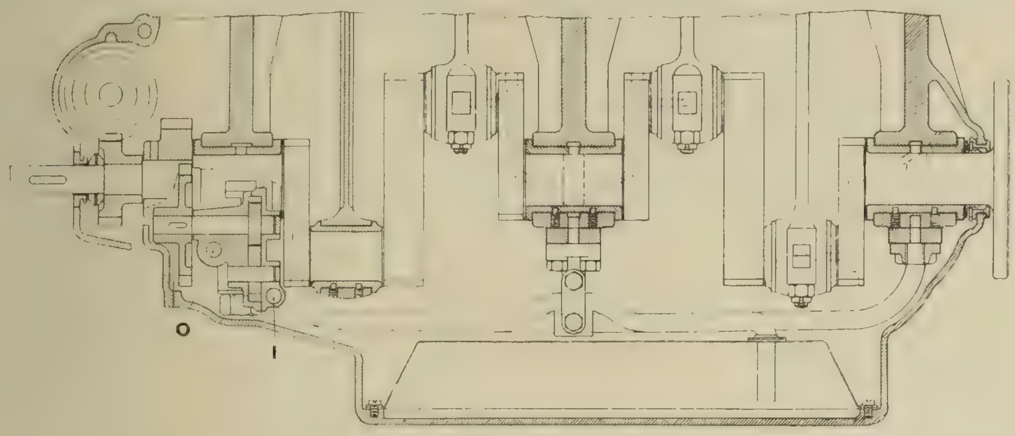


Fig. VI.

The systems that have been described have been chosen simply as typical ones, and other systems exhibit numerous small variations on them. Almost all have failed to take full advantage of the convenience of automatic lubrication, whether by trough or forced feed. Thus it is usually necessary to have the car on a level floor for filling up, so that the overflow (in itself a dirty device) shall show when the oil is at the correct level. The float used on one or two cars is much cleaner, and less susceptible to level variations when fitted about midway in the length of the crankcase. Probably its most convenient form is that of an inverted metal cup, with a wire stem, which, when it is lifted and dropped suddenly, rests for a moment on the surface of the oil, and shows the level before sinking again.

There appears to be much difference of opinion as to the best arrangement of filtering. Some makers have simply a gauze tray covering the whole of the bottom half of the crankcase, as in the Daimler, so that all oil falling from the bearings must pass through it, and sometimes this tray is the only filter. In other cases there is an additional gauze over the pump intake, as in the Lanchester. Others, again, have no tray, but only the gauze on the pump, and yet others rely upon a submerged gauze between the sump and the bottom of the crankcase (in the De Dion there is a central well in the case, surrounded by sump, and the walls of the well are gauze). The essential points which should should be remembered in designing a filter are that it should be of such dimensions that it will not restrict the supply to the pump, even when dirty, and that it shall be accessible for cleaning. In this last respect the large gauze tray is usually very bad indeed. The cylindrical cover for the pump intake is usually easy to detach, but at the expense of the oil in the crankcase, and it should not be difficult to arrange both filter and pump in a chamber cast integral with the bottom of the crankcase, about midway on one side of it, so that by undoing quite a few nuts the whole pump and filter could be withdrawn upwards, leaving the oil undisturbed. Such an arrangement would be a great convenience, and would be recognised as such by anyone. It is easy to argue that the pump only requires to be touched at long intervals, but there is no certain knowledge as to the time when one of those intervals will come to an end, and anything that improves accessibility without adding to cost of manufacture, or interfering with anything else, is well worth having. These concluding remarks apply with equal force to all systems in which a pump is used, and there is but little doubt that it is only a question of time before there is a pump on every car, either to force or regulate the supply of lubricant.

Discussions of different systems would be incomplete without mention of the Albion, which supplies small charges of oil to each bearing in turn. It does not maintain oil pressure in bearings, but it gives repeated small doses of fresh oil from the tank. The idea is that circulated oil is full of minute impurities, and that only fresh oil is to be depended upon to be free from abrasive particles. (It is true that if oil be left in circulation too long it loses in viscosity and becomes a much less efficient lubricant, and if a circulation system was never washed out it might become very foul indeed.) It has yet to be proved that this system is applicable to very high-speed engines, or very heavily-loaded bearings. The Albion pump consists of a plunger working in a cylinder mounted upon a plate, which revolves very slowly as the engine

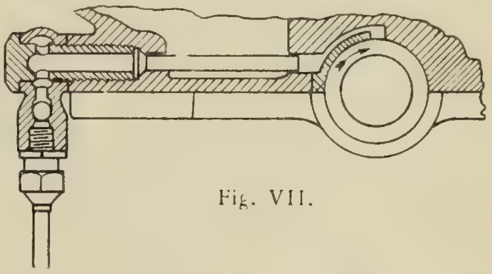


Fig. VII.

runs. The top of the box in which this apparatus is contained has a series of stops, arranged in a circle to correspond with the path of the top of the plunger. These stops are cam-shaped, and are adjustable. As the plunger passes under each it is depressed, and sends a charge of oil down a lead in the bottom of the case corresponding in position with the stop. Separate leads feed to every bearing, including the pistons, and there is no splash, as used oil simply falls to the bottom of the crankcase, and is drained away.

ALUMINIUM AND ITS ALLOYS.

By Alex. E. Tucker.

THE metal aluminium is one of the numerous examples of which metallurgists may be justly proud. The older generation of scientists will call to mind that in their youth aluminium was a curiosity. Under the patronage of the late Emperor of France, which, it will be remembered, was extended to Bessemer, the manufacture of aluminium was commercialised. One of the first manufactures in the metal was that of filigree jewellery and lace work (a severe practical test of ductility), which looked well when new, and sold extensively, but from the delicacy of the articles, and probably also from the method of manufacture, i.e., the old sodium process, the metal in this form had a short life, as it tarnished and became rotten. To the same causes may be attributed the failure of the repeated attempts to use the pure metal for domestic and other purposes. The porosity of the metal and small amounts of certain impurities suffice to bring about tarnishing and disintegration, and it follows that such conditions would accentuate the failure of the metal much more when in the form of thin wire and sheet than in the substantial products in aluminium with which we are now familiar.

When attempts were made at using pure aluminium for heavy work, which the properties of the metal seemed to warrant, it was soon found that additional difficulties arose. Its liability to oxidation, when highly heated, led to trouble; it was cast too hot, and we hear of large propeller blades being made which turned out to be a mass of blow holes, probably due to the formation, during melting and casting, of alumina, the oxide of aluminium. We read also of boats having their hulls constructed of sheet aluminium, which rotted away under the action of sea water, and rods of the pure metal showed very poor mechanical strength.

Formerly aluminium was made entirely by the sodium process, and the history of aluminium was largely the history of the sodium industry. The purity of the former depended greatly on the purity of the materials used, as well as on the character of the linings of the furnaces in which the reduction was carried on. It was extremely difficult to avoid the introduction of considerable percentages of iron and silicon, and some amount of sodium was often left in the finished metal—an impurity well calcu-

lated to make it rotten. Silicon increases the tendency to oxidation and corrosion, while sodium has an extraordinary effect in this respect. Silicon should not be present in pure aluminium in greater amounts than 0.4 per cent. or 0.5 per cent., and it has been found that as little as 0.006 per cent. of sodium brings about rapid oxidation. It may be said that the difficulty in preventing the presence of these bodies in aluminium made by the sodium processes greatly retarded the applications of the metal.

In the aluminium now produced by the electric furnace, the strengths, on account of its great purity, are very low. The table above shows the average figures obtained on a large number of tests.

The above figures well indicate the remarkable effect of working the metal, and illustrate indirectly the inherent porosity of the pure unworked material.

Notwithstanding these low results on unrolled or otherwise unworked metal, the success which has attended the introduction of alloys of aluminium in various manufactures is little short of marvellous. It is computed that the output of aluminium for 1909 was 40,000 tons, and only three, or perhaps four, metals are now cheaper, i.e., iron, lead, and zinc, the fourth being the very variable priced metal copper. The drop in value of aluminium during the last few years is phenomenal, thus not such a great number of years ago the price was £1,000 per ton; it is now about £75, and the purity of the metal is far higher than ever it was. These results have been brought about by the enormous development in the application of electric furnaces, and the production of current by water power, whereby large and very costly plants, designed for other processes, have in consequence been laid aside. Simultaneous with this drop in the cost of production there has been an equally remarkable development in the technology of its application, and it is the use of the alloys of aluminium, and not the pure metal, that has brought about the enormous consumption of it.

In the form of alloys we now find aluminium being very widely adopted for other purposes than castings, among the most important being electric wiring, and panelling for the bodies of carriages.

The alloys used for crank cases and gear boxes for automobile work usually contain from 10 to 15 per cent. of other metals, thus the alloy recently known as "90 per cent." would contain some 7.5 per cent. of zinc, and 2.5 per cent. of copper, these additions increasing the specific gravity only very slightly. It takes a very considerably-increased addition of copper to raise the weight, as is shown by the figures taken from the Report of the Alloys Research Committee, which are given below.

MECHANICAL TESTS ON PURE ALUMINIUM.

	Ultimate Tensile Strength. Tons per square inch.	Yield Point. Tons per square inch.	Elongation per cent.	Elastic Modula.
Sand Casting ...	5	2.5	25	
Rolled Bars ...	7	0.5	35	
Chilled Castings ...	5.25	2.5	35	
Hard drawn wire ...	15	13	25	9,000,000
Soft drawn wire ...	7	4	30	10,000,000

It will thus be seen that an alloy with 8 per cent. of copper is still a very light metal, and with the corresponding substitution of zinc it would be even lighter. The mechanical increase in strength due to such additions is very marked, and the "90 per cent." metal specified will give 11 or 12 tons tensile, with an elongation of 5 per cent. in the condition of ordinary unannealed sand castings. These results are perfectly satisfactory in the majority of cases, but where greater strength is desired it can be obtained by increasing the alloy metals to 15 per cent. instead of 10 per cent., and further by the addition of certain other metals.

These alloys cast beautifully, and their strength ensures great reliability when attention is given to the contour of the patterns employed.

From time to time the advisability of replacing such castings by sheet metal, gun-metal, or even cast iron has been discussed. Complaints have been made of the unreliability of aluminium and the difficulty of machining it. It has been said to be porous, and unable to stand vibration, and to be too costly for the every-day car. Perhaps all these objections were sound a few years ago, but at the present time they certainly have no foundation in relation to the proposed substitutes. There is a certain substantiality about a well-designed and well-made aluminium casting, which leaves nothing to be desired. It can be designed to hold up against its work in all planes, and allows of fittings and attachments which would be almost impossible with stamped steel. Aluminium is an excellent conductor of heat, and so is much less liable to warping, under conditions of locally applied heat, than any stamped steel crank case would be. The cost of the proposed stamped work of this character would be great, and machining and fitting is still greater than with aluminium, unless perhaps the stampings were made in very large numbers. Gun metal or similar alloys, and cast iron are also more costly to machine and fit, while if similar weights were approached they would be correspondingly weak.

Another interesting property of the metal is its great toughness when hot. In casting it is the practice to take off the boxes as soon as the metal is set. The workman will then hammer the casting smoothly, in order to loosen the sand and tumble it about in a way which would break up a gun-metal or iron casting. This property of the metal is one of much practical value, as it enables the boxes to be got ready for other moulding, and it further reduces the risks of shrinkage cracks when a complicated casting is allowed to cool in the moulds. The latter is an important point, because the contraction of the metal is high, about 3/16ths of an inch to the foot. When the casting is cold the gates and risers are cut off by a band saw, having teeth set like those for hard wood, and these cut the metal quite easily. The chisels also for trimming the casting are very keen and cutting. The metal shows no sign of brittleness during severe chipping, but, on the contrary, will curl away from the tool.

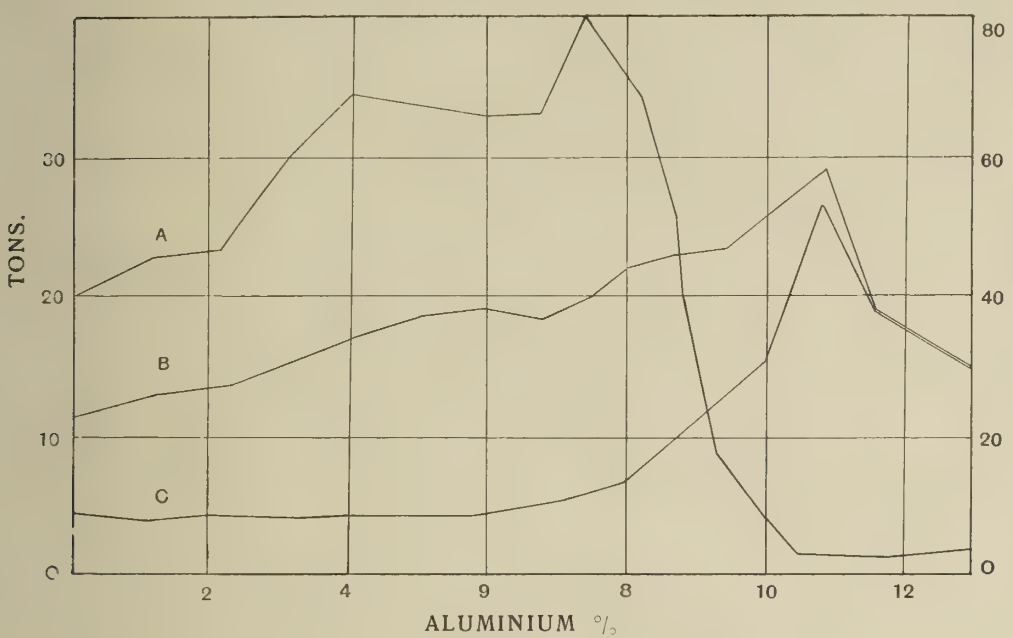
WEIGHT PER CUBIC FOOT OF COPPER—ALUMINIUM ALLOYS.

Copper per cent.	Sand Casting. lbs.	Chill Casting. lbs.	Rolled Bars 13/16 dia. lbs.	Drawn Bars 13/16 dia. lbs.
0.00	168	169	169	169
0.86	170	170	170	170
1.90	171	172	172	172
2.77	172	173	173	173
3.76	173	174	174	
4.97	173	175	175	
6.15	175	177	177	
6.97	176	178	178	
8.01	178	180	180	

To obtain the best results in casting it is essential that no remelted aluminium be used. It is probably true that during smelting and casting silicon is reduced from the sand employed, and also that some oxygen is absorbed by the metal, and alumina formed. As a flux for absorbing this, aluminium cryolite is occasionally used, and for the reasons already given, every care should be exercised in preventing these actions when best results are required, or the metal

will be correspondingly "short." Again, the copper and zinc employed must be of good quality, or the impurities common in these metals will affect the result. The best practice is to work from a stock of molten metal, which is kept stirred and covered, using plumbago pots as ladles, and maintaining this stock by additions of the various metals as required.

MECHANICAL PROPERTIES OF COPPER-ALUMINIUM ALLOYS. SAND CASTINGS COOLED SLOWLY FROM 800°C. (Alloys Research Committee).



A. Elongation. B. Ultimate Stress. C. Yield Point.

The casting temperature is also of great importance, and it is desirable to keep this as low as possible.

Aluminium is regarded as one of the most powerful reducing agents known, and use is made of this fact in steel manufacture, where it reduces any oxide which the molten metal may hold in solution. Oxides of many other metals may be similarly reduced to the metallic form. To meet these considerations, unless the casting temperature is kept low, there is a tendency to get silicon or other impurities in the alloy in addition to defective castings, through drawing or contraction. The alloy described sets in a remarkable manner, and before becoming quite solid resembles an amalgam or plumbers' solder. It is pasty for some time. Advantage may be taken of this peculiarity in the repair of broken castings. If a suitable quantity of metal is melted, and the broken casting heated locally at the break with an ordinary gas blowpipe, the pasty conditions referred to may be easily obtained, and on running the molten metal on to this pasty metal, a perfect joint can be got, the only difficulty lying in preserving the original form of the casting without undue work in trimming and finishing it.

Various attempts have been made to utilise this property of the metal by casting under pressure, but, besides the difficulty of applying such process, no advantage seems to accrue either in the sharpness of the castings or their mechanical strength.

One of the best tests of the character of these metals is the way they machine. If a bar from some of these alloys is turned, shavings may be produced, which have the appearance in contour, and much of the springiness, of steel turnings, while the screwing of the metal by taps is perfectly satisfactory, the only precaution required being that the reamers and taps used shall be well backed off to clear the cuttings.

The alloys of aluminium rich in copper have been studied by many investigators. Probably the best work done in them is that of the Alloys Research Committee of the Institution of Mechanical Engineers, whose report should be studied by all interested. Epitomising these results in respect to their possible application to motor construction, it may be said that the limit of the useful additions of copper and zinc would appear to be under 20 per cent. of these metals with 80 per cent. of aluminium; above such addition the specific gravity increases seriously. The strengths of the metal when higher percentages of copper without zinc are added are very remarkable, and similar, even in the cast form, to those of mild steel, while by suitable heat treatment of such castings results may be obtained which surpass those of such steel. It is unfortunate that the term "Bronzes" has been applied to these alloys, as they contain no tin, which hitherto was regarded as a necessary constituent of bronze. The diagram above gives an abstract of the figures obtained with these alloys.

Of course, the specific gravity of these alloys is high, the following showing a few of them:—

Specific Gravity of Copper Aluminium Alloys.

Aluminium %.	Specific gravity.
0.10	8.92
2.50	8.60
3.00	8.69
4.00	8.62
5.00	8.28
6.73	7.95
7.50	8.00
8.67	7.69
9.90	7.56
10.00	7.61
11.00	7.23
13.62	7.25

In point of weight, therefore, components made from such metals possess little advantage, the specific gravity of steel itself being 7.84.

When higher percentages of aluminium are required we find that about 10 per cent. of copper with some 5 per cent. of zinc give the highest tensile strength when dealing with such alloys of aluminium as those in which lightness and strength appear best to be obtained. The ductility of such a metal is, however, low, and this loss appears to increase when the zinc is increased.

It would seem that the properties of the binary alloys of zinc and aluminium have not been sufficiently investigated. The author has obtained some very remarkable results in such alloys. Some of these showed extraordinary toughness, while the increase in hardness is also marked. A. Saposhnikow has published (Journal of The Russian Physico-Chemical Society, 1908) figures showing the results he has obtained on such alloys. He found that with an alloy of 30 per cent. of zinc and 70 per cent. of aluminium the maximum elasticity (kilos per sq. cm.) was accompanied by the maximum hardness as measured by the Brinell test, in which a weighted steel cone is forced into the metal and the result given in kilos per sq. mm. The figures obtained by Mr. Saposhnikow are given below.

ZINC ALUMINIUM ALLOYS.

Zinc % ...	0	10	15	20	30	40	60	70
Hardness	25.4	47.8	66.0	89.0	107	85.0	67.0	64.3
Elasticity	402	985	—	1840	2530	1900	2140	2230

Zinc %	80	90	94	96	98	100
Hardness	57.0	73.3	74.0	73.0	72.0	36.7
Elasticity	1670	1670	1425	1185	1210	570

These results are very interesting, the effect of small quantities of aluminium on the mechanical properties of zinc being very remarkable. Although no data are given showing tests of tensiles, etc., it may be concluded from the above figures that such tests would show equally good results.

The addition of small quantities of manganese has a hardening effect, and raises the tensile strength of these alloys.

THE 15 H.P. DAIMLER CHASSIS.

A short description of some of the most interesting features and a criticism of some of the peculiarities of the design.

In describing this small Daimler chassis it is not our intention to express any opinion as to the merits or demerits of the engine system. There are many parts of the engine requiring special treatment, particularly the valve sleeves and the cylinder heads, but there are also many other portions common to most engines, and it is these which we think will be most interesting to the majority of our readers.

Throughout the chassis materials of the best quality are used, and most of the castings are made in the Daimler foundry. Many stampings are used, both of steel and phosphor bronze. Aluminium is used for the crank case, the gear box, and the differential case, and these parts are all well proportioned, no undue stresses being borne by them. It has always been the practice of the Daimler Company to use the best obtainable materials, and it is a wise policy. Lately advantage has been taken of the great strength of some of the steels employed, to reduce the weight of certain parts, and on the larger models the vanadium-chrome crankshaft is now hollow, and on all Daimler chassis the gear, axle, and other shafts are below the average size.

The Daimler practice of putting simplicity of handling before most other considerations is retained for the 15 H.P., and thus, while there is no doubt that a more delicate carburettor could be found, the one fitted is certainly free from complication and has no parts liable to derangement. Likewise, the use of the springs to take the torque of the axle case enables a part to be dispensed with, even if the necessary stiffening of the springs makes their action a little less easy than it might be.

The essential details of the engine are shown in Fig. 1. The sump is of enormous depth, the bottom being about six inches below the oil troughs, which seems almost more than necessary, especially as the engine is unusually deep on account of the length of connecting rod necessary to give a sufficiently small angle of obliquity to allow the rods to clear the mouths of the valve sleeves. The pairs of cylinders are easy castings, but the heads are a somewhat troublesome job, and it is to be regretted that their form makes it impossible to empty the water from them, except by the use of a squirt, which is supplied with each car. It would be a great improvement if syphon pipes were taken from the bottom of each head and (passing through a plug in the top) led down to the drain taps of the outer jackets. It would not be at all difficult to do and not costly. Also the drain taps are fitted on the off side of the cylinders in such positions that they discharge directly on to the magneto and carburettor attachments, while the opposite side of the engine is quite clear of all obstructions. It would be well worth while to go to the trouble of the slight alterations necessary to the cylinder castings to enable the taps to be fitted on the near side.

The engine is supported by four arms, cast integral with the upper part of the crank case, and is mounted in a sloping position, to give a straight line drive to the worm beneath the back axle. The crankshaft has five bearings of an aggregate length of about 250 mm., the diameter being 45 mm. As we have said already, it is a stamping in vanadium-chrome steel, and it is finished by grinding only, the webs being left in the rough, or rather, merely cleaned up to remove any excessive roughness. The main bearings and the big ends are babbitt lined, and the shaft is supported, as usual, from the upper half of the crankcase.

The pistons are cast iron, with a web between the heads and the gudgeon bosses, the former being made concave partly in order to obtain an approximately spherical form for the combustion chamber, and partly to prevent the engine having to be still deeper to get the desired compression space. The gudgeon pin fixing is made clear in the engine section, and the pin and bush are both hardened and ground steel.

The sleeves are made of a special and secret mixture of cast iron, and are cast upright, the principal difficulty, encountered in the foundry, being to obtain sufficiently thin walls without the presence of numerous blow holes. The bridge-piece which crosses the centre of each valve port is cut away slightly on its outer face, and it is found that carbon deposits in the space left and is kept smooth by the rubbing action between the sleeves and the cylinder, whereas, if the bridge is left full, deposit is liable to cause trouble. Probably the part of the Knight engine which has suffered most adverse criticism is the one-sided drive of the sleeves, but when their great length is considered, it is more mechanical than it appears at first sight, and if two eccentric shafts were used, one on each side of the cylinders, they could never be made to work in exact time together.

The eccentric shaft runs in phosphor-bronze bushes, and the sleeve connecting rods are unbushed phosphor bronze stampings. The shaft is driven by chain and carries two skew gear wheels, one at the front end, for driving the cross-shaft, and the other near the flywheel end, for driving the oil pump. The cross-shaft is one of the most excellent features of the whole car, as its use renders the pump and magneto extremely accessible, and the forward position of these fittings leaves plenty of clear space on each side of the cylinders, so that the carburettor and other attachments are easy to get at. The magneto drive is an ingenious and effective arrangement. In Fig II. it will be seen that the magneto and the shaft each carry a fork; between these forks is a disc of leather, and the four bolts for securing the forks are spaced evenly round it. One of the forks has slotted spaces for the bolts, allowing a small amount of adjustment for setting, and the forks being connected simply by the flexible leather, the drive is a quite universal one.

The pump has a good self-adjusting gland with an easily accessible nut for tightening the spring. The drive connection between the cross-shaft and the pump shaft is made by a cotter, seen in plan in Fig. II., it being secured in place by a clip ring of steel wire. In the most recent chassis the fan has been driven from the centre of the cross-shaft by another skew gear operating through a spring, or shock absorbing coupling.

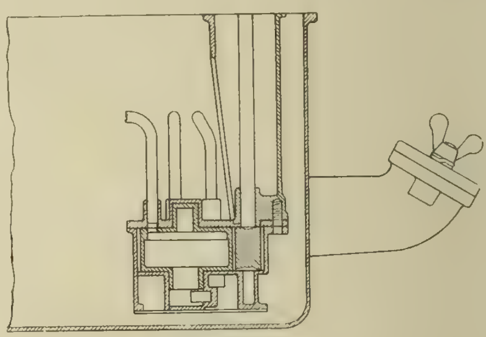


Fig. III.

The lubrication system is made plain by Figs. I. and III. The troughs in which the big end scoops dip are cast integral with the lower part of the crankcase, and it will be noticed that the dippers used are considerably larger than those fitted to the 16-20 Wolseley, though all oil is distributed by splash from the troughs in an exactly similar manner.

There is a gutter carried right across each division plate of the upper half of the crankcase, and the oil caught by these is led to the main bearings. All other parts depend for their lubrication upon the impact of flying particles of oil flung by the scoops. The oil pump, Fig. III., is arranged to supply through a rotary distributer opening to each trough in turn, and also to a dashboard indicator. Oil is drawn in from the sump by the helical gears through the intake I, and is forced by the same gears through the passage O to the

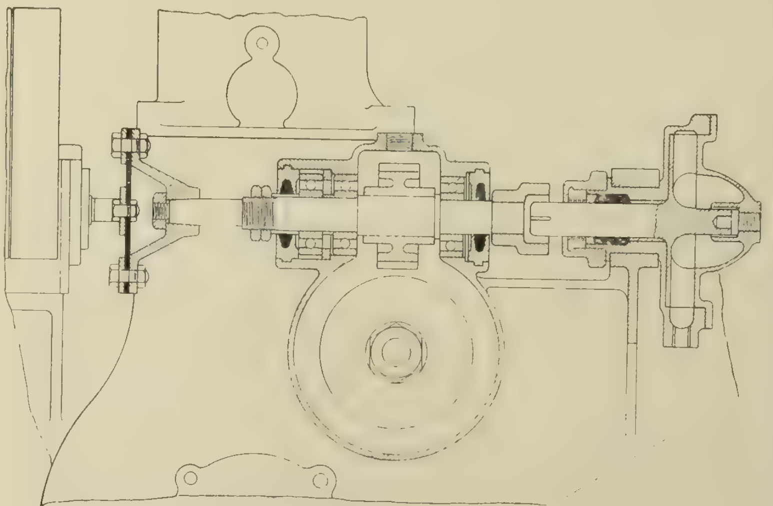
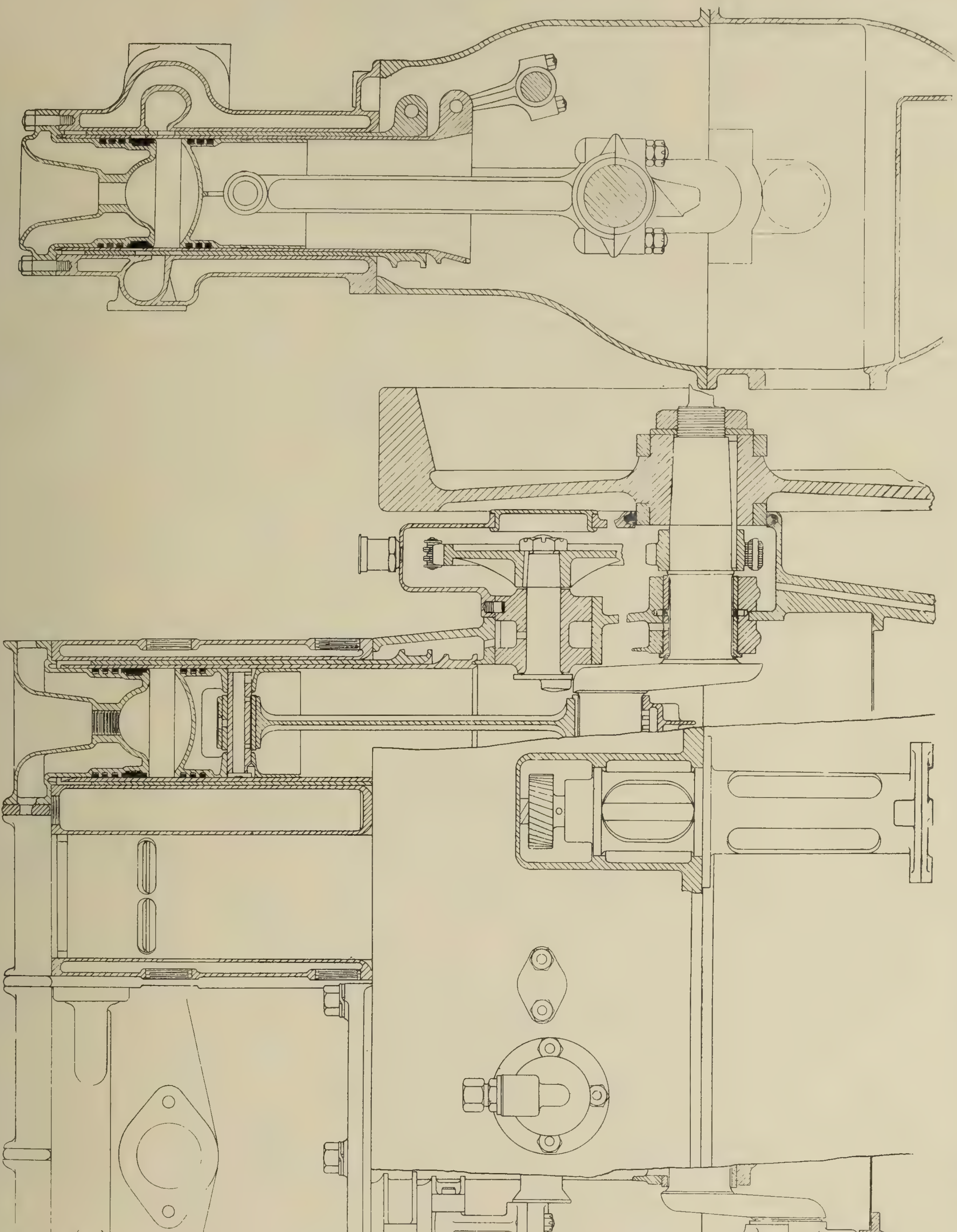


Fig. II.



Scale approximately 1 mm. = 3.9 mm.

Fig. 1.

THE 15-H.P. DAIMLER ENGINE.

inside of the large gear wheel. The upper surface of this wheel has a slot about an inch long on the top and near the circumference, and this slot registers in turn with each of the $\frac{3}{8}$in. outlets to the pipes. The pump is driven from the eccentric shaft as shown in Fig. 1.

In common with most cars with pump fed oil, it is necessary to let out all the oil if it is desired to examine the pump, though it would be easy to arrange for the whole fitting to be withdrawable upwards. A point in connection with the lubrication system which might also be improved is the filtering of the oil, as there is no filter in the filling pot which leads directly to the sump, and there is also no filter at the intake of the pump, the only gauze in the whole arrangement being a tray, or series of trays, joining the troughs and separating the upper part of the crankcase from the lower part. This is scarcely sufficient for efficient filtering, especially as gauze is not always used, but is sometimes replaced by perforated zinc.

It should be noticed that the extra air intake of the carburettor is also part of the lubricating system, as it draws its supply of air from the crankcase. This air is heavily laden with oil, which it carries into the sleeves at each intake, and

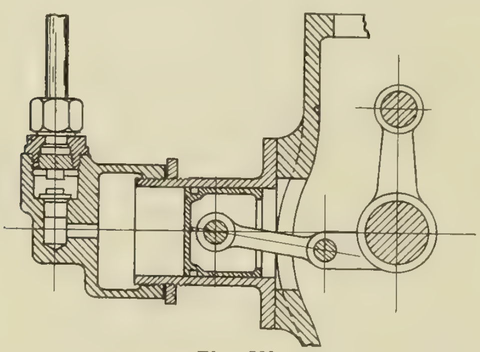


Fig. IV.

it is not possible to run the engine at its maximum speed without opening this extra supply pipe.

Petrol is supplied to the carburettor under air pressure obtained from a small plunger pump, Fig. IV., driven off the eccentric shaft. The carburettor is a modification of the G.A., and has two jets, the second being uncovered as the throttle is opened. Air is admitted round the jets, and also by the lifting from their seatings of steel balls of different sizes in the base of the mixing chamber, so as the suction increases the light balls lift first and the heavier ones later. This arrangement is fairly economical, and also it permits slow running and enables high power to be obtained with rapid accelleration. The extra air (and oil) inlet mentioned above is a small copper pipe with a central tap controlled by a pedal.

The control is by means of throttle and spark variation with hand lever only, and the connections, though stout enough to be durable, are capable of improvement. They could be operated with greater ease and smoothness if their arrangement was altered so as to reduce the angularity of the joints, which could then be adjusted more tightly.

The ignition is performed by a C.A.V. dual outfit, and the coil is carried on the right-hand front crankcase arm. This is neat and enables the wiring to be kept short, but it has the disadvantage that the bonnet has to be opened in order to press the starting button, though the throw-over switch is linked up with a control lever on the dashboard.

The cooling system has ample efficiency, in fact it errs slightly on the side of over-cooling. The pump has been described already, and is shown in Fig. II. The radiator has flat vertical tubes of deep narrow section with no gills, and the fan is an aluminium casting. All waterways are large and the jointing of the cylinders to each other by means of rubber rings is clever, and satisfactory in use.

The clutch is the usual leather-faced cone, and the spring is mounted just beneath the footboards operating through a lever gear and a ball thrust washer. The spring is certaiunly very accessible, and, being external to the clutch, allows the latter to be taken down with great ease, but the use of a thrust washer which is always running is a serious drawback. There is no objection to the leather clutch for a car of this size, especially one with such a flexible engine, but the peculiar arrangement of clutch spring does not present advantages commensurate with its disadvantages, and even the fact that it has been used with fair success on Daimler cars for many years, is insufficient to make it good practice. The grease cup which supplies this constant running thrust is also awkwardly placed beneath the bearing, while it might just as easily be above. The thrust of the clutch is resisted by the washer at the front end of the crankshaft seen in Fig. I.

The gearbox is an excellent example of compact and stiff design, and is of ample size for the power it transmits. The ratios are 4.37 to 1 on direct top speed, 7.15 to 1 on the second, and 15.8 to 1 on the first and reverse speeds. There are 36 teeth on the large wheel of the permanent drive, and 22 teeth on the smaller. The second speed wheels have 29 teeth each, and the first speed pair have 18 and 40 teeth respectively. The permanent drive pair are 30 mm. wide, the small secondary shaft pinion being made in two portions. The second speed wheels are 23 mm. wide and the first speed pair 18 mm. The depth of engagement of the annulus top speed clutch is 14 mm. These gear ratios are all low, but the road wheels are of fair size, being 870 mm.

The spring tightened felt rings for preventing the leakage of oil are very neat and are neatly applied. The box is slung by adjustable single bolt hangers from the middles of two cross members of the frame, and it is prevented from turning by an arm cast with it and anchored to the off frame side. The fact that the gears are not completely silent in operation is somewhat curious, as it would be difficult to design a box which, theoretically, would be more likely to run

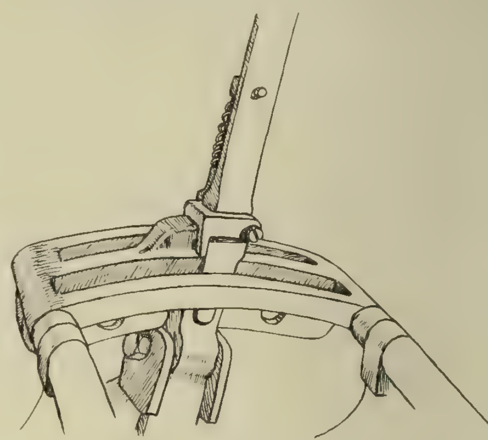


Fig. VI.

silently. Indeed, this is rendered all the more remarkable because it is not too much to say that no British firm have taken more pains in the endeavour to find a process of manufacture of quiet spur gearing than the Daimler Company. Though they have spent years in research work, it is still found necessary to test each gear on the road, and often the practice of changing gears from one box to another has to be resorted to before the result is satisfactory. This gearbox is one of the few in which the reverse pinion is not running, except when it is in use. It is engaged by a fork made solid

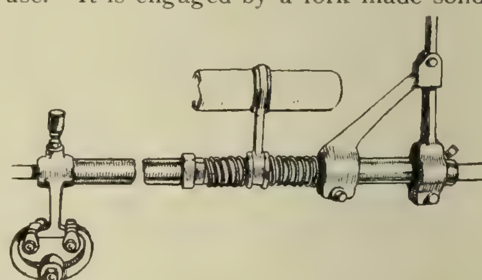


Fig. VII.

with the first speed striker, and moves idly on its shaft when the latter is engaged.

The method of controlling the changes is shown by Figs. VI. and VII. There is a double locking arrangement, one on the gate itself and one at the inner end of the cross shaft, external to the box. The gate has two holes in the dividing bar, and the lever has projections on either side, which drop into the holes when the lever is in either of its limiting positions. The lever rocks on a hinge at its foot, and is held in the neutral position by the spring seen in Fig. VII. The action of the horseshoe selector lock for the striking arms needs no explanation.

Transmission from the gear-box to the back axle is by means of a double jointed propeller shaft with joints of the pattern shown at opposite ends of the

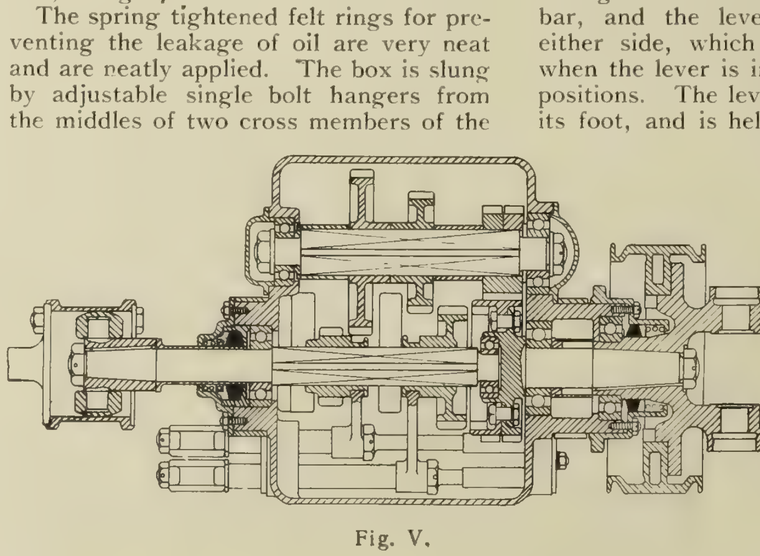


Fig. V.

gear-box in Fig. V., the pin joint being, of course, the one that is connected to the propeller shaft. The front joint has renewable bushes, but there is no other point which calls for comment in connection with either of the pair.

The back axle is shown in Fig. VIII. It is built up with an aluminium central case, split horizontally and bolted to conical steel tubes, which are brazed to the cylindrical ends of the sleeves, being a tight driving fit before this final securing. The malleable iron brake brackets and spring pads are likewise brazed to the sleeves. The weight is carried principally upon the extreme ends of the sleeves, though a certain amount of it is necessarily resisted by the shaft, which, by the way, is supported at its inner end only. The design would be mechanical enough if the point of contact of the tyre with the road came directly under the single ball bearing in the hub, but as it is, the slight bending moment, which must always be present, can only be resisted by the differential. It is, however, probably cheaper to make the axle large enough to be sufficiently strong as it is, than it would be to incorporate a second hub bearing.

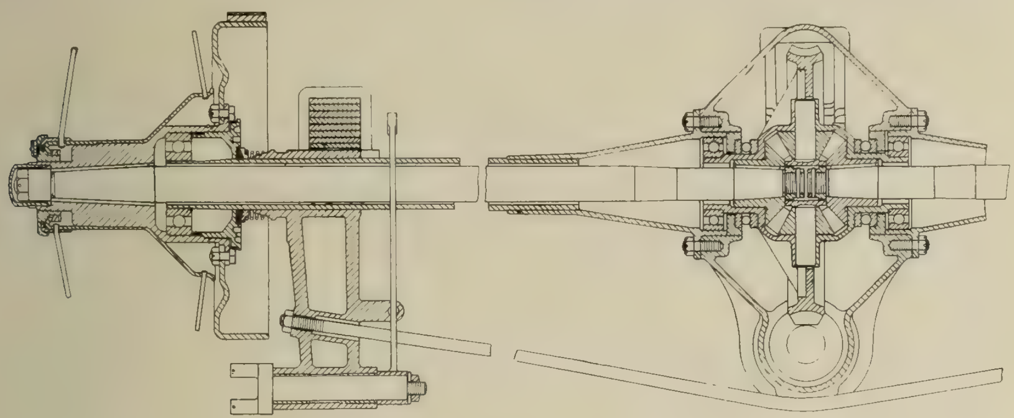


Fig. VIII.

The exact details of the design of the worm drive are a closely guarded secret at the present time, but it is the result of investigations based on the Lanchester worm gear. It runs rather cooler than the majority of its kind, whence it is safe to assume that its efficiency is satisfactory. The worm is supported by the usual ball bearings and thrust races.

The Rudge-Whitworth wire wheel shown is the standard fitting. The brake drums are pressed and bolted to the cast steel hub centres, and the external bands of the brakes are contracted by a toggle, which also serves to separate them positively when in the off position. The hub brakes are foot-operated, and it is difficult to see why they should be external when the expanding pattern is equally efficient, much more easy to protect, and generally sweeter in action. The tie rod is not in a position of advantage, as it too nearly straight, and would be more effective were the ends brought as close as possible to the axle, and this would not appear to be a very difficult alteration to effect. A peculiar point about the axle is that there is no drain plug to it, which means that it is not possible to remove old oil, and the particles of metal it contains, without taking down the whole axle. This is, no doubt, an omission which will shortly be rectified.

There are but few other parts of the car of peculiar interest, except, perhaps, the use of the springs to take the torque of the back axle. This does not appear to affect adversely the ease of the springing, but supplementary spiral springs are fitted to improve the suspension. The car is unusually comfortable, but it does not hold the road very well, and it is possible that the amount of rolling which the spirals permit has some connection with this fact. The front springs are mounted on sliding shackles at their rear ends, as shown in Fig. IX. The bolt may appear to be rather small, but is quite sufficiently strong to withstand all ordinary shocks. The steering might be improved, and is peculiar in that the bearings in the worm box are plain, which tends to heavy action, and the substitution of ball bearings would make a considerable difference, but taking into account that the car does not hold the road really well, we might suggest that some experiments as to the setting of the tie rod arms would be likely to yield the best results.

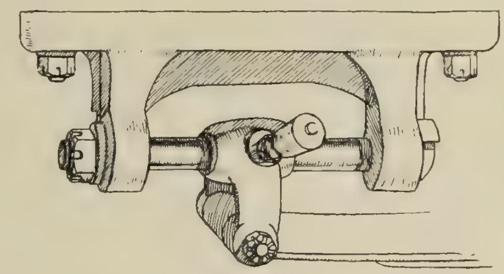


Fig. IX.

The very low direct top speed enables the car to accellerate with great rapidity, and also renders the use of an indirect gear very seldom necessary, but the absence of a fourth speed cripples the car in a most striking manner. The running is extremely smooth up to a speed of about 30 m.p.h., but above this the engine begins to make its presence felt. That is to say, the best running speed for the engine is between 1,000 and 1,300 r.p.m. The fourth speed would, of course, raise the cost of production a few pounds, but no possible owner who had tried the four would ever be content with three, even if the extra ratio meant an increase of price of £25, which would more than cover the addition. Another improvement which could be made at small cost is the addition of thrust washers to the front hub bearings, which are devoid of any means of resisting axial loads, except the lateral stiffness of the two single row ball bearings used.

As we have criticised the design freely in several respects, we should like to lay stress in an equal degree on its good points, which are numerous, and in some instances striking.

Firstly the engine is capable of exerting unusual power at very low speeds of revolution, and it pulls smoothly at even the lowest of them. The power of acceleration on hills is also unusual, even when due allowance is made for the low gear ratios and the capabilities of other cars of similar cylinder capacity. A most impressive point is the complete absence of any sense of labouring when running up a hill sufficiently steep to slow the car down to little more than a walking pace, with the direct drive still in engagement.

The springing is distinctly above the average of ease, especially on really rough surfaces. The easy riding over bad roads is assisted by the large wheels (the use of which cannot be too highly commended), and is probably affected to a great extent by the supplementary spiral springs at the rear ends of the rear springs, and by the slides which replace the usual shackles for the free ends of the front springs. These slides certainly work more freely and smoothly than the ordinary shackle.

At speeds below twenty to thirty miles an hour, according to the nature of the surface of the road, the car is practically noiseless, and this means that great care has been taken to so dispose all the small parts, such as brake rods, pedals, dashboard fittings, and so forth, that they are either rigid and incapable of movement, or so that there is free space for any vibrations to which they are liable.

From a machine-shop point of view the whole chassis is a straightforward job. The largest single piece is the crankcase, and it will be seen from Fig. I. that there is nothing of a complicated nature about it. Needless to say, every operation is done to jig, and great advantage is taken of the recent development of grinding machines, as instanced by the crankshaft, which is finished almost entirely by grinding, and is not turned at all. In very few places are two parts employed where one would serve the same purpose with equal efficiency, but on the other hand parts are not combined for the sake of combination when they are more easy to handle separately.

Probably the fairest criticism of the performance of the 15 h.p. Daimler engine which could be offered, would be to say that it is an exceptional combination of touring smoothness with something very nearly approaching to racing efficiency. At the lower speeds it is exceptionally quiet, and at the higher speeds it develops really high power for its size, but it is no quieter or freer from vibration than an ordinary engine when it is working hard and fast. At the same time it will be seen that this gives an unusual combination. It is easy enough to make an efficient engine, one which will develop high power for its dimensions, but which will never be pleasant when running at slow or moderate speeds. *Per contra,* it is easy to make a quiet, smooth engine which will never be efficient for its size, so that to some extent in the 15 h.p. Daimler we get two engines in one: a silent smooth engine at one end of the scale, and a very powerful, albeit, rather hard running engine at the other, though it is no harder than other efficient engines of similar dimensions.

CYLINDER MACHINING.

A consideration of the different methods in use and of their relative values.

It is impossible to discuss the various methods of treating automobile engine cylinders in the machine shop without some consideration of the advantages and disadvantages of different forms of cylinders. The matter is principally important on account of the growing popularity of four cylinders cast in one piece, which will be called "monobloc" casting for the sake of convenience. Admittedly monobloc casting has grave faults from the purely foundry and workshop point of view, and at the present, this is the only standpoint to be considered.

First of all there is the indisputable fact that the patterns for monobloc castings are much more costly than equivalent single or pair castings. Not only are the patterns larger, but they are far more intricate, and so require to be made with unusual care to ensure their ability to withstand foundry wear and tear.

The founder's opinion in the matter is quickly given, and it is adverse to the monobloc casting, but he should not be given much sympathy, simply on the plea that the work is more troublesome and requires more care, but, assuming that the work is sent to a first-class foundry, it remains that with every possible care the percentage of wasters is higher. Unfortunately for all concerned the defects in cylinder castings are often only discovered when the hydraulic test is applied, after they are bored out, by which time a considerable expense has been incurred.

The larger the bore, the more serious does this defect become, and although monobloc cylinders can be cast successfully up to about 130 mm. bore, it seems very questionable whether a line should not be drawn at 80 mm. or 90 mm.

One of the leading French firms who have studied this question of monobloc cylinders very closely, and have achieved a unique measure of success in casting them, has fully confirmed these views, although, viewed apart from the question of wasters, they find that the actual cost of production in labour and material is proportionately less for the monobloc system. This is more than counterbalanced by the percentage of bad castings, as may be gathered from their statement that in extreme cases fifteen or twenty castings have to be made before a good one is secured.

It is very interesting to note in regard to this question of wasters that the percentage is highest in those castings which have the induction and exhaust passages cored out of the jacket body, so the æsthetic simplicity of design which this system produces has to be paid for somewhat dearly.

The opinion of this French firm is that the monobloc system should not be adopted for bores in excess of 90 mm. or 95 mm., as it is difficult to maintain the very high casting temperature necessary for getting large areas of metal only 5 mm. thick.

Arriving at the machine shop further objections rapidly reveal themselves. The extra weight, for instance, is troublesome from start to finish. Any ordinary pair cylinder casting can be handled and carried about by one man, the monobloc requires two or three men to handle it, and a truck to carry it about. Much larger or special machines are required, and this point alone might be enough to condemn the system in small shops. This, however, is an argument of small account, as it is based on the circumstances of the shop, and not on the intrinsic merits or demerits of the system.

Considering the jigs and fixtures, one authority estimates the extra cost, as compared with double cylinders, as about one and a half to one. This is not very serious when spread over a large number of engines, but the greater size and weight of the jigs entails an increased cost in handling which is continuous.

There is the final disadvantage that when, as occasionally happens, some defect only shows up on the test bench, it is a serious waste of money to have four cylinders to scrap instead of only one or two.

It would be unfair to ignore the good points of the monobloc, even though the strongest be the æsthetic one. There is a clear saving in having only one water inlet and outlet in place of two or four of each, and still more in the simplification of the water piping. Then an exhaust pipe can be cast integral with the cylinders, which also saves money, though, as has been pointed out, it increases liability to waster castings. Still, if a money value is put on all these savings, 25s. or 30s. will cover it easily, and this is hardly enough to compensate for the disadvantages.

For quite small cylinders, say up to 80 mm. bore, the objections do not apply so strongly.

So far the matter has been examined purely as a general problem, but when it is considered from the individual workshop standpoint it is impossible to ignore the circumstances of the particular works, or in other words, the plant available in each case. It is safe to say that the average automobile factory has not got special tools suitable for dealing economically with monobloc castings, and might hardly be justified in incurring the expense of getting them, even if funds were more plentiful than they are as a rule.

Summing up the conclusion on the question of monobloc, double, or single cast cylinders, it appears that for all-round convenience, pair castings are best. This is corroborated by most of the opinions we have received, and by current practice, as indicated by the exhibits at the last Olympia Show, where the monobloc, although more in evidence than on previous occasions, was almost entirely confined to the smaller engines.

When cylinder castings come into the shop the object should be to complete the machining in the quickest possible time and with the smallest possible amount of handling. Although this may appear to be a platitude, it will bear repetition, as it certainly has not been appreciated everywhere at its full value. The marking-off table still flourishes in many shops where, on the other hand, jigs are conspicuous by their fewness, because they are "too expensive"! One wonders in such cases whether those in charge have ever totalled up the annual cost of the marking-off staff!

There can be no doubt at all that in any factory there should be such a complete system of jigs and gauges that the marking-off table need only be used for occasional new and experimental work. In making jigs for cylinders it is hardly possible, nor is it desirable, to manage with one only, but two ought to be sufficient. In designing them it should be borne in mind that although great rigidity is necessary this should be attained with the least practicable weight, and also, that not only should the jig locate and accurately gauge every opera-

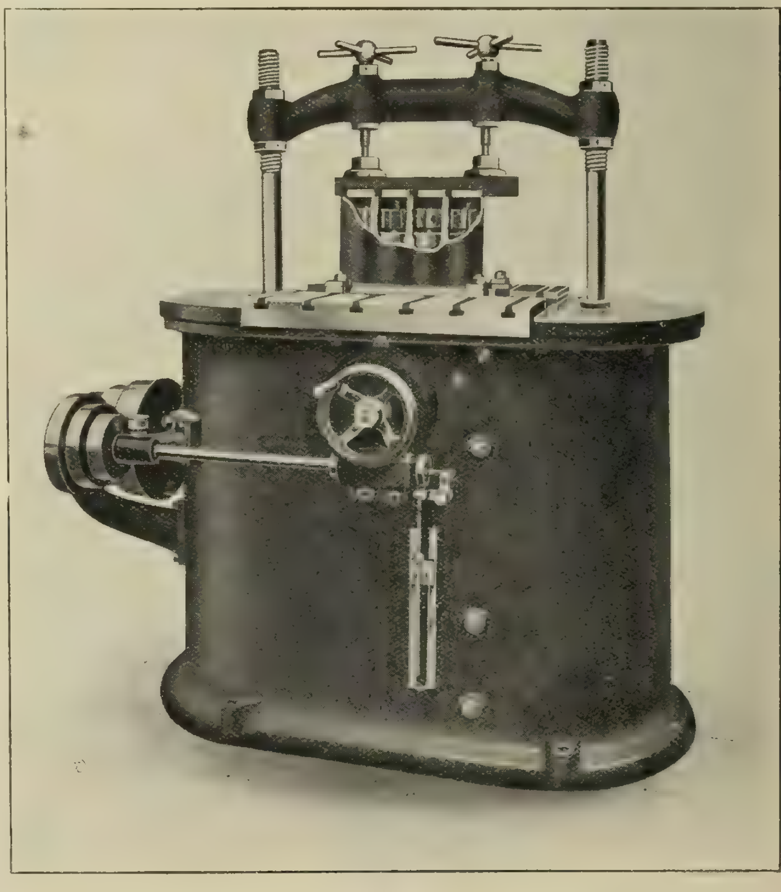


Fig. 1.

tion required, but, by its own shape, it should lend itself to ease and convenience in handling.

In commencing to deal with a monobloc or double casting, the first operation is to plane or mill off the bottom flange, which will then form a datum for further measurements. The second will then be to plane off the top to a height fixed by a standard length gauge. Here it may be of interest to refer to a special milling machine which is very useful for this initial work on monobloc cylinders, especially if they are cast, as in the old Deasy engines, in one piece with the upper half of the crankcase. It is called the "Giant" and is made in two sizes, of which the larger has a table 59in. by $21\frac{1}{2}$in., the traverse being 59in. by $27\frac{1}{2}$in. The maximum height from table to frame (under side of head) is $16\frac{3}{4}$in., and the nose of the vertical spindle can be lowered to within $7\frac{7}{8}$in. of the table. Fitted with a 4-step cone pulley and a back gear, also 5-step pulleys on the traverse motion, this machine is of very rigid construction (as may be judged from its weight, which is 7,900 lbs.), and although not fitted with very refined automatic stops, it does capital work and is sold at a remarkably moderate price. Its long traverse enables it to deal with a number of castings at once.

The faces having been machined—in the case of double cylinder castings this is often done in a lathe—the next step will depend upon the available shop equipment and the bore of the cylinders. It is possible to bore out either monobloc or the double cylinders on a good large engine lathe, but it certainly cannot be called a nice operation in either case. For convenience and speed a two-spindle boring mill is a more suitable tool for the work, or failing that, a single bar mill will do. For this operation the casting can conveniently be put into a jig which will locate the bores. In works with small output this jig might do for all the other operations which follow the boring, but generally speaking it will be found more convenient to have a special jig bolted to the boring machine, and forming a semi-permanent attachment to it. When this is used it will be best to drill the bolt holes in the bottom flange immediately after the first facing process, so that the casting at once locates itself on the boring mill jig by the bolts which are used to secure it to the latter.

There is a point in cylinder design which suggests itself at this stage, and that is the desirability, or otherwise, of having a close top or one with a plug hole. Current practice in respect to this is very varied, but it would seem that makers decide the point to suit their equipment. Some boring mills require an outer bearing for their bars, and for these a plug hole to suit the bar becomes a necessity. In some cases the hole is left for the support it affords the cores in the foundry. The closed top, however, should usually be preferable, as a plug involves extra cost, besides the remote possibility of trouble from leakage. It is better to have a good stiff boring spindle which does not require any outside support. Owing to the very thin walls of motor cylinder castings and their consequent liability to distortion, it is not desirable to take heavy cuts, so each boring spindle must have two, three, or even more, boring heads, and take successive light cuts with a comparatively quick rate of feed. There ought always to be, but unfortunately is not, a special tool for cleaning up the tops of cylinders, not only for the purpose of removing possible minute roughnesses and projections which might become incandescent in working, but also to ensure uniformity in the cylinder volumes.

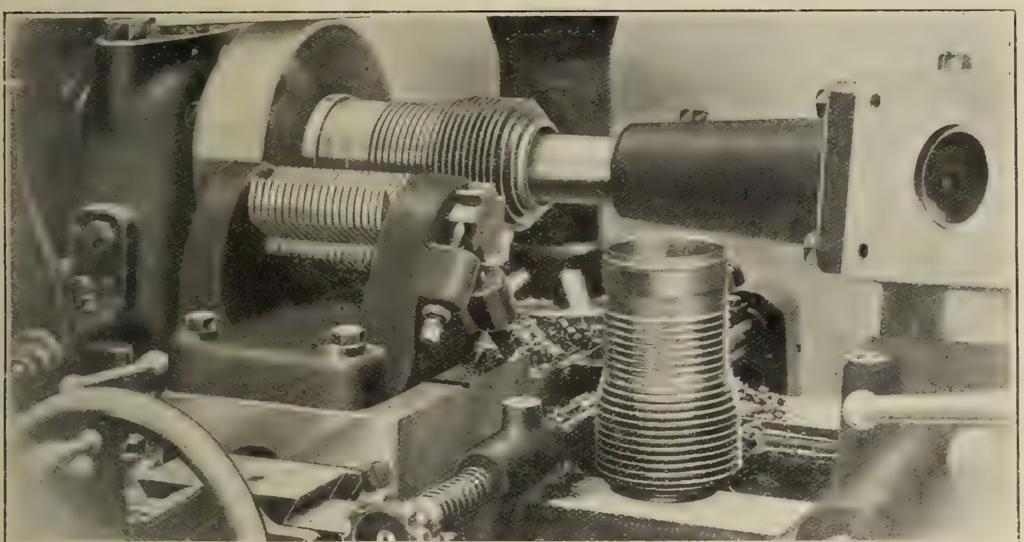


Fig. II.

A works manager has not only to consider how to do a thing, but how to do it quickly, and there is no doubt that constant stimulation has led, and is still leading, to the evolution of some highly ingenious and wonderfully efficacious tools. A two-spindle boring mill will obviously get through twice as much work as a single one, which again leads to the equally obvious further step forward, a four-spindle machine.

We have not yet heard of such a machine being used in this country, but the one shown in Fig. I., supplied by Messrs. Burton, Griffiths, is being used by several French makers, who have specialized in monobloc cylinders. Amongst others, the Renault Company have several of these machines at work on their cab engines. It consists of a heavy oval casting with a table top, and a vertical slide inside on which travels a carriage containing the boring bars. The latter have bearings of square form bored to receive the adjustable bushes, in which the bars work.

The pitch of the bars can be varied from 75 mm. to 140 mm. by means of suitable packings placed between the bearings, which, when adjusted, are all bolted solidly together thus forming a rigid block, which in turn is bolted to the carriage. The maximum bore available with this machine is 140 mm., and the minimum 75 mm., whilst the greatest length which can be bored is 300 mm.

The bars of cast steel have four gears at their lower ends, which mesh two and two with two long worms placed right and left of the bars, which arrangement not only permits of very close setting of the bars, but also allows the gears to be of larger diameter than they could otherwise be.

These worms are connected together by means of two spur gears, and are driven from a vertical shaft by bevel gears from a three-step cone. The boring bar carriage feed is by a vertical screw and bevel gears driven by a worm gear from a belt cone, a friction device being provided which allows the feed to slip when the required depth is bored. The time taken to bore out four cylinders 75 mm. bore is thirty-five minutes.

Another special machine—or rather a special fixture in connection with an already existing type—which has been produced in response to the aero-motor designer's demand, is seen in Fig. II., which shows a Herbert combination turret lathe dealing with a "Gnome" cylinder. These cylinders are cut from sections of steel bar weighing 67 lbs. each, and when finished they have walls $1\frac{1}{2}$ mm. thick, and weigh $5\frac{1}{2}$ lbs., the whole operation being completed in three hours.

It has already been suggested that the Gnome type of engine might be used for car work, but without going so far as this it might be worth while to consider the great saving in weight which could be obtained by the use of such thin steel cylinders with sheet water jackets and overhead valve gears.

Another suitable machine for dealing with ordinary cylinders, either single or pair castings, is a combination turret lathe, and we will take another of Messrs. Herberts' as an illustration. It will be found, we think, a more suitable machine in every way than a single or double spindle boring mill. Fig. III. shows a pair cylinder casting CC, clamped up in position for boring and facing. Fitted to a heavy face plate B, by means of a V-groove slide, is a cradle A, in which the

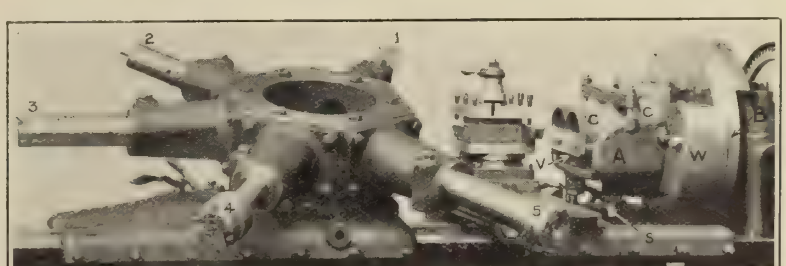


Fig. III.

casting is clamped by means of the cross-bar and studs as shown. For locating the casting there is placed, centrally under one cylinder barrel, a V block, the height of which can be adjusted by the screw seen below it, and under the other barrel is a plain adjusting screw S, both adjustments being fitted with lock nuts. For lengthwise adjustment there is another square-headed screw on the slide at the back.

In the cradle slide is a steel positioning pin, and in the back plate are suitably located steel bushed holes, into which this pin fits. A counterweight W fits into the slide and is held—at either side, as required—by the knurled head pin. This balances the cylinder, which is "off centre."

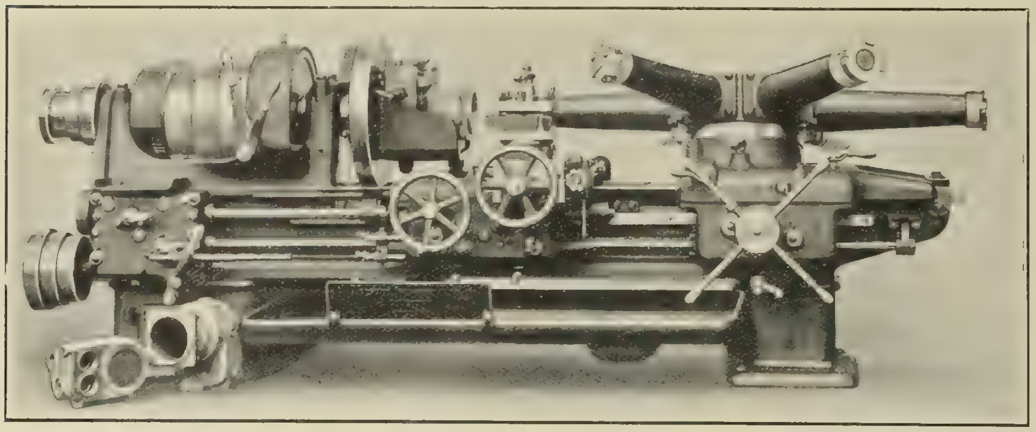


Fig. IV.

The first casting of a series having been adjusted, together with the various feed stops on the machine, No. 1 boring bar on the main turret is swung into place for the first, or roughing cut. The illustration shows a single pointed tool, but as a matter of fact the latest practice is to use double cutters on this first cut. Simultaneously the facing tool on the square turret T is fed up so that the bottom flange and first boring cut are performed at the same time.

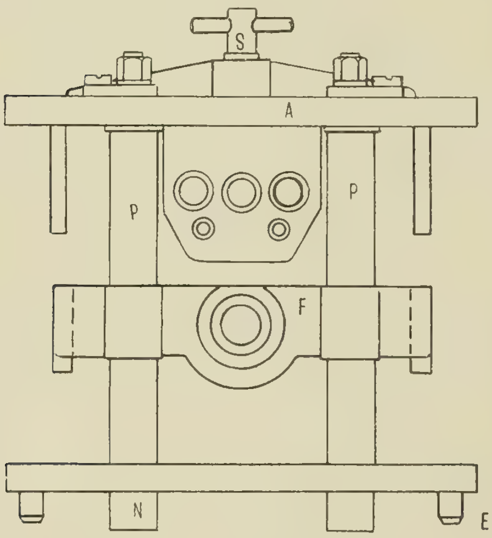


Fig. V.

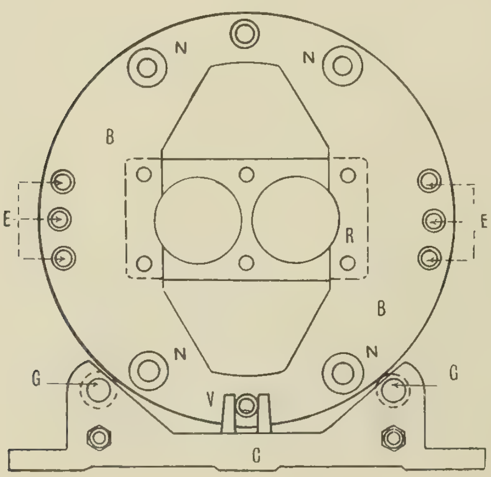


Fig. VI.

This is followed by two further light cuts by bars 2 and 3, and, if it is the intention to grind out the cylinders, this finishes the boring. On No. 2 bar, near the socket, will be noticed a small projection. This is a cutter which effects the slight coned enlargement of the bore at bottom of the cylinder. No. 4 bar has a head with serrated cutters for cleaning the cylinder tops, behind which will be noticed a steady bush which fits the cylinder bore accurately. The serrations of this tool overlap, so a smooth, clean surface is formed. No. 5 bar is used when cylinders are not to be ground and is provided with a floating reamer head, having four 3-tooth cutters, the teeth of which are unequally pitched so as to prevent jarring. This tool leaves the bore so clean that subsequent grinding is said to effect no improvement.

Having completed all the boring and facing, the cradle is removed from the back plate, and another sliding plate is substituted, which holds up and locates the cylinders—reverse way on—for machining the valve seats and plug holes. For this, of course, the main turret bars have to be changed, but that is a very simple operation; and the subsequent processes are executed very quickly. The time occupied in completely machining one pair-cylinder casting in this machine should be about 2¾ hours.

Needless to say, there are many good makes of turret lathes, in which the work can be handled on similar lines, but the above description gives a good idea of the equipment and procedure. Modifications in the holding fixtures will be necessary according to the shape of the cylinder casting and whether the valves are all on one side or on opposite sides, but the general lines of treatment will probably not vary to any great extent. Single cylinders are similarly dealt with, as shown by Fig. IV.

American practice would seem to follow very similar lines to our own, but if the machining processes do not exhibit any striking differences compared with European ones in the matter of jigs, they perhaps show more enterprise and ingenuity. The Pierce Arrow Co. have kindly placed at our disposal some particulars of their shop practice. They evidently believe strongly in six-cylinder practice for all their 1910 models are of this type and of three powers, viz., 36 h.p., 48 h.p., and 66 h.p. The cylinders are cast in pairs, with valves on opposite sides. The first process is to face off the bottom flange. It is not stated how this is done, but it is clearly an operation by itself, requiring the casting to be clamped up in a machine and taken out again after the facing—which, it will be admitted, compares rather unfavourably with the turret lathe system described above. Then follows another separate operation in the drilling of the six bolt holes in the bottom flange by means of which it is bolted to the crank case, and which are important in that subsequent operations are worked off these holes. For drilling them a special jig is provided, which deals with only these six holes. Next, the cylinders—two pairs at a time—are bolted up to a heavy angle plate arranged on a cross slide in a double spindle horizontal boring mill. Two sets of boring heads are used, one taking a roughing cut, and the second finishing within limits required for grinding, which follows as a later operation. Suitable stops are provided, so that after the first pair of cylinders is bored the next pair is brought into absolutely correct centres by sliding the frame across.

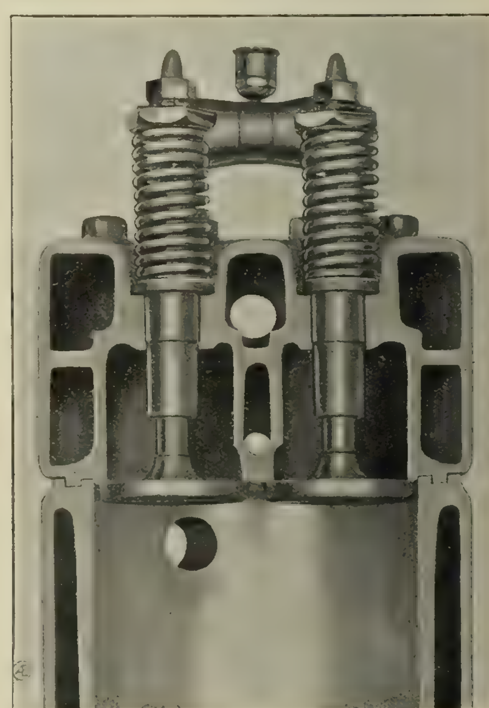


Fig. VII.

The next step is to drill, and tap where necessary, all the small holes, and for this purpose a very useful and ingenious jig has been devised, in which ease of handling has been effectively considered. It is shown in Figs. V. and VI., and consists of two circular plates, A and B, held together by four posts P, and having an intermediate supporting frame F. Four turned nuts N, which hold the bottom plate in place, also form the feet on which the jig stands when in a vertical position. The top plate is easily removable when it is required to put in or change a cylinder, and is provided with a central clamping screw S, by which the casting is held

tight down on the base, where it is located by four dowel pins R, fitting into the end holes on the bottom flange. A cradle C, with two long rollers G, is used to support the main jig when it is laid down, so that the latter can be rolled round as required, with a minimum of exertion. Registering pins E fixed near the edge of the lower plate, are arranged to engage with a pair of stops V on the cradle frame, which rapidly and accurately locates the holes under the drill. When the side holes are finished the jig is up-ended for the top holes to be done.

Following the drilling operations, the cylinders are taken to a Gisholt lathe to have the valve chambers bored out, and after that they go to a grinding machine of horizontal type to have the bore finished to gauge.

Taking the above as a fair sample of American practice, we do not consider it shows much advantage on our own, in fact it would seem that there is still much handling. Facing bottom flange, drilling same, boring out, drilling, boring valve chambers, and grinding, makes six separate times in which the casting has to be set up. This certainly does not compare favourably with our turret lathe method, as described above, in which there are only two settings, after which the drilling would only necessitate one more, plus one for grinding, making a total of four, or three if grinding is dispensed with. There seems to be no good reason why the valve chambers should not be dealt with by a vertical drilling machine while the casting is in the drilling jig.

It is, however, hardly fair to draw conclusions from one solitary example, even if it is by one of the leading firms. Two other makers, the Knox Automobile Co. and the Regal Motor Car Co., have kindly furnished some details of their practice, but not sufficient to enable comparison to be made.

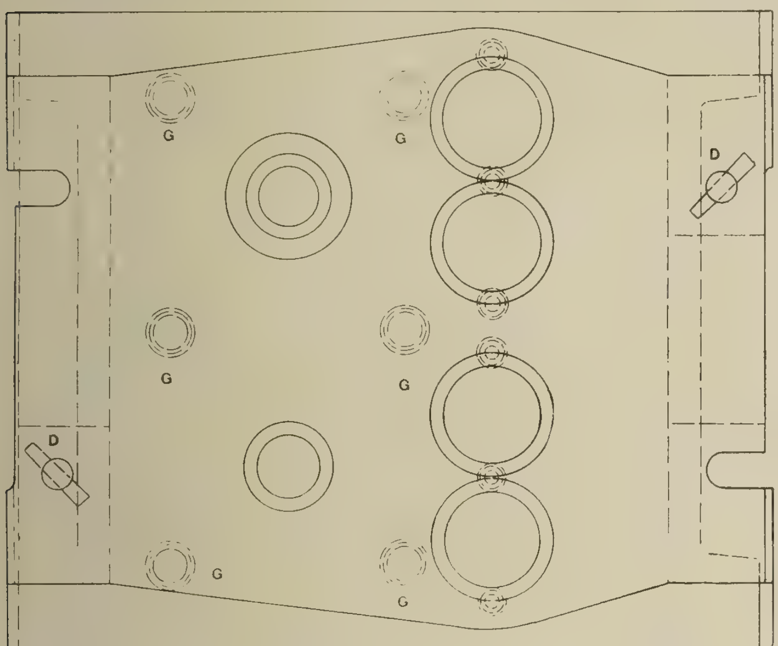


Fig. VIII.

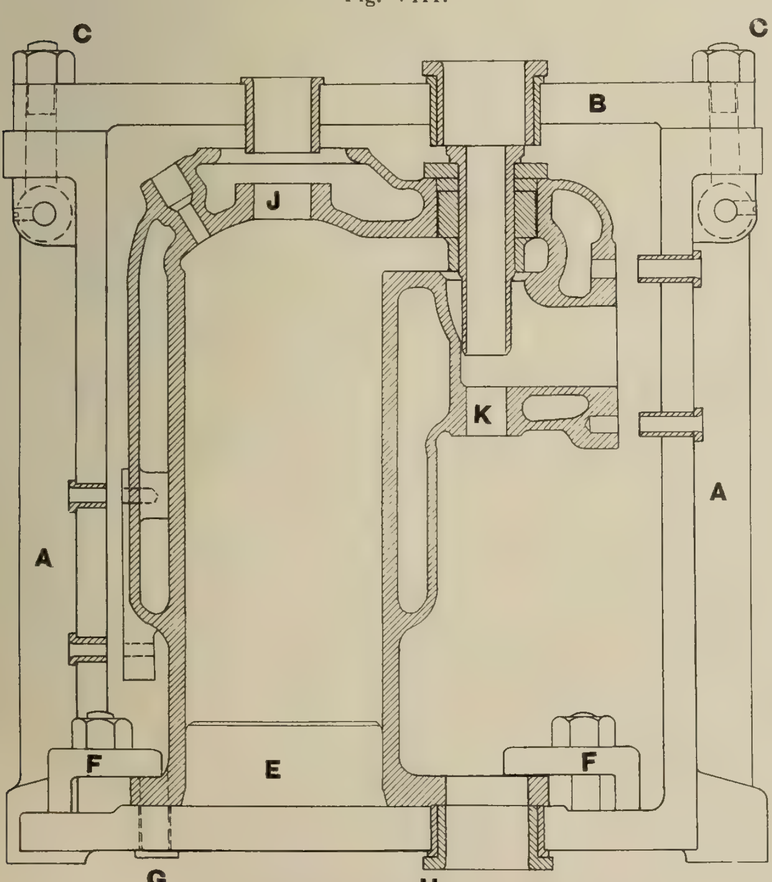


Fig. IX.

The Knox cylinder, as will be seen from Fig. VII., is of rather unusual design, consisting of a straight through cylindrical barrel, with independent cooling jacket and a separately cooled head containing the valves. The company makes three powers of cars, the 40 h.p. and 48 h.p. having singly cast cylinders, whilst the 60 h.p. has them cast in pairs, the bores of the two former being 5ins. and 5½ins. respectively, and the 60 h.p. 5 ins. diameter. It will be noticed that the valves are not in cages, and the construction is simple and clean.

The boring and facing of these cylinders is done on a Gisholt turret lathe, and is obviously a very simple operation; so much so that the company do not find any special jigs necessary. The inside surfaces are all clean and polished, and the machining is of such nature that it must be very cheap to execute.

The Regal Company prefer twin cylinder construction for bores of 4 ins. and over, and all their own cylinders are cast in pairs. They are bored in a Forte-Burte boring machine, and are not ground. Valves are on one side only, and cylinder tops are cast close, i.e., without plug hole. They do not find it necessary to have any special plant for handling their cylinders, nor do they clean up the tops inside. It is a little surprising to learn these facts, because it is difficult to conceive how any modern concern can handle motor car work at all without a most liberal system of jigs and gauges.

Having given an interesting example of an American cylinder drilling jig, it will now be useful to give particulars of a British one, for which we are indebted to the kindness of the Argyll Co. This jig is shown by Figs. VIII. and IX., and consists of a heavy cast iron frame of open box type A, provided with

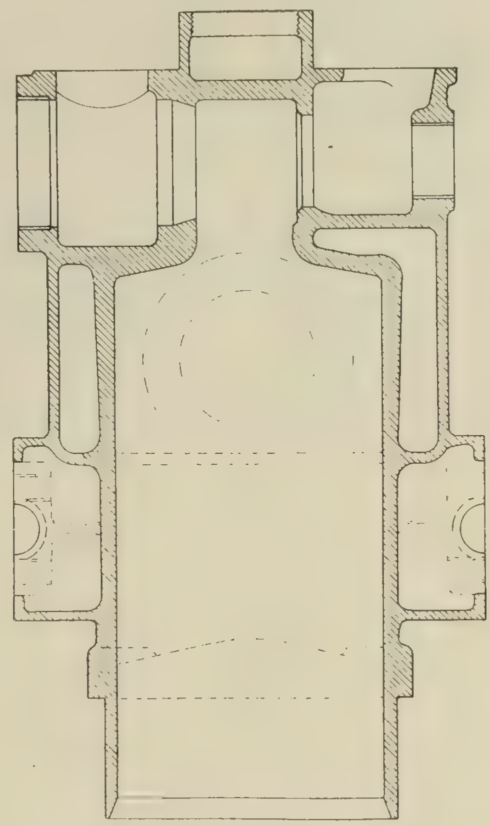


Fig. X.

a cover plate B secured by hinge bolts CC, and two taper position pins DD. The cylinder casting, indicated by dotted lines, is located by means of the hollow plugs E, which fit into the bore, and is secured by four clips and studs F. The frame has projecting lugs, machined off true and square, which form the feet on which it rests.

The first operation, which is done before putting on the cover plate, is to turn the jig upper side downwards, when the holes G for holding-down bolts, and the holes H and stud-holes for the push rod guides, are drilled.

The jig is now turned up and the cover plate B is put on and secured, which brings the combustion head plug J, and the valve chambers under the drill. The joint faces of the latter are machined at the same operation with the top and bottom flange. It will be noticed that there are two steel bushes in the cover-plate for dealing with the valve chamber. The larger or outer one gauges the boring of the cap hole, which is afterwards tapped,

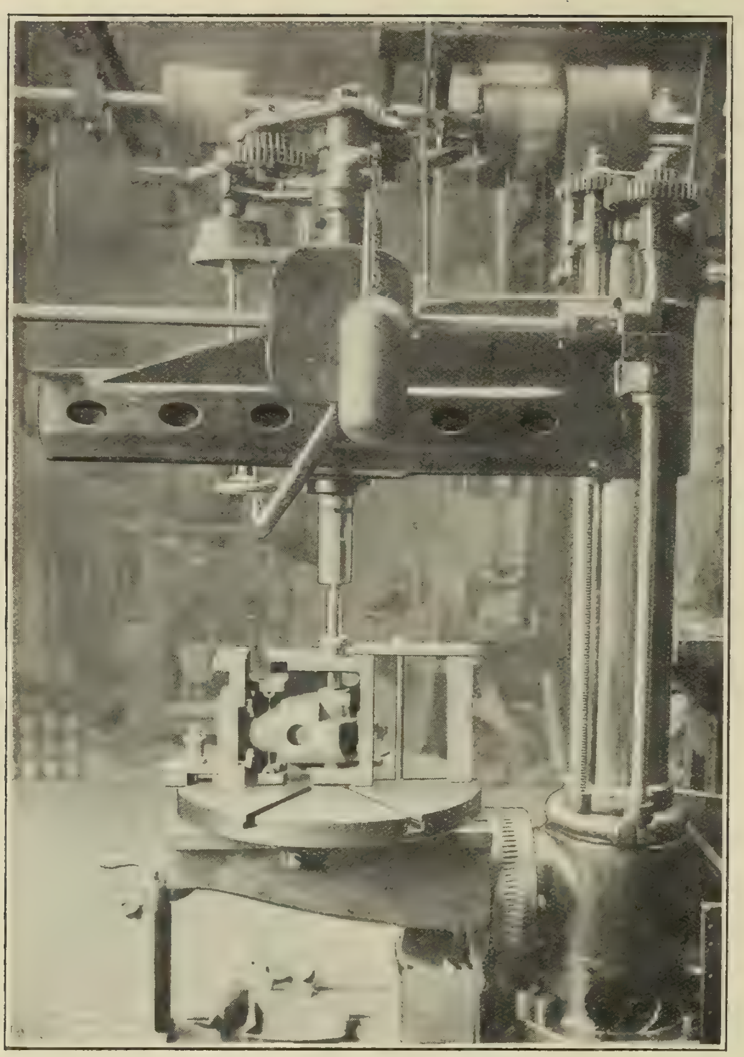


Fig. XI.

the cover being removed for this purpose. The smaller bush guides the tool which trues up the valve seat hole, which can then be mitred through the larger bush. Then the dummy plug and bush are inserted, as shown in the sketch, thus providing for the drilling of the hole K, which takes the valve stem guide. The very long bushing provided to keep this hole K in true alignment is highly important, because any little defect in the casting, or in the homogenity of the metal, or even in the drill itself, causes the drill to run out of centre very easily.

Comparing this jig with that used by the Pierce Arrow Company, it might, at the first glance, appear to be inferior, from not having the circular ends enabling it to be rolled, but it has to be noted that the American cylinder has its valves on opposite sides, and has some end holes to drill, whereas in the Argyll all the holes are on one side, except the two small stud holes for the water inlet. The four clips F in the Argyll jig are perhaps not quite so expeditious as the central screw of the Pierce Arrow for clamping the cylinders into the jig.

There is little practical difference, but a consideration of the two types may suggest fresh ideas, which is, after all, the chief object of an article like this. The Argyll cylinders, of which we show a section in Fig. IX., is a good average sample of present-day design. The top and bottom flanges are faced simultaneously by means of a double-headed miller with inserted tooth heads of fairly large diameter. It will be observed that the inlet flange is set at an angle which necessitates a separate setting up on an angled jig to make this flange offer vertically to the vertical miller in which it is faced. This extra operation is the price that has to be paid for avoiding a sharp bend, which in an induction pipe is doubtless objectionable. Some engine makers avoid this difficulty by putting the flange on the other side and bringing a passage through between the cylinder barrels.

The boring of Argyll cylinders is done on a type of horizontal double-spindle machine of a rather ingenious design, in which the casting is set up on a special saddle which is fed up to the boring bars, the latter having no end movement at all. A simple and easily adjusted trip stop is provided, which throws the feed out of gear at the end of each cut. The feed is at the rate of ½in. per minute, so the time required for each cut is only 8½ to 9 minutes. As originally designed, this boring machine was used for singly cast cylinders, when the sockets of the boring bars were arranged as facing cutters for doing the bottom flange. Argyll cylinders are all finished by grinding, an allowance of .006 ins. being left for this operation. The number of settings required for this system of machining is as follows :—(1) Top and bottom flanges, (2) exhaust and water inlets, (3) angled induction flange, (4) drilling, (5) boring, and (6) grinding.

The processes used for cylinders with valves on both sides is practically the same as the above, but it will be borne in mind that both types can be done more expeditiously, and with a considerable saving in settings, in a turret lathe, as previously described.

The Lanchester system is interesting, because this firm still continue to use only singly cast cylinders. The cylinder, Fig. X., is peculiar in design in that the valves are set horizontally, one on each side, with a combustion chamber between the two valve heads. The bottom flange, projecting nose of barrel and the bore are all machined on a No. 6 Herbert turret lathe, the bore being afterwards finished to gauge by grinding. For drilling a special jig is used, the construction and use of which are clearly shown by the photo Fig. XI., where it is in position under a radial drill, having the valve ports drilled.

Summing up the results of our investigations, we conclude as follows :—

> Monobloc cylinders, from a machine shop point of view, have not much to recommend them, and their use in sizes over 80 mm. bore is not advisable.
>
> Cylinders cast in pairs in all engines up to 5 ins. or 5½ ins. bore are quite convenient for machining and handling, and appear to be relatively more economical all round than single cylinder castings.
>
> Single cylinders made with detachable heads, of the American Knox type would appear to afford superior possibilities in regard to cheapness of machining.
>
> The best makes of cylinders are ground out in the bores, and as, apart from the first cost of the machine, the process is neither very lengthy nor expensive, it is certainly to be recommended.

The American circular plate drilling jig suggests the desirability of following out closely the idea of making all jigs in such a form that their handling may be facilitated with lessened exertion on the part of the workman, bearing in mind that the less resistance the latter has to overcome the greater will be his output. The number of different settings required in connection with the instances we have given of machining pair castings seem to be excessive, and it might be reduced with obvious advantage. Cylinders are far from being the cheapest part of a chassis, and even fairly heavy expenditure on jigs or special tools can soon be recovered if considerable handling is thereby done away with.

THE CUTTING DURABILITY OF TOOL STEELS.

A brief account of some recent experiments.—By Edward G. Herbert.

In two papers read respectively before the Manchester Association of Engineers, in March, 1909, and before the Iron and Steel Institute, in May of the present year, the writer has described at length some investigations into the behaviour of cutting tools. A brief summary of the results obtained, more especially in their bearing on the practical cutting operations of the workshop, is all that will be here attempted.

The investigations were made with the aid of a machine specially designed for testing tool steel, and the essential features of the method of testing are, that the specimen of steel is made into a cutting tool of standard and very simple form, with a straight cutting edge ground at an angle of 70°. The tool is brought to bear against the end of a revolving steel tube of standard dimensions (¾in. diameter and ⅝in. bore) in such a manner that the tube is turned away in very thin shavings, the tube and tool being flooded with water. The durability of the tool is measured by the length of tube it will reduce to shavings before

it attains a standard degree of bluntness, the blunting of the tool being actually measured in thousandths of an inch as the test proceeds. Tests are made at cutting speeds from 20 feet per minute up to the highest speed at which the tool will stand, the results being autographically recorded. The durability of the tool is plotted against the corresponding cutting speeds, and the result is a "speed curve," which shows the characteristics of the steel. Four speed curves are shown in Fig. II., of which A is from a carbon steel, and B, B[1], C from high speed steels.

Several unexpected results have come from these tests, of which the following may be mentioned :—

Every tool that has yet been tested has been found to possess very little durability at the lowest cutting speeds, and a much greater durability at some higher speeds. At 20 feet per minute every tool has been rapidly blunted, and always before it has turned away 2 inches of tube.

At the low speeds a carbon steel tool properly hardened has greater durability than the best high speed steel. At the high speeds great durability is shown by high speed steels properly hardened, and the greatest durability by some of the new vanadium steels.

At first sight these results seem to be directly at variance with the experience of the workshop, where low speeds are generally believed to be conducive to durability of the cutting tool. High speed steel is generally adopted as more durable than carbon steel, and the new vanadium steels were for some time believed to be incapable of cutting at higher speeds than the older tungsten steels, though more durable at ordinary speeds.

Before attempting to co-ordinate these results with each other, and with the facts of workshop experience, it is necessary to refer to some experiments which were made with a view to discovering the cause of the changes of durability displayed by the speed curve. These experiments appear to have established the theory that the governing factor in the durability of a given tool is the temperature to which its edge is raised in cutting. The tool attains its maximum durability when its edge is raised to a certain definite temperature (the particular temperature depending on the quality and condition of the steel). If this temperature is either raised or lowered, the durability of the tool declines.

Bearing in mind this relation between "cutting temperature" and durability, we are in a position to place the results obtained on the testing machine alongside our workshop experience, and to establish some points of correspondence.

The edge of a tool cutting a thin test tube at 20 feet per minute, flooded with water, and taking a shaving only a thousandth of an inch thick, is probably at (or very little above) the temperature of the atmosphere. What have we to correspond with this in the workshop? Take a fine finishing cut and drop water at the point of cutting. Steam rises, showing that the tool point must be considerably above air temperature. In order to obtain the corresponding condition we must place the edge of the tool lightly against the surface of the work revolving at a very slow speed. If the pressure is light enough the tool will rub without cutting and without becoming sensibly heated, and every workman knows that under these conditions the tool will be worn away with great rapidity. It will become rapidly blunted, as on the testing machine. Here is our first point of correspondence.

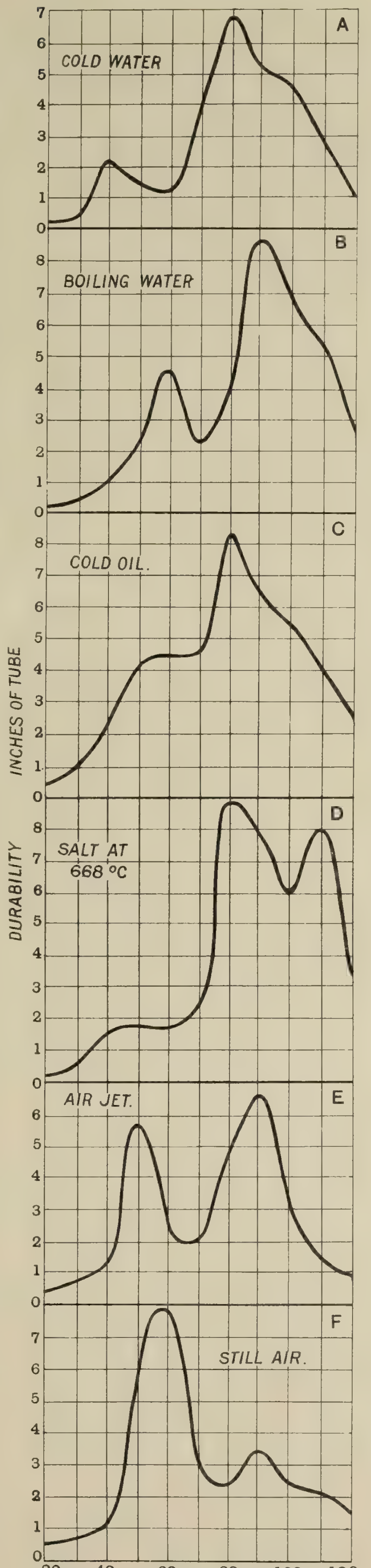


Fig. I.

Ever since the introduction of high speed steel, it has been recognized as generally unsuitable for light finishing cuts *at ordinary speeds*. For such work carbon steel has been found more durable. But such light cuts do not generate much heat. They may correspond to the testing machine speeds of 30 or 40 feet per minute, and it is seen from the curves in Fig. II. that at such speeds the carbon steel is more durable. But it appears also from Fig. II. that the same light cut, if taken at a high enough speed, will render the high speed steel more durable than the carbon steel. This important fact has now been amply confirmed in workshop practice. A high speed steel tool will keep its edge on a finishing cut *if the speed is high enough*, but not otherwise. It has likewise been found that the new vanadium steels are capable of cutting at much higher speeds than the older steels, thus again confirming the results obtained on the testing machine.

If it be a fact that the low durability of high speed steel on finishing cuts is due to the lowness of the temperature to which the edge is raised in cutting, then it would seem to follow that such tools will be more durable when cutting dry than when flooded with water, since in the former case the temperature is higher. This has been found to be the case, and is a fact of some practical importance. Bearing in mind the relation between cutting temperature and durability, it is obviously immaterial whether we raise the temperature of the tool to the point of maximum durability by a high speed or by a heavy cut, since both have the same effect on the cutting temperature. High speed steels attain their maximum durability at high temperatures, and this is why they are specially durable under the heavy cuts, which generate a large amount of heat. It is, of course, possible for the temperature to be too high; above a certain temperature the durability falls rapidly, as shown by the speed curve, and in such a case it is advantageous to reduce the temperature by means of a jet of water or oil.

It has long been known that there is an intimate connection between cutting speed and weight of cut, as regards their effect on durability, and it has recently been shown that this relation may be expressed by the "cube law of cutting speeds." For constant durability of the cutting tool, the speed must vary inversely, as the cube root of the product of thickness of shaving by area of cut.

$$S_2 = S_1 \sqrt[3]{\frac{t_1 a_1}{t_2 a_2}}$$

One of the most interesting further remarkable facts brought to light by the tool steel testing machine is that tool steels of all the classes in general use, give double-peaked speed curves under certain conditions, that is to say, the durability reaches maximum values at two distinct cutting speeds, and is lower at all intermediate speeds. Examples of double peaked curves are given in Fig. I. The cause of this peculiar phenomenon is not yet fully understood, but it has been found that an alteration in the method of hardening the steel usually

changes the form of the speed curve, so that the same steel may give either two distinct peaks, or a single peak at either a high or a low speed, according to the method adopted in hardening it. The curves in Fig. I. were all obtained from tools cut from the same bar of high speed steel, heated to the same temperature, and cooled in different media as indicated on the diagrams. The vertical lines giving cutting speed in feet per minute.

Several important practical conclusions follow from these results. In the first place it is obvious that the hardening process which renders a tool most durable for heavy or high speed work may, and usually does, render it less durable under light cuts at low and moderate speeds. Perfectly distinct hardening processes should be adopted for the two classes of work, but it must not be assumed that the curves in Fig. I. are a correct guide for the hardening of all high speed steels, since the sequence of changes varies according to their composition.

Again, it is obviously unsafe to assume that a tool which rapidly loses its edge is incorrectly hardened, or that it is overworked. The tool may be cutting at a temperature corresponding to the depression between the two peaks, in which case an increase of speed will improve its durability. This has often been confirmed in practice.

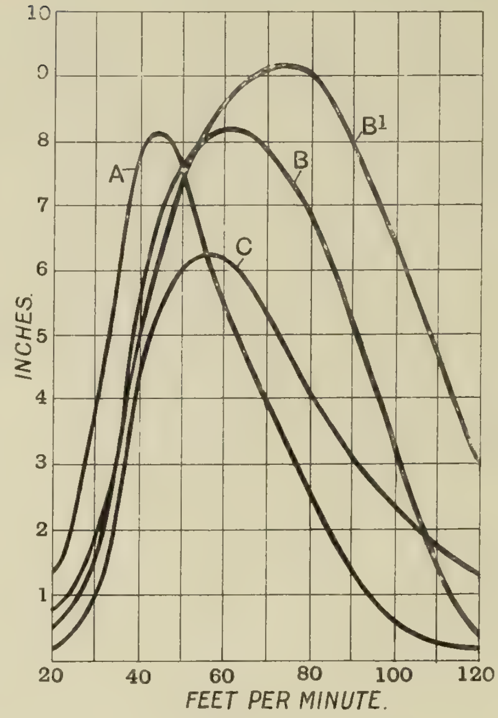


Fig. II.

A further conclusion is that it is not possible to ascertain the quality of a steel by means of a test at any one cut and speed. Such a test may show which is the most suitable steel for that particular combination, but it will not necessarily show which is the best steel, and it would be incorrect to assume that the steel which is most durable under a particular set of cutting conditions is the best for general work. This can be seen by comparing the durability of tools D and F at 60 and at 110 feet per minute. Tool D was hardened by being plunged at a white heat into a bath of molten salt, and this method of hardening high speed steel has generally been found, on the testing machine and in practice, to give the tools great durability under heavy cuts and high speeds.

Such are some of the practical conclusions which follow from the present series of experiments, but these experiments must be regarded as only a commencement of an investigation into the cutting properties of tool steel. Many unexplored paths of enquiry already suggest themselves, and it cannot be doubted that further light will be thrown on the important problems connected with the treatment and use of the many different varieties of tool steels at present adopted for engineering machine work.

THEORY AND PRACTICE.

A comparison of their relative value in the past and at present.

By Lawrence H. Pomeroy, M.I.A.E.

WHEN the history of automobile design comes to be written, the study of the influences which have brought it to its present state will make an interesting chronicle. Kindred branches of mechanical science have developed under pressure from users who have appreciated the various machines and contrivances offered by manufacturers, and who have offered skilled criticism of defects in design. This action and re-action between manufacturers and users has been very largely free from fads and superstitions, and to this the progress that has been made is due. In the design of the automobile, however, it is only within very recent years that rational methods have been applied at all. The locomotive has been steadily perfected, and is a striking example of construction which remains practically the same now as it was at first, so far as its elements are concerned. Redundant parts have scarcely ever existed in it, but the automobile of the present differs in that modern designs are characterised by the elimination of parts and the simplification of construction, whilst the rate of improvement is apparently increasing as time goes on. This is due to the fact that it has only recently been recognised that automobile design is just as susceptible to the application of analytical methods as the design of steam turbines and dynamos.

It has often been urged as evidence against the value of the technical data investigated in engineering laboratories that as the amount of such information available ten years ago was, for the purpose under consideration, and except as regards a few new steels, nearly as extensive as it is to-day, the improvements which have been made must therefore be due to experience alone. It becomes, for this reason, a matter of interest to discuss the relative importance of the work of designers who may be termed empirical on the one hand and scientific on the other, in bringing about a state of affairs which, though far from being perfect, is at least comparable with any other branch of engineering so far as effectiveness and reliability are concerned. In fact in this connection, it has often struck the writer that the petty breakdowns of a modern workshop equipment would surprise those clients who expect a ton of perfection for about £300.

In the earlier days of automobilism the design of motor cars was usually prescribed by those whose knowledge of engineering was by no means in keeping with their enthusiasm for a new pastime. The lessons of a century of experience in steam engineering were deliberately ignored, even by men who were fully acquainted with them. Materials were used which no sane mechanic would think of advising; in fact automobile design was assumed to be wrapped in mystery to such an extent that it formed a class of construction to which the laws of Nature could not apply. In consequence, factors of safety were mentioned only to be ridiculed, the nature and magnitude of working stresses were not considered, and the application of the principles of elementary physics were regarded as a kind of sacrilege.

The arbiter of design was "the road," and the tacit assumption that this criterion necessarily excluded all others is, in the opinion of the writer, the chief cause of the backwardness of the industry which was noticeable in this country until a very few years ago. Some authorities on design graduated from the cycle track minus any mechanical qualifications. Under these conditions then, with improvement occurring as the result of trial and error (mostly the latter) automobile design grew up with more than its fair share of the ills of infancy. Early in this century, however, there sprung up a remarkable combination. Engineers who were not quacks began to realise that the problems of the car were worthy of solution, and incidentally might also bring certain emoluments which the slackness of business in other directions rendered highly desirable. A fresh strain being thus introduced, improvements in design were rapidly made. Simultaneously the cycle trade with its manufacturing facilities also saw in the motor business something in the nature of a promised land. However little of the milk and honey has come to them, the fact remains that the simplification of parts essential to quantity manufacture brought with it a far-reaching and beneficial effect. The cheap car was produced and other manufacturers realised that "the more the bits the more the cost." Cars were re-designed to comply with the new requirements, and to the surprise of their makers showed that, although so many parts were thrown out, the last state of the car was considerably better than the first.

Another stimulus must not be overlooked, namely the competitions which were so largely in vogue a few years ago. The automobile, though reliable, was heavy and uneconomical, and speed was attained by brute force rather than by

brains. The conditions laid down by the Tourist Trophy races forced the development of a car with small cylinder capacity and light weight, combined with economy in fuel consumption. These cars were the progenitors of the modern car with a small high efficiency engine which has practically beaten the "roaring forties" out of the market. The growth of speed and hill-climbing competitions rendered some form of handicapping necessary to correlate the performances of different makes and classes of cars, and this subject has probably been productive of more argument, from a theoretical point of view, than any other connected with the industry. Even now a committee is sitting to investigate the question of horse power rating. University professors have contributed more than their quota to this and other subjects which in reality form the very basis of automobile design, and it is not an illogical conclusion to state that, if the horse power rating of engines is still a matter which is obscure, then engine design has so far been based on nothing less than guess-work. One has only to study the papers read before the various engineering institutions to realise their academical nature. A position of great interest is thus set up. At one end of the scale we have the empiricist plodding along trying one combination after another, with little reason and often less result; at the other, the mathematical physicists giving the weight of their authority to support propositions that are often practically untenable. An interesting example of the latter is to be found in the many horse power formula which include as a factor only a fractional power of the stroke of the engine. The reasoning which has deduced these formula is based upon the idea that the limit of piston speed is a function of the stresses set up by the inertia of the reciprocating parts, and much good type has been set up to demonstrate this by very high-class mathematics.

A simple calculation will show that in order to exceed the safe stresses which may be imposed upon the reciprocating parts of engines *such as are used in automobiles*, the speeds of rotation would need to be far higher than those of actual practice. This being so, the index of the stroke factor naturally becomes unity. The cubic capacity of the cylinders is then the criterion of relative horse power and this harmonises with the well-known methods for determining the horse power of steam engines.

The present R.A.C. rating rule is an example of the use of purely empiric methods which is more pernicious than ever. It will be seen that between the empiricist and the mathematician the nature of rational design is apt to be missed, but fortunately there has sprung up within very recent years a school of designers mathematicians enough to know how liable the physicist is to find and select facts which support and verify his theories, and with sufficient scientific training, grafted on to their practical experience, to enable them to apply analytical methods to their work. Many of these men have gone to quite a lot of pains to examine the first principles underlying their business, and to weed out the superstition on the one side, whilst steering clear of the misleading mathematics on the other. In many cases they have had to write down very largely the experience of their seniors and start again on a clean sheet.

It has been realised that the broad principles of mechanics can be applied to almost every problem which arises in automobile construction. The principles underlying their solution have been established, and the constants required to rationalise these principles have been determined by legitimate and scientific experiments. Throughout, the motto of doing nothing without a reason has been adhered to. When failure has resulted, the previous work has been re-examined to find the reason for the failure, before setting out afresh. The consequence is that automobile design has been raised to a standard undreamt of a few years back. The steering wheel engineer is still estimated at his proper value, which is a very high one indeed when his suggestions have been thoroughly thrashed out, and their value verified.

The near future will probably see chassis and engine design revolutionised, as the principles now established point the way to further simplification. The spirit of true efficiency is pervading the whole industry, and can only result in the lasting benefit of those who are willing to accept it and the extinction of those who are not. It is safe to say that the dark age, when so-called theory and so-called practice indulged in mutual recrimination, has now passed, and that in the future no feature of construction which is not mechanically sound will be allowed to exist.

THE GNOME ENGINE.

By Robert W. A. Brewer, A.M.I.C.E., M.I.M.E., M.I.A.E., F.S.E.,

THE subject of this article is one of the most interesting engineering achievements of modern times.

The engine has been remarkable both for its successful working during lengthy periods in the propulsion of aeroplanes, and for its daring design and construction, and, though so radically different from all other successful aerial engines, it runs with remarkable freedom from breakdown or temporary stoppages—when the delicacy of many of the working parts and the adverse conditions under which some of them have to work, is considered.

One may ask why it is that complication in construction, such as results from the employment of a large number of cylinders, is not detrimental to a motor of the Gnome type. The chief answer is that by such an arrangement of cylinders an attempt is made to reduce to a minimum such masses of the engine as are inert, and consequently to employ the useful masses, such as the pistons and cylinders, for a second purpose, viz., as storages for energy. There is a second reason for disposing the cylinders in star shape, for this construction facilitates the balance of the engine, both longitudinally and rotarily, without the addition of balance weights.

A rotary engine such as the Gnome requires a minimum of appliances for cylinder radiation, as it will be seen that the cooling effect of the air currents generated by the revolving cylinders, and by the propeller, bears some proportion to the heat generated by the combustion of the fuel in the cylinders themselves. This is made apparent more forcibly when we consider that at a speed of rotation of 1,000 revolutions per minute of the engine the mean velocity of the cooling ribs through the air in a rotary direction is 80 miles per hour, and at this speed effectual cooling would take place, even though the ribs were eliminated.

The writer has carefully noted the temperature of Gnome cylinders after the engine had been in flight for various lengths of time, and in cold weather the cylinder walls have only been at a temperature of a few degrees (say 30° F.) above that of the atmosphere *immediately* after the engine had stopped. The cylinder temperature afterwards rises on account of conduction of heat from other portions of the engine. Even on a summer afternoon the cylinder wall temperature seldom rises above about 150° Fah., or rather too hot to bear the hand upon at the hottest part.

The advantages of a rotary type of engine are too well known and appreciated to be dealt with here, but a few of the difficulties in the production of such a design will be briefly enumerated to enable us to appreciate the clever way some of them have been overcome by the designer of the Gnome.

(a) The engine must be in perfect static balance, so that when fixed on a horizontal axis there will be no tendency for it to rotate in one direction or the other, whatever the position in which it may be placed.

(b) The valves must be balanced so as to counteract centrifugal force, that their opening or closing will not be affected in any way whatever the speed of rotation of the motor may be. The engine must be capable of being started slowly by hand, so that the inlet valves, if of the automatic type, must be fitted with light springs.

(c). The group of pistons must be in running balance in such a manner that the sum of all forces acting upon their masses will be zero.

(d) A suitable arrangement must be made for the admission of the working fluid to the cylinders and for the escape of the burnt gases therefrom. The engine must therefore be self-contained in this respect in order to eliminate piping.

(e) The ignition and its distribution must be certain, and free from risk of breakdown owing to the action of centrifugal force upon rotating wires or connections.

(f) Provision must be made against any possible distortion of the cylinders due to unequal temperature of the walls affecting the tightness of the pistons in the cylinders.

(g) The cylinders must be securely fixed to the crank chamber to prevent

risk of their coming adrift under the combined force due to the explosions and centrifugal action.

(h) The engine must be effectively lubricated in an efficient manner, and in such a way that each piston will receive an equal quantity of lubricant.

(j) The valves and working parts of the engine must be accessible, as in an engine for aviation purposes, upon which the life of the aviator depends, it is essential that all working parts should be kept clean and free in action.

It is not to be supposed that all such conditions can be carried out perfectly in any one motor, but the Gnome engine is an excellent attempt.

The first point that strikes the observer is that aluminium is entirely eliminated from the construction, and lightness is obtained solely by the use of nickel steel, very finely proportioned throughout the whole engine. The cylinders are turned out of a solid steel ingot, as it is only by such means that absolute uniformity of material and thickness can be obtained. The thickness of metal in the cylinder walls is 1.5 mm., and strength is secured by the ribs, which vary in diameter being smaller as the region of the greatest pressure is departed from. The removable end of the cylinder contains the exhaust valve seat, which is screwed into place by means of a special box spanner. A very neat arrangement will then be observed for the removal of the inlet valve seat from the piston head.

Round the aperture left by the removal of the exhaust valve seat a number of slots are situated, and also a further ring of slots will be observed on the piston outside the inlet valve seat. When the piston is brought to the outward end of its stroke a special tubular spanner is inserted into the cylinder, having two rings or keys, one at each end, forged upon it. These keys fit into the slots in the piston at the inner end, and the outer ring of keys fit into the slots in the cylinder. It will be seen that the piston is thus prevented from any tendency to rotate.

A second tubular spanner is now placed inside the fixing spanner, and its end engages with the inlet valve seat. The spanner can be turned with impunity in unscrewing this valve seat without risk of placing any twisting stresses upon the connecting rod. The inlet valve seat and the valve itself can thus be drawn.

The cylinders are fixed to the crank chamber in a manner as neat as it is original. On the trunk end of the piston a groove is turned similar to that provided in a piston for the reception of a piston ring. Each cylinder is turned perfectly cylindrical outside, and the seven cylinders, comprising one 50 h.p. unit, are pressed into circular holes bored out of the crank chamber. When a cylinder is in position a contracting sring ring is fitted tightly into the groove, which has now just entered the inside of the crank chamber, the ring bearing partly in the groove and partly on a face turned for the purpose inside the crank chamber. Seven holes are drilled right through the crank chamber, in a direction parallel to the crank-shaft, one between each cylinder seat, and when all the cylinders are in position long pins are inserted in these holes. The diameter of the pins is such that each pin bears against two neighbouring cylinder retaining rings, and holds them firmly in position, prevent them from rotating.

A key is fitted at the end of each cylinder trunk, engaging in a seat in the crank chamber in order to counteract any tendency to rotate on the part of the cylinders themselves. The cylinders are bored out to 110 mm. diameter, and the piston stroke is 120 mm., the normal rate of revolutions being about 1,000 per minute, though this is sometimes rated as 1,200.

The pistons have been fitted sometimes with one ring and sometimes with three, of a special form, each consisting of an L-shaped gun-metal ring, reinforced with a steel ring of the ordinary Ramsbottom type behind it. The gun-metal rings are very thin, and their action is similar to that of a pump leather, while their object is to conform to any distortion of the cylinder in working.

A duplex ring of such a type is apt to accumulate carbon, and become more or less solid, resulting in considerable loss of compression, and it is noticeable that after working for some hours the compression is considerably below the normal.

Mention has already been made of a part of the inlet valve seat, viz., that which is removable through the cylinder head. The second portion of the inlet valve seat is made in one piece with the eye for attachment to the gudgeon pin and small end of the connecting rod, which is clearly shown in the accompanying drawing. The method of balancing the inlet valve is also of particular interest, as is the lightness of the parts and the shape of the valve-retaining spring. This spring has four leaves at the centre, and its ends are curled over in such a manner that each end embraces one end of the small levers balancing the weight of the inlet valve. The inlet valve is drilled a part of the way up its stem to reduce weight, and the spring clips easily over the valve stem.

This arrangement, though very neat, is scarcely a happy one, as should anything break considerable damage might result, though the writer's experience may be fortunate in this respect, as he has not encountered any accident due to failure of the inlet valves or springs. The inlet valve mechanism is lubricated by splash from the crank chamber, but the springs are working under adverse condi-

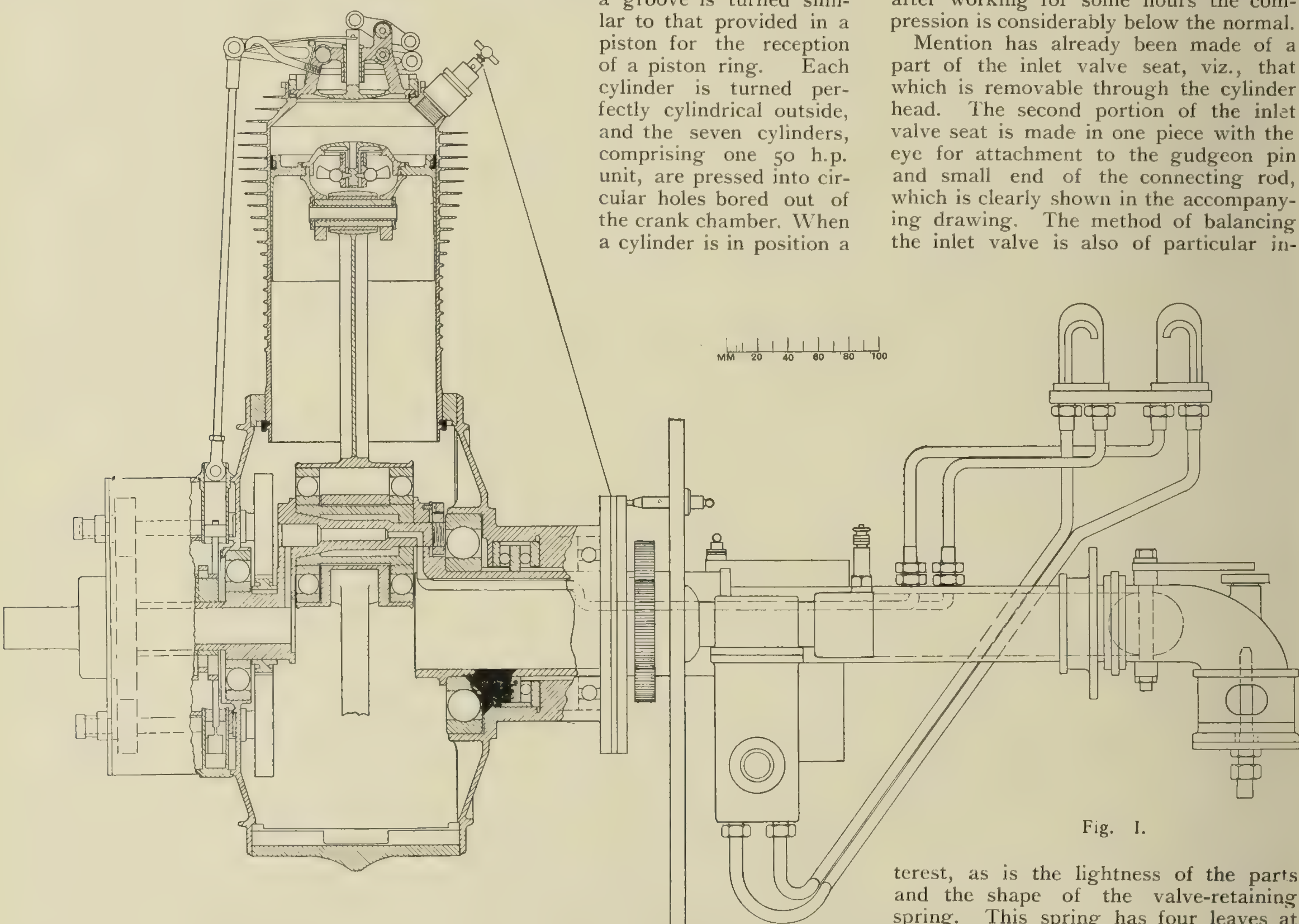


Fig. 1.

tions of high temperature, and there are a number of small parts, each of which might possibly fail or come adrift.

The exhaust valves are somewhat similar as regards their balanced lever gear, but the springs of the four-leaved type are of a different shape. The arrangement of the springs is shown clearly in Fig. I. These springs are in direct line with the escaping burnt gases, but the probability is that they never attain a high temperature owing to the rapid rotation of the engine in cool air. The valves are operated by tension rods of small diameter actuated by female cam rings in a very interesting manner. At the end of the crank chamber remote from the propeller is a cylindrical casing, containing seven thin steel rings, female cam shaped inside. Each ring is formed with a pair of projecting rods situated across a diameter, one being merely a guide rod, and the other, slightly longer, terminating in a knuckle eye. This eye runs inside a guide, and at its outmost position the pin just appears outside the gun-metal guide. From this eye an adjustable rod is connected to a pair of levers of the valve-operating mechanism, the motion of which is reversed by a second pair of balanced levers actuating the valve. Thus there are a pair of long levers and a pair of short ones of less than half their length.

These seven female cams are arranged radially and equidistantly inside the cylindrical casing, are all alike, and are guided by seven pairs of gun-metal guides, whose planes vary exactly as the thickness of the cams, the total difference between the seven being about 50 mm. It will thus be seen that the valve-actuating rods are not all at quite the same angle.

A single cam of the ordinary type and of a length equal to the thickness of the whole seven female cams rotates inside them, and as the hump on the male cam engages the flat on any female cam, the latter is displaced a distance of 10 mm. The male cam is rotated at half-engine speed primarily by a gear wheel on the crank-shaft operating a pair of secondary gear wheels, these transferring their motion to a single gear wheel running freely on the crank-shaft, and to which the male cam is attached. The pair of secondary gear wheels run on a pair of pins, of which the ends pass through the end of the crank chamber covering the cylindrical cam casing.

Mention has been made of the crank chamber of cylindrical steel, the cam end of which is enclosed by an end plate and cylindrical cam casing. The propeller end is plain, with a cylindrical extenion piece having two keys attached to it. The diameter of this plain part on the 50 h.p. engine is 130 mms., and it allows of a propeller 70 mms. thick at the boss, the diameter of the propeller boss outside being 300 mms.

End plates retain the propeller in position, and immediately adjacent to them a toothed wheel is keyed to the crank-shaft, which engages a pair of smaller toothed wheels of similar dimensions operating the magneto and oil pump.

The gear ratio between these wheels is 7 to 4, that is, the magneto runs faster than the engine in the above proportion.

Of the two end plates retaining the propeller, one is a distributer plate for the ignition, and has seven gun-metal segments let into it, connection being made to seven wires passing out between insulated surfaces and between the two end plates.

These surfaces are about 1.5 mm. apart, to allow for the wires to pass. The high-tension terminal of the magneto is connected by means of a bare brass wire to a carbon brush held in position by another end plate at the magneto side of the operating gear wheel. This plate does not therefore move, as it is attached to the base plate, which retains the crank-shaft, and fixes the latter to the engine fuselage. As the engine revolves each segment in turn comes opposite to the brush, and is thus fed with a high-tension electric current. Bare brass wires of small gauge are attached to the plugs, but not firmly, as a firm fixing would result in fracture in working.

The odd number of cylinders permits of equal angular intervals between firing, thus the sequence of 1, 3, 5, 7, 2, 4, 6 enables the engine to make two revolutions with a perfect firing balance.

The crank-shaft, which is stationary, is held in ball bearings fixed in the end plates of the engine, and the shaft being hollow, forms the passage for the explosive mixture between the carburettor and the crank chamber. Advantage is also taken of the hollow shaft to convey the two copper lubricating oil pipes to the engine crank chamber.

The seven connecting rods all operate upon a single crank pin, and are all in the same plane. One rod is made in a single piece with two large drilled rings of L section, there being six pairs of holes, situated at $51\frac{1}{2}^{\circ}$ apart, to take the pins for attachment of the large ends of the six remaining connecting rods. It is necessary to make a rigid attachment between one connecting rod and this large cage in order to locate the latter.

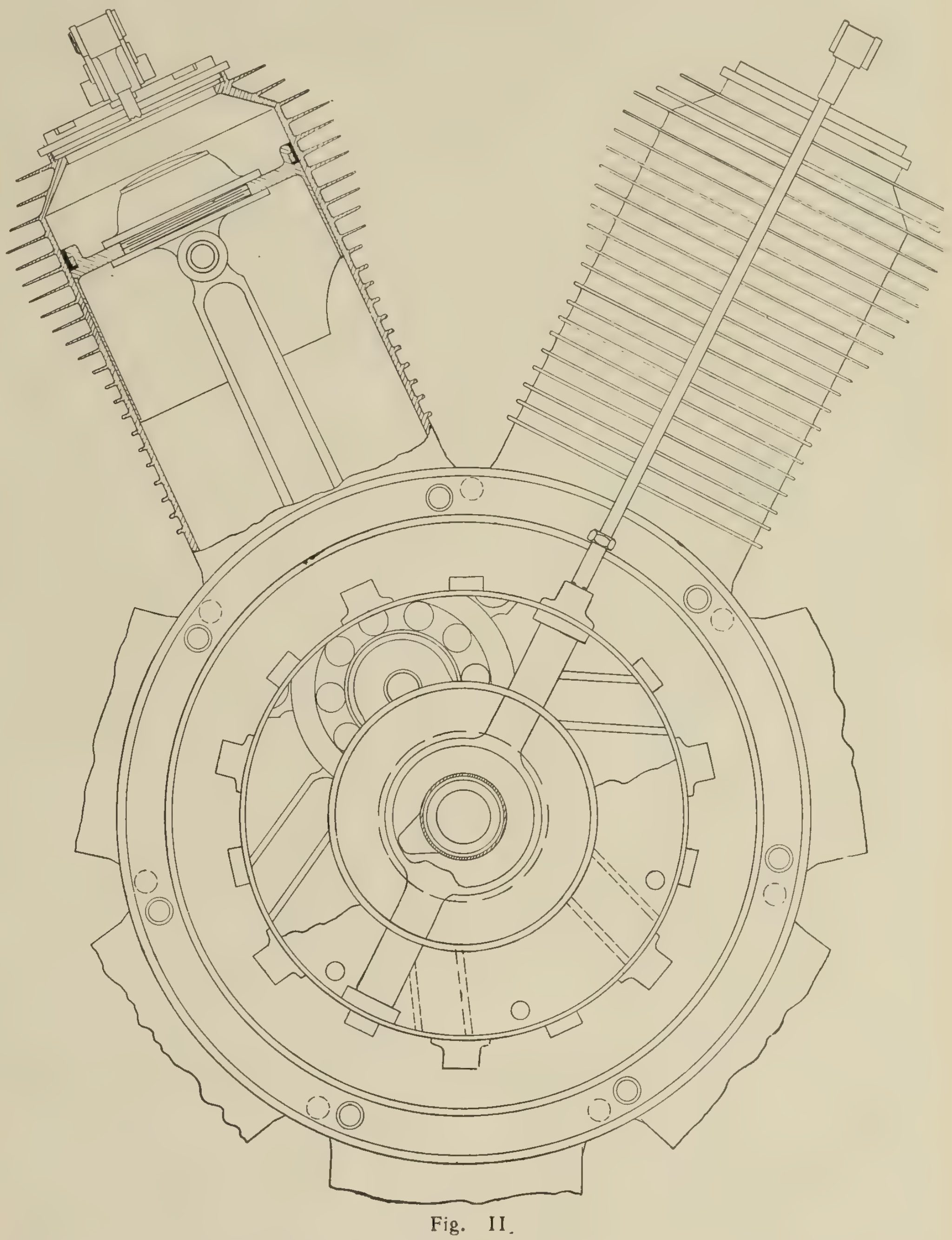


Fig. II.

This cage is fitted with two ball races, one on either side, and these are fixed on the crank pin, taking the thrust of the seven connecting rods. When the connecting rods are assembled in place the six short rods are fitted in position in the cage, and a hollow steel pin is placed through each rod and its corresponding pair of holes in the cage.

It is obvious that in such an arrangement the angularity of the six short rods is slightly greater than that of the longer rod, but this makes no apparent difference in working. The rods are of H section steel, as shown, and are well milled

out at their sides. The oil pump is of the two-cylinder type, working in conjunction with a distributor and without valves, this rendering its action certain. Oil gravitates from a tank, and is delivered to a pair of sight feeds, which can be placed at any convenient position. The flow of oil can here be regulated to give about one drop each per fourteen engine revolutions.

Pure castor oil alone is used by the writer, and the consumption, though stated by the makers at two litres an hour, is generally about seven, and the consumption of petrol about eighteen litres an hour, the makers' statement being 300 to 350 grammes per H.P. hour, which works out at about fourteen litres per hour.

The carburettor is of the single-jet type, and without float chamber, regulation of flow being controlled by a screw-down valve placed near the aviator's seat. A plain throttle is fitted between the carburettor and the crank-shaft end, the admission of air being directly across the top of the jet orifice. A rotating air shutter is fitted, so that the aviator can manipulate either the air or petrol supply when in flight.

The brake horse-power of the nominal 50 h.p. Gnome on the test bench is from 45 to 47 h.p. at 1,000 revolutions per minute, at which speed about 5 h.p. is consumed in overcoming the air resistance to the rotation of the cylinders. The engine weighs 76 kilogrammes, or 167 pounds, which is equal to 3.55 pounds per horse-power.

As fitted to a monoplane of the Bleriot type, the rate of engine revolutions can conveniently be reduced to about 500 or 700 per minute, at which speed ample power is developed for all ordinary purposes, and it is possible to throttle the engine down to 200 revolutions per minute, at which speed the consumption of oil and petrol is greatly reduced.

The weakest point of the Gnome engine is its lubrication system and its enormous consumption of lubricating oil. The oil passes through the engine and out of the exhaust valves, being blown either into a brass catcher ring, or all over the aeroplane.

As fitted to a biplane, this leakage of oil becomes a nuisance, as it accumulates on the tail surfaces, thus adding to their weight, and when stationary the oil is blown by the propeller wake all over anybody who happens to be near.

The Gnome motor should preferably be directly connected to the propeller, and fixed with its shaft on a horizontal axis, the thrust of the propeller being taken by a ball race provided for the purpose. It may, however, be fitted on a vertical axis, and the drive transmitted through a shaft arrangement.

CORRESPONDENCE.

No responsibility is accepted by the Editor for the opinions of correspondents, and the right is reserved to omit any portion of a letter if it is considered desirable to do so. Letters to be inserted with a nom de plume must be accompanied by the name and address of the writer.

ARCHED AXLES.

Sir,—Having read the first issue of the *Automobile Engineer* with great enjoyment, I feel I must write and express my wishes for its welfare, and I hope the *Automobile Engineer* will meet the measure of success it deserves, for if the coming issues are up to the standard set by No. I. it deserves much.

I was exceedingly pleased to read your remarks upon "Arched back axles," it surprises one, indeed, to realise the number of clever engineers that have been led away by the fallacious arguments for the arched axle and the dished wheel. Such ideas are continually cropping up in automobile work. Only a few weeks ago I saw by the papers that some gentleman proposed to place the sparking plug of an engine in a kind of pepper-box, and suggested thereby that a large increase of power was to be obtained. This kind of idea does not represent a waste of much brain energy, but it does usually represent a scandalous waste of the coin of the realm.

One suggestion, if you will allow me to make it, Sir. Do, please, give all measurements in metric, and not in decimal parts of an inch? This would, I feel sure, be the means of saving considerable brain effort upon the part of your readers.—Wishing you every success,

ARTHUR J. TOBY.

TESTING SYSTEMS.

Sir,—We have read with interest an article appearing in the May issue of *The Automobile Engineer*, entitled "Testing Systems for Engines and Chassis." In the course of his remarks, the writer gives it as his opinion that there is a real need for an automatic recording instrument which will give a power and speed curve direct. As manufacturers of the "Acer" Portable Dynamometer we naturally endorse this statement, as also the conclusion that the use of such an instrument will save a large proportion of the wages bill in shops where large numbers of engines are tested. We enclose a specimen chart showing the power curve, and it will be noticed that not only is the B.H.P. ascertainable at a glance, but that also the speed in feet per second and the pull in lbs. may be similarly observed.

An electric installation is no doubt, for extended trials and under certain conditions, an almost ideal method of testing B.H.P., but where portability and rapidity of attachment is concerned we think all motor engineers who have experience in tuning up will appreciate the introduction of a dynamometer such as the "Acer."

For ACER LTD.,
A. E. S. CRAIG, General Manager.

CARBURETTORS FOR PETROLEUM SPIRIT.

Sir,—Mr. Brewer appears to have had in mind the old carburettors of five or six years ago when writing this article, in which the negative pressure was allowed to fall to vanishing point at the slowest speed of the engine. Within the last year or two the demand for a carburettor capable of supplying an engine running at very low speed with a suitable mixture has caused an important change to be made in the method of inducing the supply of petrol through the jet, and incidentally the same change has improved the efficiency of the modern carburettor wonderfully.

These improved conditions have been brought about mainly by increasing the pressure ratio between the inside and outside of the carburettor at low engine speeds and by making use of the direct effect of this pressure to induce the flow of petrol instead of relying partially or wholly upon the inductive effect of a current of air passing around the tip of the jet, thus relieving it of the pressure of the atmosphere, or making use of vacuum formed at the point of greatest velocity in a double conical choke tube (Venturi Tube). When it is considered that a partial vacuum has to be created within the carburettor in any case to induce a flow of air through the choke tube it is difficult to see what advantage was expected to be derived from either of these methods.

Mr. Brewer speaks in many cases of the velocity of the air around the jet inducing the flow of petrol. While this may be correct in the case of the G.A. carburettor he mentions, which, so far as my experience goes, possessed all the undesirable features common to carburettors of five or six years ago and is quite out of the question for use with a modern engine capable of a speed variation of, say, 10 to 1, I think the term is rather misleading when applied to the majority of carburettors. I have tested many carburettors by forcing air through them instead of inducing it to enter by means of a vacuum. The results in almost every case have confirmed my views. In the White and Poppe carburettor although the velocity of the air entering at slow speeds is very great and therefore the partial vacuum in the carburettor high, the velocity of the air past the tip of the jet may be quite low as the jet is placed in the centre of a comparative large annular space. Practically the same conditions prevail in the Trier and Martin. Here the velocity of air is adjustable at the point of entering the carburettor, and although this adjustment alters the partial vacuum within the carburettor the velocity past the jets remains the same. A spring-controlled valve is also fitted at the back of the jets. This, to my mind, would be better placed in the more orthodox position between the choke tube and the throttle.

The carburettor problem is greatly simplified if the partial vacuum within the carburettor is maintained practically constant at all engine speeds by the now generally accepted method of a spring-loaded air valve arranged so as to admit air between the throttle and the choke tube. An attempt has been made with some carburettors to get the same good results as an air valve by arranging an air port operated in conjunction with the throttle, but as the auxiliary air opening should manifestedly be controlled by the speed of the engine and not by the position of the throttle this plan introduces some very undesirable features, which I have previously gone into in a contemporary. The Napier hydraulic air control is the only system I know which competes satisfactorily with the suction-operated air valve, but it lacks the simplicity of the latter.

Mr. Brewer mentions the air regulator as being an admirable device, with which remark I quite agree, but when he adds "when the slots are proportioned," etc., like many others he overlooks the fact that the shape of the slots, if their total area is sufficient, cannot influence the quantity of air admitted. The effective area of the slots will be the same at the same engine speeds no matter how the openings are shaped, for it only means that the movement of the piston will be greater or less for the same speed according to the length of the slots. Of course, it follows that if the valve is spring-controlled the shorter the movement of the piston to give the maximum opening the closer will be the regulation. Steam reducing valves which are required to regulate very closely to a set pressure are almost invariably furnished with mushroom valves, and I have found this system perfectly satisfactory for air regulators for carburettors, as they may be operated by means of a diaphragm, which is proof against wear and cannot jamb.

A valve of this nature (having sufficient capacity) fitted to a carburettor simply prevents the partial vacuum from rising above a certain maximum, and if the choke tube be proportioned so that the velocity of air entering cannot fall below a certain value even at the lowest engine speed we have the modern constant vacuum carburettor.

With a carburettor working on this principle there is something definite to work upon, we are assured of a supply of petrol together with a strong enough current of air to effectually break the petrol stream up into a fine spray at the very lowest speeds. There will be no tendency for the jet to "dribble," and therefore no necessity for turbulence of the air or petrol at low speeds as suggested by Mr. Brewer, which, at any rate in the case of the petrol, would, to my mind, be objectionable on account of the increased inertia effect produced.

Now we can investigate the conditions of petrol supply under constant vacuum conditions, ignoring for the time the pulsating action of the charge entering the engine. Under these conditions the petrol would issue from the jet at a fixed rate without regard to the speed of the engine or the power required, therefore if the jet were of the correct size to supply the engine at top speed and full power, the mixture would obviously be on the rich side at low speeds. Now as the amount of petrol required and the throttle opening bear some relation to each other, one way over the difficulty is to arrange a variable jet operated by the movement of the throttle so that as the throttle is opened the area of the jet is also increased.

I do not intend going fully into the question of the inertia of the petrol in the jet here, but a careful study will reveal the fact that it can almost be ignored if the carburettor is fitted with a variable jet as described above, one reason being that as the speed of the engine increases the quantity of the mixture taken in per stroke gradually diminishes, and as the effect of the inertia of the petrol is to maintain the flow at low speeds. the two facts tend to neutralise each other to some extent when the variation of the size of the jet by the opening and closing of the throttle is also taken into consideration. Probably this accounts for the poor results obtainable with any specially constructed jet arranged to counteract the inertia of the petrol.

I have found from practical experience that the best results are obtainable from a carburettor when the negative pressure is not allowed to fall below about 14 ins. of water or ½lb., and with this pressure maintained throughout the full range of the engine's speed. Below this pressure there is a loss due to "dribbling" at the jet and also through the petrol not being thoroughly mixed with the air, and therefore some of it passes through the engine in an unconsumed state. The low velocity of the air through some of the older pattern carburettors certainly accounted to a large extent for their wastefulness.

A negative pressure of ½lb. corresponds to a velocity of about 240 feet per second through the choke tube, but this would vary to some extent with the temperature at which the air is delivered to the carburettor.

The size of choke tube should be calculated on this velocity at the lowest engine speed, and therefore may be based on the full piston displacement, as practically a full charge would be taken in at low speeds, whereas at, say, 20 miles per hour, as recommended by Mr. Brewer, this would not be the case. In calculating the size of the passage for the air entering the carburettor necessary to create the difference in pressure (it may be in the form of a plate (White and Poppe) or tube), this must be based on the effective area and not the measured area.

I cannot quite see the usefulness of Mr. Brewer's figures if a guess has to be made to arrive at some of the results, also the curve which he states shows the consumption of petrol per h.p. hour seems very vague.

W. BOURNE-DALE.

UNIVERSAL JOINTS.

Sir,—There are one or two points in the article in your June issue, on Universal Joints, upon which I should like, with your permission, to comment.

First of these is the writer's criticism of the De Dion type of joint. I very much doubt whether this joint can be shown to wear more rapidly than any of the cross-pin types, given equally correct design and similar conditions; at any rate, such is not my experience. As a joint which will telescope to any extent, and which at the same time gives a true universal motion, it stands almost alone, a fact proved by its wide adoption.

The use of a joint sliding upon a feathered shaft is to my mind, and in my experience, most unsatisfactory. The diameter of the feathered portion cannot be large, consequently, the pressure on the feathers must be very intense, and may, in any but a small car, easily reach 3,000 or 4,000 lbs. per square inch; coupled with this is the fact that the lubrication so near the centre of the shaft is, to say the least, doubtful, so that the potentialities for wear, and even seizure, are considerable.

With the De Dion type of joint the pressure on the sliding portion is, by reason of its greater distance from the centre, and the possibility of providing sufficient wearing surface, reduced to reasonable proportions.

The 1,500 lbs. pressure which the author considers allowable, is, to my mind, a maximum, and I have in mind several instances of cars in which the bearing pressure in the universal joints was no more than this, and which showed exceedingly rapid wear, coupled, of course, with a tendency to seize. Personally, I should favour keeping the pressure down to not more than 1,200 or even 1,000 lbs. per square inch, where possible.

The reduction of pressure on these bearings is all the more imperative on account of the difficulty of lubricating them satisfactorily. Grease is far from satisfactory, even when a separate grease cup can be fitted to each bearing, and oil should be used wherever possible. In most cases the outer cover can be designed to retain it.

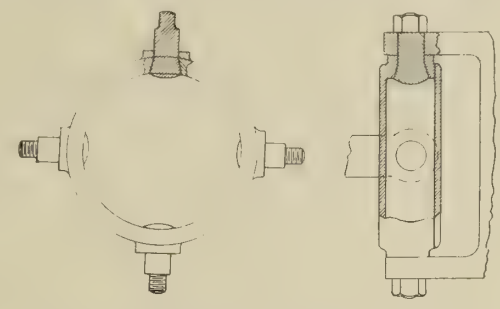
Another point touched upon by the writer of the article, and which is almost worthy of separate treatment, is the use of the rear springs as radius and torque rods. Now, while the omission of radius rods and the use of the springs as such is perhaps not a serious matter, and merely involves the use of an extra heavy top plate, the infliction upon them also of the stresses induced by the driving and braking torques cannot surely be upheld upon any mechanical grounds.

I am aware that the system is followed by some people, but then so much bad design may be seen in many directions, not only on motor cars, and that much of it is induced to work more or less satisfactorily scarcely justifies its support by those who know, or should know, better.

Trusting that I have not encroached too far on your valuable space, and wishing your journal every success.

HUBERT C. CLARK.

Sir,—Referring to your article on "Universal Joints," in your June issue, the enclosed rough sketch may interest you. It is distinctly clever, and obviates several difficulties. There is a plain malleable or forged drum (floating), having four taper holes drilled opposite to each other, into which pins are put, two connecting



to the propeller shaft, and two to the gear box shaft. The pins being taper, and put in from inside the drum (one head of which is removable) can never come adrift and let the shaft down to act as a forward sprag; also the drum containing heavy oil always keeps the pins lubricated, owing to centrifugal force driving oil outwards. The joint is very simple and cheap to make. The pins being drilled for lubrication, act very satisfactorily.

Congratulating you on the success of your first edition.

A. R. LANGTON.

Sir,—I notice in the detailed description of the 8 h.p. Rover car, which appears in No. 1 of *The Automobile Engineer*, you reproduce a sketch of the spherical type of universal joint as fitted to these cars. Until I saw the mention of it in your journal, I was not aware that the Rover Co. were using this type of joint, which, however, is not "unique," unless it has some detail of design not shown in the perspective sketch, but is one that I have favoured since my experience with it on a well-known French car some three years ago. It appeared to me to be a type presenting very large bearing surfaces in proportion to the external dimensions, far in excess of those of any other type of similar size. It can be so constructed that wear can be taken up easily and thoroughly by means of an assembling bolt, not shown in your sketch, and owing to the fact that I had some of these joints made for another purpose, I know that the cost of manufacture is very low when once the necessary jigs and tools have been made. I imagine I am correct in saying that neither the Rover Co. nor the French firm referred to claim any patent rights in connection with this design, and I must say I am surprised that no other makers of automobiles have adopted this type as standard. As you state in your reference to it, every bearing surface can be case-hardened, and, as I have intimated, my experience shows that, when passably well made, greater satisfaction and longer life is obtained from it than from any other type I have met with in ten years' experience. I may add that those which I had made for another purpose were of far larger dimensions than would be required for automobile work, and although their external dimensions were much smaller than the joints which were replaced, they have given far greater satisfaction. I am wondering whether the Rover Co. are using this joint on all their models.

Whilst on the subject of universal joints I should like to criticise the use of ball bearings in these. I really cannot see that they are of any advantage, for if the angle of the universally-jointed shaft is as it should be, i.e., approximately horizontal to the line of drive, there is a very small relative movement between the sections of the joint. It seems apparent, therefore, that the races of the ball bearings will very soon become "pitted" where individual balls make contact, each ball making a bed for itself on one of the races. When this stage is reached the "ball bearing" will resolve itself in a plain case-hardened bearing with a dozen or so minute points of contact. Surely by case-hardening the bushes and pins of the ordinary type of star joint a far greater area of wearing surface can be obtained at less expense of manufacture and giving better results as regards length of life than where similar joints have ball bearings, especially if the joints in both cases be efficiently lubricated.

B. W. M.

Sir,—In reference to the article on "Universal Joints and the Wear on Same," I accept your invitation to criticise.

I have been in the automobile industry from its infancy, but I have never yet seen a propeller shaft designed where its chief points, viz., wear and strain and stress on the pins have been properly considered, yet the matter to me seems to be very simple.

Take the case of a propeller shaft as generally designed with the jaws parallel to each other. The same thing applies to all three types. When coupled up to the gear box and axle we find we have two pins parallel to each other and two vertical.

Now consider the action of this. We find that we only get four points of vertical movement without cross strain on the pins, in one revolution, so when the shaft is in such a position that none of the pins are horizontal, and the axle nose at that moment wants to dip down, owing to inequalities of the road surface, neither of the joints are in a position to accommodate it, hence a great twisting and wearing strain on the pins.

Now consider a jointed shaft so arranged that no two pins are parallel to each other by having the jaws standing at an angle of 45 degrees to each other. Now consider the action of this when coupled up to the axle and gear box by its knuckles and jaws on the axle nose and gear shaft. We find we have not four points, but eight in one revolution when it is possible to get a vertical movement, thus reducing the strain one-half. By this method the shaft is practically flexible at all points. I have tried this many times when repairing and overhauling cars, and found it excellent practice, and the lack of lubricant seems to have little effect on the joint. I suggested this to one firm, and they altered theirs, and found a great difference in both wear and noise, but they have now copied the De Dion joint, and have fallen back into the old way, making the pins at each end of the shaft at right angles to each other, with the resulting noisy axle and gear box.

I should like to have some other people's views on this. Perhaps you can arrange it.

MANCHESTER TESTER.

[Although it is obvious that the usual arrangement of joints gives perfectly free movement without any straining of the parts, our correspondent's letter is interesting in so far as greater durability is claimed for joints arranged in the manner suggested.—ED.]

MULTI-ROW BALL BEARINGS.

Sir,—Your description of the 16-20 h.p. Wolseley car in your June number interested me exceedingly, and you must allow me to congratulate you upon it and upon "The Automobile Engineer" generally, as it is what our branch of the engineering profession has been wanting for so long. I am glad to note that there is a flavour of friendly criticism in your description, and this brings up a matter which I should like to see ventilated. You do not criticise the Wolseley practice with regard to ball bearings. It appears to me that so far as the rear wheels are concerned the theory is that half a dozen rings of quite small balls are equal to two rings of balls of larger diameter. This seems to be a perfectly reasonable assumption, but I believe it is based upon a fallacy, because I have found in practice that five or six small rings are not equal to two rings of larger balls, though in each case the size of the balls used has been based upon advice from the ball bearing manufacturers.

I believe the reason that the small and numerous rings wear out more quickly than the two

large ones is because there is no such thing as rigidity or absolute truth. The theory is that each of the five or six rings of balls will take its fair share of the load, but a very small degree of flexion or a trifling inaccuracy in manufacture results in one or more of the small rings getting more than its fair share of load, while another is perhaps only half loaded. This inequality of load starts wear, and the multi-row bearing does not last so long as the two-row type. The great advantage of it is that it enables a small and neat hub to be produced, but I do not see that it has any other point to recommend it.

DOUBLERACE.

AUTOMATIC ENGINE STARTING.

Sir,—With reference to the article on "Automatic Engine Starters," by Mr. J. Bell, in the first issue of your excellent publication, your contributor has apparently overlooked the fact that a dynamo for accumulator charging is always shunt-wound, i.e., the field magnet winding is connected as a by-pass circuit of high resistance across the armature brushes, through which only a small portion of the main current passes.

The effect of a discharge back from the battery to the dynamo will result in the machine becoming for the time being a motor capable of giving off useful power at the pulley and continuing to revolve in the *same* direction as before, for, while the armature current is reversed, the exciting current will have the same direction as when worked as a dynamo. Permanent magnet machines are never used for this purpose. However, as your contributor points out, it is improbable this method will become popular, owing to the size of the battery, though it is often made use of in country house electric light plants for starting the gas or oil engines usually employed as power producers.

F. L. SCHOFIELD.

STEERING CONNECTIONS.

Sir,—May I venture to suggest that expert opinion and advice is badly wanted on the subject of pin joints in the steering connections of automobiles, particularly those using solid tyres. Many cars now on the road are faulty in this respect, the steering joints wearing out extremely rapidly, those on the front cross tie rod especially.

The joints are made in various ways; sometimes a forged steel fork-end is brazed into a tube, the end of the steel steering arms on the stub axles providing a bearing for the pin, which is prevented from turning in the fork-end by a dowel peg. Such joints are satisfactory when quite new, but as soon as there is any slack developed between the pin and its bearing a hammering action takes place. Eventually, the dowel peg works loose, and the pin cuts the holes in the fork-ends oval. Putting a hardened steel bush in the steering arm bearing lengthens the life, but does not entirely get over the difficulty. The fork-end, bearing, and pin all require to be hardened, but where the cross tie rod is tubular hardening presents difficulties. Also the proportions of these joints vary largely.

It would be extremely interesting to know if anyone can give an accurate account of the strains and stress to which the various steering joints are subjected.

"DECIMAL SIX."

GRINDING MACHINERY.

Sir,—In your article on Grinding, appearing on page 8 of your June issue, you have inserted our name as agents for the Norton Grinding Co. This is incorrect, our friends, Messrs. Ludw. Loewe and Co., Ltd., being the representatives for this company.

You have, moreover, omitted to include our name as builders of modern grinding machinery, although we can claim to be the largest makers of such machinery on up-to-date lines in this country.

This will readily be seen from the range and variety of our line, which includes the following:—Plain grinding machines in 11 different sizes, from 4ins. by 24ins. upwards; Universal Grinders, from 10ins. by 24ins. upwards; Internal Grinders of two types; Surface-Grinders, both horizontal and vertical; Ring Grinders; Single and Double Wet Tool Grinders, etc.

The above are standard patterns, exclusive of various special grinders made expressly to meet unusual requirements. They are all designed and constructed entirely in our own works at Frederick Road, Pendleton, Manchester.

To rectify this omission we must ask you to insert this letter in your issue for July.

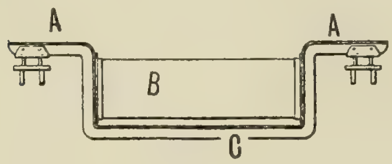
CHARLES CHURCHILL & CO., LTD.

RECENT AUTOMOBILE PATENTS.

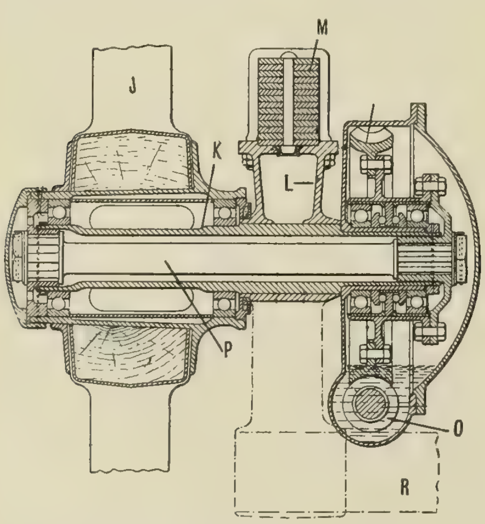
By Eric W. Walford, F.C.I.P.A.

An Interesting Motor Omnibus Construction.

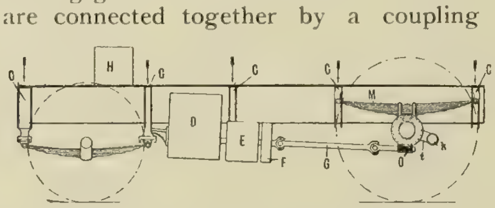
THE frame is of channel section with flanges at each side at A, which conveniently form the seats, the bottom of the channel acting as the footboard. The frame is stiffened by a transverse tubular member B, which constitutes the petrol tank. Throughout



the length of the frame is arranged a number of stiffeners C, which support the frame directly upon the main springs. On each side, under the flanges A, is arranged an engine D, dynamo-motor E, and clutch F, which drive through a uni-



versal jointed shaft G on to each road wheel. The power system is thus duplicated and evenly distributed, whilst a further feature is that the load is carried directly in the line of its points of support, that is to say, the wheels. The petrol electric system described is not essential, as ordinary mechanical gear boxes can be employed, the two change-speed levers being connected together. In the particular case illustrated an accumulator is employed, this being illustrated at H. Each of the rear road wheels J is supported on a fixed sleeve K, carried by a bracket L, which forms the platform for the rear spring M. On the inner end of the sleeve K is arranged a driving-gear wheel, which is engaged by a worm O on the shaft G. The driving-gear wheel and the road wheel are connected together by a coupling

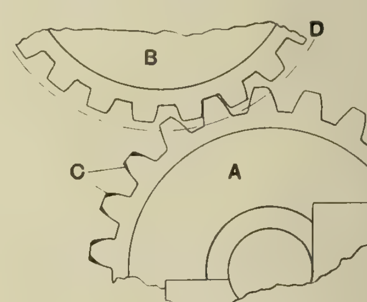


shaft P. A feature of this construction is that the gear and road wheel are arranged on opposite sides of the supporting bracket L, enabling the road wheel to lie very close up to the supporting spring M. The rear axle R is cranked underneath the frame. It will be noticed that none of the mechanism lies underneath the frame, and it is therefore easily accessible.

Daimler Motor Co. (1904), Ltd., and F. W. Lanchester. No. 9,259/09.

A Cutter for Finishing Involute Gears.

This ingenious cutter is used for finishing the profile of gear wheel teeth. In the illustrations the wheel being operated on is marked A, and the cutter B, and it will be seen that the teeth of the wheel A are larger than they should be, and require cutting down, as indicated at C. The cutter takes the form of a corresponding involute gear, having one tooth less than the wheel A, and its periphery is ground eccentrically, as shown by the chain line D. The result is that the corners of the cutter teeth are left sharp and of gradually varying height. In operation the wheel A is mounted in a bearing in a carrier, and the cutter D is revolved, for which purpose it may be mounted upon a milling machine arbor. The wheels are set slightly in mesh, the cutter acting as driver, and are gradually fed closer, so that the cuts are taken further in until the pitch circle is reached. The cut is progressive, owing to the variation in height of the cutter teeth, and by using one tooth less a



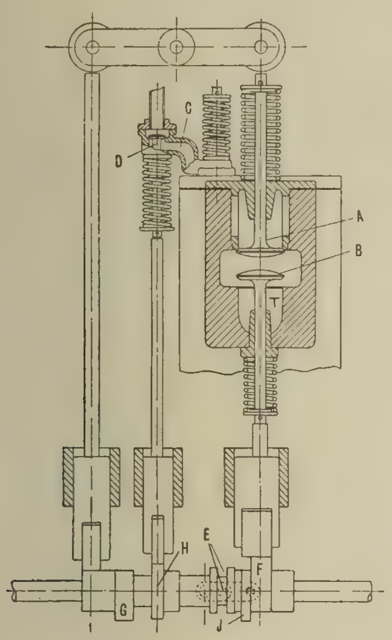
"hunting tooth" effect is obtained, so that any errors are spread out and corrected and accurate spacing ensured. When the wheel is finished in one direction it is withdrawn, turned over, and run in the opposite direction. An important feature is that the cut is taken in the direction of wear, that is to say, lengthwise of the tooth surface, and not transversely. The cutter can be accurately, quickly, and evenly sharpened, as its periphery is circular.

P. A. Poppe. No. 14,763/09.

A Compressed Air Starting System.

The ordinary engine inlet valve is shown at A, and exhaust valve at B. Compressed air under pressure can be admitted through the tube C by way of the poppet valve D. The cams for operating these valves can be moved laterally by means of a control lever arranged on the spindle E. When arranged in one posi-

tion the exhaust cam F and inlet cam G operate normally, but when the lever is moved the cams are slid along into the position illustrated, when the cam G is drawn out of action, and a cam H brought underneath the tappet for the



valve D. Further, a supplementary exhaust valve cam J is brought under the exhaust tappet, so that the exhaust valve is opened on each stroke, on one stroke by the cam F, and on the other by the cam J. As soon as the engine has fired the lever is thrown over, and the cams assume their normal position. L. A. Hindley, H. D. Hindley, and W. Standford. No. 15,390/09.

An Interesting Magneto.

So far as ignition apparatus is concerned, a new and interesting feature is involved in this system by the combination of two different principles of producing a high-tension current from a low-tension one.

In an induction coil a current is induced in the secondary winding by varying the strength of the current in the primary winding, and the greater the variation the higher the efficiency of transformation. In sparking devices the more common method of varying the strength of the primary current is to first establish it, and then to suddenly interrupt it, bringing about a variation between its maximum value and zero. Another method, not so commonly used, is to suddenly establish the current in the primary winding, this being, until recently, the case with the Eiseman magneto. In the case under consideration the primary current is first of all established, then suddenly interrupted, and then suddenly established in the opposite direction. The result is that the variation, instead of being between a maximum and zero, is between a positive maximum and a negative maximum.

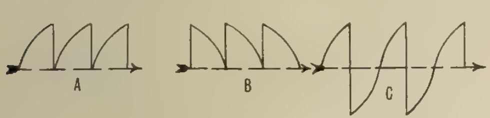


Fig. I.

These effects may be diagrammatically demonstrated as in Figure I. At A is shown a current suddenly interrupted after being established. At B a current suddenly established is represented, while C represents the effects produced by the Brooks-Alston magneto.

The specification shows several methods of carrying this into effect. The primary winding of the induction coil may consist of two separate coils wound in opposite directions, and this arrangement is shown diagrammatically in Figure II.

The two primariees P P are connected, as shown, to the armature A, through the double contact maker, or breaker, C. When the blade D is moved into contact with the spring E, the contact is simultaneously broken, and the effect is to suddenly interrupt the current in one of the primary coils, and to establish it in the other, and as the coils are wound in opposite directions, the effects are relatively positive and negative.

Figure III. shows diagrammatically a different arrangement for producing the same effect. The armature winding A is divided into two coils by earthing it at the centre, and by means of the contact breaker C, and the system of wiring

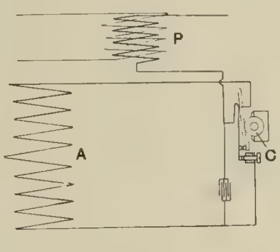


Fig. III.

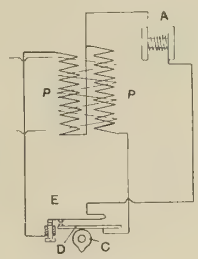


Fig. II.

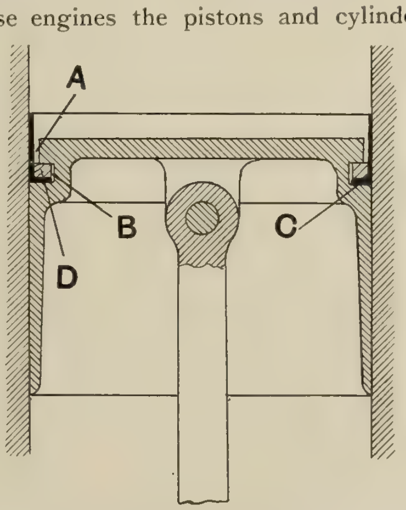
shown, currents, opposite in direction, are interrupted and established in the single primary winding of the induction coil P.

The patent specification and drawings describe and illustrate a complete practical form of magneto embodying these principles.

B. Brooks and F. H. Alston. No. 15,803/09.

Piston Ring.

This piston ring is of the type employed on the Gnome revolving cylinder aero engines. It will be remembered that in these engines the pistons and cylinders

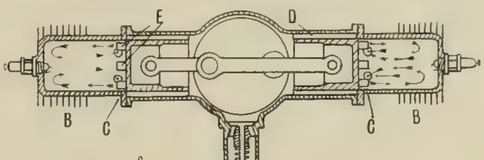


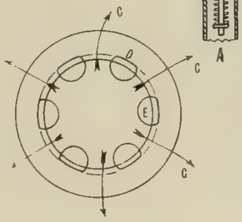
are of steel, so that some special piston ring arrangement has to be provided. In the case illustrated the piston at A is reduced, and a groove is formed at B, into which is let one end C of an L-shaped brass ring, this ring being split, and extending above the piston head. The ring is held in place by a segmental ring D, which is jammed into the groove to lock the brass ring. As will be gathered, the ring expands under the internal pressure, and maintains a tight joint.

Société des Moteurs Gnome. No. 21,664/09.

A Two-Stroke Engine.

This engine is of the two-cylinder opposed two-throw crank type. The gas is sucked into the crank chamber through the inlet valve A, and transferred to the working cylinders B, exhausting through the ports C. The feature of the invention is that the transfer passages are formed at D in the cylinder. To prevent



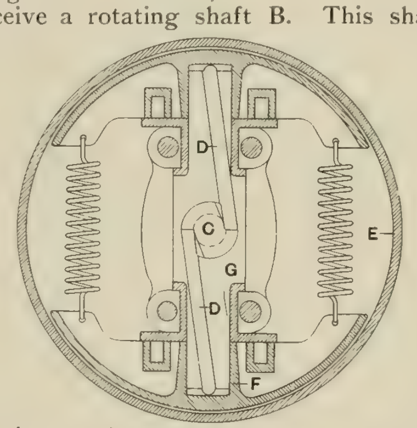


the fresh gas passing out through the exhaust ports C, the pistons are arranged with scoops or shields E, which surround the ends of the transfer passages, and shoot the gas up to the cylinder ends, as shown by the arrows.

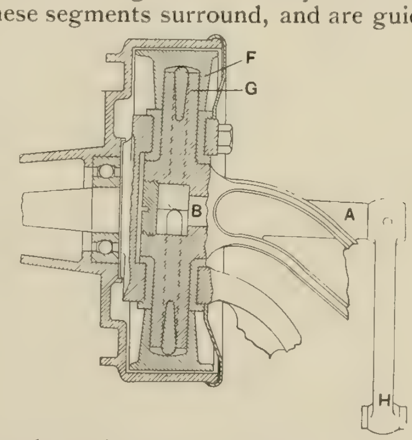
M. and W. Edwards. No. 21,710/09.

Front Wheel Brakes.

At each end of the front axle is arranged a bracket A, which is bored to receive a rotating shaft B. This shaft



carries at the end a cam C, engaging push rods D, which when actuated force the brake segments radially outwards. These segments surround, and are guided



by, the axle head G, and are held together by springs, as illustrated. The two operating levers H are connected to any compensating device.

R. O. Harper. No. 1,708/09.

CARBURETTORS.

During the past month we have received communications from several owners of carburettor patents who claim that their particular device fulfils all the requirements for the perfect carburettor, as set forth by our contributor, Mr. R. W. A. Brewer, in our last issue. It is extremely difficult to forecast the behaviour of any delicate piece of apparatus with any degree of certainty, and a carburettor must always be delicate in action, however robust its construction may be. The carburettors which are described hereunder both appear likely to give good results in use, and, in fact, they have already been proved to do so. They are perhaps already well known, but their description will bear repetition, because it enables comparison to be made between each one of them and Mr. Brewer's ideal design.

One carburettor which operates in a manner which is a close approximation to the theoretical is the Polyrhoe, shown in Fig. I. It is called by its makers an "expanding" carburettor, as both the air entry port and the jets exposed vary in area directly with the speed of the engine. The essential part of the carburettor may be described roughly as a rectangular opening, with a row of tiny jets along one side. Beneath this opening is a cylindrical chamber, and in this chamber

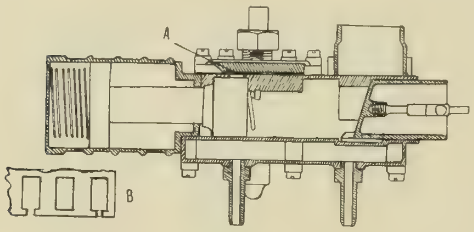


Fig. I.

a piston slides. When the piston is at one end of its travel, there is only a small opening between the piston and the end of the rectangle, and air can only reach the engine by passing through this small space. As the speed increases the piston moves back, uncovering more and more of the rectangle, and so keeping the velocity of the incoming air at an approximately unvarying amount. There being a row of jets all along the side of the opening, each movement of the piston causes the entering air to pass a number of jets in proportion to its volume per unit time. Thus we have a construction which supplies a mixture of constant proportions at constant velocity, whatever the engine speed.

The movement of the piston which controls the size of the opening is controlled by another piston on the same rod, moving in a separate cylinder, and backed by a spring. Increasing suction causes compression of this spring, and so movement of both pistons.

The construction of the jets is unusual, they being formed by cutting slots in a thin brass plate. Each slot is, of course, extremely narrow, but each ends in a comparatively large hole, the method being to punch a row of holes and to then slit the plate between its outer edge and every alternate hole. Several plates are made thus, and are then placed one above the other, with the slots staggered. The holes register, and petrol is led from a float chamber, through a filter, and up a wide passage, to the holes in the plates. This means that a constant supply is maintained at the inner end of each slit, so that inertia troubles are reduced almost to vanishing point. It is claimed that there is no leakage whatever from the inoperative jets when running with only one end of the carburettor in action, and also that there is no overflow if the throttle is shut and the engine speed suddenly reduced. The jets are shown at A in Fig. I, and a portion of a jet plate at B in the same figure.

In order to provide for differences in atmospheric conditions the width of the rectangle can be varied by a slide controlled by a Bowden wire mechanism, or otherwise. The use of this hand control merely varies the speed of the air past the jets, and acts equally for all degrees of opening. It is supposed to be set only occasionally, when alterations of temperature demand it, and it is not a hand adjustment which requires to be made while the car is being driven. The obvious disadvantage of the design is the extreme delicacy of the jet plates, but their accurate manufacture is not a matter of great difficulty, not nearly as great as would be the drilling of correspondingly small holes. There are remarkably few parts to require cleaning, as the filter being between the float chamber and the jets protects the latter effectively. The whole carburettor shows evidence of the designer's appreciation of the theory of carburation, and may certainly be regarded as one of the best attempts to produce a perfect carburettor that has yet been made.

Another carburettor for which considerable claims are made is the S.U. Its principal characteristics are that all the air passes the jet, that the speed of the air is kept approximately constant, that the fuel feed is increased in proportion to the air feed, and that there are but few moving parts, which are of the most simple description, while no springs are employed.

The action is made clear by Fig. II. It will be seen that the jet is an annular space between a taper needle and a cylindrical tube, so that as the needle is withdrawn the area of the jet increases. As the jet passage makes only a small angle with the horizontal, and as the length of the restricted part of the passage is small, there should be theoretically, but small liability to inertia troubles, and this is proved to be the case in practice, as it is impossible to cause the jet to overflow by any manipulation of the throttle. The air passage is normally filled almost entirely by the cylindrical gunmetal piece A, which is secured to a spindle also fixed to the base of a leather bellows B. This bellows is in a casing open to the atmosphere, and the only outlet from the inside of the bellows is by a passage leading to a point in the inlet pipe just below the throttle. Thus, if there is a partial vacuum in the inlet pipe at this point, the air in the bellows expands, becomes rarefied, and so reduces the internal pressure. The external atmospheric pressure instantly closes the bellows, thereby lifting the cylinder A, and so opening the air passage. The needle of the petrol jet is also attached to the same spindle as the bellows, so the jet area increases as the bellows closes. It might be pointed out that the part A does not touch the walls of the chamber into which it rises as the bellows close, the whole of the guiding being performed by the spindle.

Adjustment to any engine is made, firstly by choosing a carburettor of such a size that it will admit enough air to supply the engine at its maximum speed without undue air velocity, and secondly, by altering the shape or size of the jet needle till sufficient petrol is supplied at all speeds to correspond with the needs of the engine.

It is obvious that if the proportions of the

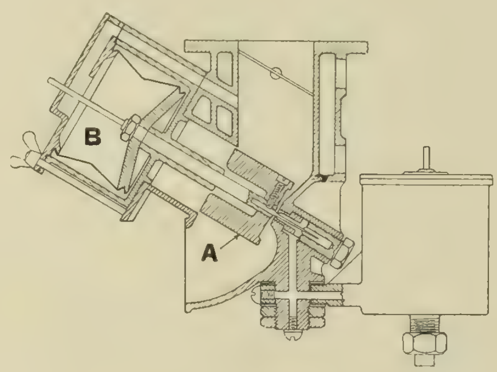


Fig. II.

parts are correct, a very close approximation to a steady mixture would be supplied, and it is equally obvious that there are only two wearing parts, namely, the spindle which guides the part A, and the leather of the bellows. The spindle should be almost everlasting, and we are assured that the leather is extremely durable.

The carburettor is one which might be rather troublesome to adjust to a single engine, but it would be easy to manufacture to suit a standard engine, and should then require practically no tuning-up at all. One very curious claim made for it is that differences in the temperature of the air or the specific gravity of the fuel do not affect the adjustment; in fact, it is stated that once it is adjusted for, say, petrol, the same carburettor can be used for benzole, or even paraffin, without adjustment. A certain evenness of temperature is ensured by the water-jacketing, which doubtless has something to do with this peculiarity.

We recently witnessed a short demonstration of the behaviour of the carburettor as fitted to a Zust, and noticed that there was a complete absence of choking when the throttle was opened suddenly, while the engine could be run very slowly without any leakage of fuel from the jet. So far as it was possible to judge, it appeared that the position of the controlling weight A was quite unaffected by throttle opening, and varied only with the engine speed. On our expressing a desire to see the bellows the latter was removed, together with the needle, in a very few moments, and on the parts being replaced the engine was started, on benzole, at the second turn without flooding (it has been stopped by the removal of the bellows, and not by the switch). The S.U. has been proved to be economical as compared with a number of other carburettors, and no doubt this is due to the fact that it wastes no fuel at low speeds.

A NEW BALL JOURNAL.

The British agents for the F. & S. ball bearings, the Tormo Manufacturing Company, have informed us that they are now able to take orders for their double row bearing, which was first introduced last November. It is claimed that this bearing will prove reasonably durable if used for hub work, even when unsupported by a thrust washer, and it must obviously be at least twice as strong as a single row bearing. Fig. 1 shows the method of fitting recommended. The outer and inner rings of the bearing are

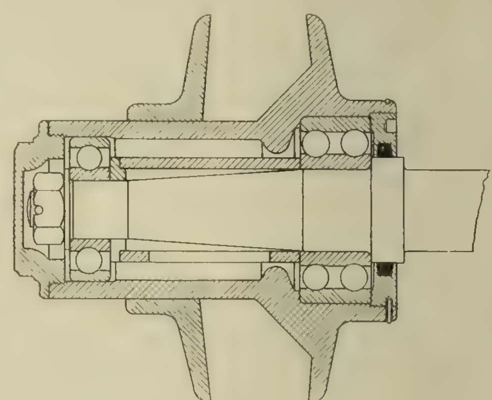


Fig. I.

both hardened right through, the material being chrome-nickel-steel, and the balls are separated by an aluminium cage situated in the centre, between the two races. We understand that the price is a little less than that of a single row journal, plus a thrust washer. The new bearing is slightly less than double the width of a single row journal, and we should think it will prove still more useful for gearbox or axle work, where it will only be called upon to support journal loads, than for the purpose for which it is recommended.

CATALOGUES RECEIVED.

LATHES.—Messrs. Alfred Herbert, Ltd., have issued a new edition of the section of their catalogue relating to their patent hexagon turret lathes. Not only does it describe the lathes with the utmost minuteness, but many examples of their work are illustrated, especially those produced with the aid of the patent roller steady. This attachment is extremely useful in that it permits a very deep cut to be taken on quite small work, while leaving a very good surface. An example of this fact, quoted in the catalogue, is as follows:—A 1½in. mild steel black bar was reduced to ¾in. at a single cut, firstly with a carbon steel cutter and a flat steady, and secondly with a "special" high-speed steel cutter and a roller steady. In the first case the best output obtainable, consistent with reasonable good work, was .735 lbs. per minute, at 149 revolutions with a feed of 2.01ins. per minute. In the second case the output was 8.8 lbs. per minute, at 470 revolutions, with a feed of 23.5ins. per minute.

FANS AND MOTORS.—A new small catalogue of ventilating fans and small power motors has been recently issued by C. A. Vandervell and Co. In it are illustrated and described fans of various sizes for shop, office, or marine use, and a number of small motors for polishing or other light work. These fans and motors are made for direct current only, at the present time, but we understand that they are becoming quite an important section of the C.A.V. manufactures.

TYRES AND INDIARUBBER SUNDRIES.—A convenient desk card, giving the prices of their inner tubes for car tyres has been issued by the St. Helens Cable and Rubber Co., Ltd. A similar card without prices gives a list of some of the more important sundries manufactured for the motor trade by the same firm.

THE

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Articles of a technical nature relating to the design or construction of automobiles for land, air, or water, will be carefully considered by the editor. Matter must be clearly written or typed on one side of the paper only, and a stamped addressed envelope must be enclosed for return. No responsibility can be accepted for the safety of contributions although every reasonable care will be taken.

Correspondence on interesting subjects, which come within the scope of this journal is cordially invited. All letters must contain the name and address of the writer, but not necessarily for publication. If a reply is desired by post enquiries must be accompanied by a stamped addressed envelope.

CONTENTS.

FASHION AND DESIGN.

FASHION may be regarded as a folly, whether its application be to battleships or feminine apparel, but it should never be forgotten that while the mode of the moment may be a foolish one, it may also be of an entirely contrary nature. During the short history of automobilism fashion has exercised a great influence and, though it has led to the reproduction and multiplication of more than one folly, it has likewise assisted towards the popularising of some valuable novel features of car construction. Probably the first time that the copying of the designs of others became common was contemporary with the successes of Panhard et Levassor, and here who shall say that the copying was otherwise than the wisest course that could have been followed? The Panhard (if we except the belt-driven Benz) was the first reliable car to be manufactured in anything more than the smallest quantities, and though it was inefficient, noisy, and unmechanical in many features, it excelled most other chassis, in that it could be relied upon to give satisfactory service. Surely it was better to follow the example of such success than to continue to make experiment in other directions, especially as manufacturing firms were faced with the necessity of making a profit within a reasonable number of years. Following the general adoption of the main features of the Panhard came the astonishing success of the Mercedes. The Canstatt built car was in many respects itself a copy of the Panhard, but it was much neater in appearance, and thus the reproduction of its peculiarities exercised an undoubtedly beneficial effect upon cars as a whole.

Then for a time came a period of quiesence, when each manufacturer continued to make small improvements and to copy those made simultaneously by his neighbours, and, in fact, there was not any startling deviation from the Mercedes type of chassis, save the gradual replacement of the propeller shaft for the chain drive, until it was noticed that the Renault company were supplying an increasing number of chassis, both in France and this country. For the last two years there has been an epidemic of pseudo-Renaults, and in this case we cannot see that anything has been gained by the creation of the third prevailing passion.

However, before examining the merits of each of the cases we have quoted, some lesser instances may be mentioned. The change from quadrant control of gears to various modifications of gate was principally due to fashion and the desire to have the most up-to-date chassis. Also the internal expanding brake owes its universality to popularity more than to its undoubted convenience. Yet another instance is the arrangement of clutch, brake, gear, and engine controls, and again, most recently, the craze for a "clean" chassis, which means one with all parts simplified, and all that cannot be simplified concealed in a casing with a smooth exterior.

If all these cases of the effect of fashion are considered together it appears that automobile construction has gained more by universal copying than it has lost. It is impossible to tell what the standard practice of to-day might have been had, say the Lanchester, become the leading car, before the Panhard. Eventually the result might have been the same, but on the other hand, it might have been quite different, and in passing, it is worthy of remark that the adoption of Lanchester features by other makers, which has taken place to a small extent, is an example of commendable imitation, as it has been brought about by the intrinsic value of the designs and not by fashion.

In most of the instances given the effect of copying was good, because each innovation was made by an experienced firm after experiment, and those with less experience to some extent reaped the benefit of the labours of the former. The danger arises when new features are introduced for the mere sake of their novelty, and when innovations are alterations rather than improvements.

Thus with some Renault copyists who often, speaking metaphorically, try to enclose the skeleton of a lion in the hide of a gazelle, the effect on the convenience and efficiency of their chassis is either nothing at all or perhaps even detrimental. In the haste to snatch up and imitate a feature which has long been a jealously guarded patent everything else has to go by the board. There are, of course, many good points about the Renault system, but like most systems which are the result of years of study, the details must be co-ordinated and are not singly transferable to other systems composed of similar isolated details of various origins combined as best they may be. Those who make a complete copy of another car down to the minutest unpatented details, are in a much more favourable mechanical position, as the effectiveness of the car depends simply upon the ability of the manufacturers to make an accurate reproduction.

Probably the frequency with which the details of popular cars are reproduced on rival chassis has been influenced to a very great extent by the commercial side of the business, as opposed

to the engineering reason of the constructive side, and there can be no doubt whatever but that many ideas of value are forgotten altogether, or are lying neglected in the memory of many designers, simply because they were opposed to standard practice and were therefore feared by the sales department. That there is reason in this commercial attitude is undoubted, because it is never so easy to sell the unusual type as to dispose of the common variety of any article, unless the unusual is obviously better to the eye of an uneducated purchaser. Lately, however, there have been signs that originality will soon be afforded a wider scope. New engines, new transmission systems, and new forms of suspension are being given a fair trial, by large manufacturers, as well as small ones. Thus in the matter of lubrication, the splash system has been tried and found wanting, under modern conditions of piston and crankshaft speed. Something better is needed, must be found, in fact, and so there is now no standard practice in this particular. This is a matter for rejoicing, not only because it is good in itself, but because it lays bare the fact that the road automobile is still far from perfect, and if leading makers can differ so much in chassis practice as they do now, others will not hesitate to produce cars still more different from standard ideas. The present state of knowledge is such that there is no danger of a car being put upon the market in an unreliable condition. It may be more efficient or less efficient than the present average, but it will in any case run, and be more or less saleable. Ten years of such experimenting will teach us more than even the last ten years of development have done.

It would be easy to choose individual examples of peculiar forms of chassis that have special advantages of their own, and have for years continued to be made without becoming really popular. The naming of them we may leave safely to our readers. There is no doubt but that they are no longer regarded as impracticable, either by the car-using public or by the rank and file of other manufacturers, and while admitting that the standard practice of the day is probably nearer to the ideal, for general purposes, than are the few peculiar chassis, yet as the latter increase in popularity so may it be found that there are features in them which embody ideas well worthy of the consideration of others.

The individualism of engineers has had more free scope with traction automobile design than it has had with the pleasure car, and the truth of this is obvious from the many widely different types of vehicles, in use for similar purposes. It took but a short time for the discovery to be made that many pleasure car practices had disadvantages when applied to heavy chassis, and as there was no one example conspicuously meet to be copied, each works struggled with the difficulties it encountered in its own way, and found its own method of surmounting them. Often there is nothing to choose between one system and another for reliability or economy, the two being equally efficient, but the use of alternative constructions has made the evolution of the traction vehicle much more rapid than it could have been had it been confined to a single line of progress, and perhaps been forced to go back to a branching point and there commence afresh, as is now happening in the case of omnibus chassis.

For these reasons we welcome the fact that fashion, in some respects, is losing its power, and trust that it will ere long do so in others. The great danger of the moment which is threatening the era of enlightenment is unfortunately the most deadly and most foolish of all fashions—the fashion of appearance. If a car has good clean mechanical lines (using the adjective in its fullest and truest sense) small fault can be found with it on the score of good looks, and solely from the point of view of appearance, distinctiveness is far more telling than the most superlative embodiment of the modes of the moment.

THE DEVELOPMENT OF THE I.I.A.E.

It may certainly be said that the new secretary to the Incorporated Institute of Automobile Engineers will receive his appointment at the close of a session which has been the most memorable in the history of the Institute. Year by year the work done by the youngest engineering body has increased in importance, and there can be no doubt but that the graduates' section has been of very great assistance to those who will in the future be men of authority both theoretically and practically. The educative influence of the discussions at graduates' meetings can scarcely be overestimated, for these gatherings serve well to demonstrate the intimate connection between the theoretical and the actual.

Rapid though the development of automobile engineering has been, it would have been still more rapid had the rule-of-thumb designer paid more attention to the theorist, and if the theorist had given ear more readily to the arguments of the men who depended upon trial and error alone.

At the present time the erstwhile common contempt for theory has almost vanished from the workshop, and this is certainly partly due to the fact that experimental work is being undertaken more and more by men actually engaged in the industry. Theories are much more easily put to the proof than it was possible for them to be a few years ago, especially as regards automobile work, and theorists are, on the other hand, concerning themselves much more closely with the lesser problems of the profession. That is to say, men who encounter difficulties, which they cannot understand, in the ordinary course of their work are acquiring the habit of calling science to their aid, and science is granting that aid readily.

There were many points in the design of early cars for which no one, with any mechanical training, could have been responsible. These points were contrary to the laws of nature, and the laws have since proved their very real existence by unmistakeable manifestations of their importance. This has vindicated the theorist and is exercising considerable influence on the choice of men to take responsible engineering positions in the industry, while the immediate effect has been to create a large body of comparatively young men with a good engineering education and a highly specialised practical knowledge of automobile work with all its peculiarities.

It is certain that there are many who have as good a right to the honour of membership of the Institute of Automobile Engineers as any who already possess that distinction, but who are still in the lower ranks, or even altogether outside the body, and we think that it would be possible to encourage these men to take an active part in the work of the Institute, greatly to the advantage of the latter.

Scientific theory may be divided into two parts, namely, speculative theory and mathematical theory, as may be instanced, for example, by the theory of carburation and the calculation of horse power formulæ. In the former case theory is useless without experiment to furnish data for further theory; in the latter case exact calculation can be made if sufficient mechanical data are available. Speculative theory is often of greater practical value than mathematical theory because it serves to indicate the direction which experiment should take, while mathematical theory gives exact statements which are only true if all the necessary data are exact, and if no factor is omitted.

We would therefore suggest that the speculative theories of the men who are actually engaged in making automobiles, and who have to contend against the practical troubles they encounter, are likely to be of extreme value in the direction of experiment at the present time. The number of unknown quantities which are present in almost all automobile engineering problems causes the collection of data for mathematical investigation to be extremely difficult, and often impossible. So we trust that an effort will be made to induce a greater number of men, actively engaged in automobile work, to take part in the proceedings of the Institute, so that the value of their experiences, and of their deductions therefrom, may be increased by discussion.

Though we do not wish to, in any way, belittle the work of the I.A.E., there is no doubt but that an increase in the number of papers by the class of men we have described would add greatly to the value of the work done during a session, and would thereby enhance the value of elected membership, which should be a high honour, reserved solely for those who can truly be said to have advanced knowledge.

An examination of the proceedings of the Institute during the first two years of its existence shows conclusively that the occasions when a suggestion or a prophecy was made that has since proved of value to other engineers, were chiefly connected with the reading of papers written by men actually engaged in the manufacture of automobiles. Of course, many suggestions have been made, during the discussion of papers, which have since proved to be quite valueless, and even to be based upon misconceptions, but it is noticeable that few such have emanated from those in close touch with the manufacturing side of an automobile factory.

The matter does not, of course, rest entirely with the Institute. It rests almost equally with those whom we suggest the Institute should encourage, but if it becomes understood that the ideas of the men we have referred to will be welcome, we do not think they will be long withheld.

HARD RUNNING AND ITS CURE.

By Archibald Sharp, B.Sc., M.I.A.E., etc.

FOLLOWING the lines of argument contained in the editorial article in the July issue, I propose to discuss briefly a few of the factors that seem to have an influence on the problem. Probably the most important factors are the balancing of the engine, and the evenness of the torque on the crankshaft.

As pointed out in the editorial, while the perfect primary balance of the four cylinder engine may permit the secondary and higher orders of unbalanced forces to be neglected, when comparing with an engine in which the primary forces are unbalanced, these secondary and higher orders of unbalanced force are by no means small. In a single-cylinder engine, let m lbs. be the mass of the reciprocating parts (this includes the mass of the piston and part of the connecting rod), r feet the crank radius, θ the angle the crank has revolved from the inner centre, ω radius per second the angular speed of the crankshaft. If the revolutions per minute be N, $\omega = .1047$ N; q, the ratio of crank to connecting rod length; F lbs. the accelerating force at the instant, then

$$F = \frac{mr\ \omega^2}{g}\left\{\cos\ \theta + q \cos\ 2\theta - \frac{1}{4}\ q^3 \cos\ 4\theta + \frac{9}{128}\ q^5 \cos\ 6\ \theta + \ .\ .\ .\right\}$$

(See "Balancing of Engines, Steam, Gas and Petrol," Sharp, pp. 85-87.)

The first term on the right hand side of the above expression is the primary force of acceleration; the others are respectively the secondary, of the fourth, sixth, and higher orders. When the crank is on the inner dead centre, $\cos\ \theta$, $\cos\ 2\theta$, $\cos\ 4\theta$ . . are each equal to 1. If q be taken as 1/5, the ratios of the maximum values of the primary and higher orders of accelerating force are 1, 1/5, 1/500, 9/400,000 . . . . Thus if the primary force is 300 lbs., at a speed of 1,000 revolutions per minute, those of the 2nd, 4th and 6th orders are 60lbs., 0.6lbs., .007lbs. respectively.

These forces may be considered as due to masses driven with simple harmonic motion from imaginary cranks of the same radius as the actual crank, revolving at 2, 4, 6 times the speed of the actual crank. These imaginary masses are respectively 1/20th, 1/8,000th, 1/1,600,000th parts of the primary mass m.

In a four-cylinder engine the primary forces balance each other, those of the 2nd and higher orders remaining. In a six-cylinder engine the forces of the 1st, 2nd, and 4th orders balance each other, those of the 6th order remaining. Thus in a four-cylinder engine the largest secondary unbalanced force is $4 \times 60 = 240$ lbs.; in a six-cylinder engine the largest unbalanced force, of the 6th order, is $6 \times .007 = .04$ lbs. If the engine speed is increased to 2,000 revolutions per minute, these forces are increased fourfold, viz., 960 lbs. for the four-cylinder engine, 0.16 lbs. for the six-cylinder engine.

Another point of view of imperfect engine balance is to consider the vibratory motion of the spring-supported chassis. Neglecting for the moment the elasticity of the various parts, and their own natural periods of vibration, but considering only the principle of the conservation of movement of the mass-centre of a system of bodies, the vibratory movement of the chassis can be easily calculated. Taking as a numerical example in a four-cylinder engine, m=4 lbs. for each piston and connecting rod, stroke 5 inches, mass of engine and parts rigidly fixed to forepart of chassis 400 lbs. The imaginary secondary mass for the four pistons is $\frac{4 \times 4}{20}$ =4/5 lb., with an amplitude of vibration 5 inches. To preserve the position of the mass centre the 400 lb. mass of the chassis must have an amplitude

$$\frac{4}{5} \times \frac{5}{400} = \frac{1}{100} \text{ inch.}$$

This remains the same for all speeds of the engine. At low speeds it may be scarcely noticed, but at high engine speeds it may well give rise to the sensation of harsh running, even when the disturbing forces are apparently negligibly small, as in the six-cylinder engine. Their effect is probably analogous to the drawing of the bow across a violin string, tending to set into vibration any part of the car that is capable of vibration. In this connection, the interposition of non-elastic packing strips at joints and fastenings may be well worth the trouble of experiment.

The two-cylinder one-crank engine, with its cylinders at right angles, is in better balance than the four-cylinder engine, but the unbalanced forces are still of considerable magnitude. The secondary forces from the two cylinders are at right angles in phase the resultant is therefore $\sqrt{2} = 1.414$ times the secondary unbalanced force for a one-cylinder. Taking the same figures as above, this would be $60 \times 1.414 = 84.8$ lbs.

The five-cylinder one-crank engine, like the Anzani aeroplane engine, seems worthy of attention for motor cars. Its largest unbalanced force is of the fourth order, and with same data as above, is $5 \times 0.6 = 3$ lbs. The theoretical amplitude of vibration of the forepart of the chassis, with the same suppositions as above, is

$$\frac{5 \times 4 \times 5}{8000 \times 400} = \frac{1}{32000} \text{ inch.}$$

an amount negligibly small, in comparison with the errors of the best possible mechanical workmanship.

Concerning torque on the crankshaft, the multiplicity of factors to be considered in a close study of the transmission from the engine crankshaft to the road driving wheels is such that in a short article it is difficult to state clearly one's opinions without running some risk of misconception on some point or other. The outstanding fact, as pointed out in the editorial article, is that the torque on the crankshaft is accompanied by an equal torque of reaction tending to rotate the engine frame in the opposite direction. If the crankshaft be fixed, the cylinders and crankcase will revolve, the mutual relative action between cylinders, pistons and crankshaft being exactly the same in engines of fixed cylinder and revolving cylinder types. If the engine frame is supported on springs, as in the usual live axle car, and if the torque is steady, the engine frame is displaced from its normal position, until the torque of reaction of the springs balances the crankshaft torque. If the torque is variable, the engine frame oscillates about the position of equilibrium. In all explosion engines there is a periodic excess and deficiency of energy above and below that required to overcome the uniform resistance to motion of the car. If the reaction of the springs be regarded as constant throughout the small range of oscillation of the engine frame, the frame oscillation is determined by the equation $\triangle$ C = Ia, where $\triangle$ C is the excess torque, I the moment of inertia of the engine frame about an axis through its mass centre parallel to the crankshaft, and a the angular acceleration about this axis. The amplitude θ of the oscillation is proportional to at^2 where t is time during which the excess torque is exerted. That is, θ is proportional to $\frac{\triangle\ C}{I} t^2$. The amplitude of vibration therefore is inversely proportional to the square of the speed of the engine. This vibration soon reduces to vanishing point as the engine speeds up.

Comparing a small high compression engine with a larger low compression engine, both of the same speed and power, I do not think there is much to choose between the two, in this respect. The indicator diagram of the high compression engine will be higher, but thinner, than that of the low compression engine, the maximum value of the excess torque is greater, but it acts for a relatively shorter time.

With a chain driven car, having the engine set with the crankshaft parallel to the axles, the conditions are different; there is no lateral displacement of the cylinders from the vertical, the torque of reaction of the cylinders being transmitted to and equilibrated by the torque exerted by the differential shaft on its casing, the latter casing being rigidly connected with the engine cylinders.

The engine develops its energy not uniformly, but with periodic fluctuations. This energy is converted into useful driving impulse at the points of contact of the back tyres with the ground; the tangental driving effort of the tyres, therefore, varies periodically. The whole mass of the car acts as a huge flywheel, which has apparently steady motion under the action of the fluctuating driving force. When the car is running on the highest gear the engine flywheel adds very little to this effect, as a short calculation will show. The elasticity of the tyres has probably more effect in damping out jerkiness of the drive, than has the flywheel.

A comparison of the torque curves from a four-cylinder and six-cylinder engine respectively will show that although the latter engine has the advantage of smaller cyclic variation of torque, this advantage is not nearly so great relatively as that due to the better engine balance.

THE WHITE AND POPPE ENGINE TESTING SYSTEM.

With some original notes on the calculation of the dimensions of fan dynamometers for different engines.

IN a contributed article published in our June issue, it was remarked that the systems of testing in use by different manufacturers were many and various, and that they differed largely in efficiency. Having regard to the fact that few makers know what goes on in the works of their competitors, we think that an account of a thoroughly sound testing system will be interesting to many, and we have chosen the one employed by Messrs. White and Poppe, because it gives excellent results, is not costly, and is also complete, in the full sense of the word. There is no need to tune up a White and Poppe engine in the chassis if it is fitted with the makers' carburettor and provided with suitable gear ratios.

With the construction of the engines we are not, at present, concerned, but their careful erection has an effect upon the testing, as it is only by serious accident that an engine can reach the testing department with the valve setting and ignition timing other than exactly correct. Care is taken to obtain a petroleum spirit of, as near as possible, a standard specific gravity, and the consumption of each engine is closely watched.

The engines are handled easily and quickly by means of the rails which are laid throughout the shops. Low trucks can be wheeled down practically any gangway on these rails, so that the transportation of heavy parts from one place to another is a quick and easy job. A portion of the testing shop is shown in Fig. I., and it will be noticed that the arrangements are in every way convenient. The overhead water tanks are not seen, but are a feature of the testing system, and will be referred to later. The setting of the fan dynamometers in recesses completely shut off from the rest of the building is a good arrangement, as it reduces the probability of accident to a negligable amount, being a great deal safer than the more usual railings, with guards that can be easily detached.

Petrol is supplied by gravity from fixed tanks of definite capacity, and ignition is performed whenever possible by the magneto belonging to each engine, or, if coil ignition is to be fitted, by the contact maker and distributor to be used afterwards.

The first run after the engine is assembled is made in another part of the works and consists of driving it from shafting, while the cylinders are fed with crocus powder of a very fine grade. This running-in lasts for one and a half hours, and the rings are a thoroughly good fit at the end of this time. After washing with paraffin the assembling is completed and the engine passes to the testing shop, where it is set up and run at different speeds for four hours. During this first run no power readings are taken, and it is merely watched to see that no bearings are running hot, that there are no knocks, and that the carburettor and ignition gear are in working order. After the rough test another hour's run is made, during which b.h.p. readings are taken, together with the other particulars mentioned below, the engine test book having the following headings, well spaced out :—

No.	5th Cylinder.
Date.	6th ,,
Bore.	Temp. of Water Inlet.
Stroke.	Temp. of Water Outlet.
No. of Cylinders.	Gallons Water in Secs.
Compression Ratio.	H.P. lost in water.
Drawing No.	Gravity of Petrol.
Exhaust open.	Gallons Petrol in Secs.
Exhaust shut.	Pints per H.P. hour.
Inlet open.	Temp. of Testing Shop.
Inlet shut.	Barometer.
Revs. per Min.	Moisture.
H.P.	Carburettor.
1st Cylinder.	Make of Coil.
2nd ,,	Make of Magneto.
3rd ,,	Remarks.
4th ,,	

The purpose of many of these observations is obvious, but the remainder require a certain amount of elucidation. Firstly, the observation of the time of each valve opening and closing serves as a check upon the erecting. Secondly, the readings for single cylinders are only made in special cases. Thirdly, the barometric pressure, the humidity of the atmosphere, the temperature, and the specific gravity of the petrol are noted with a view to future investigations as to their influence on the behaviour of the internal combustion engine in general, and White and Poppe engines in particular.

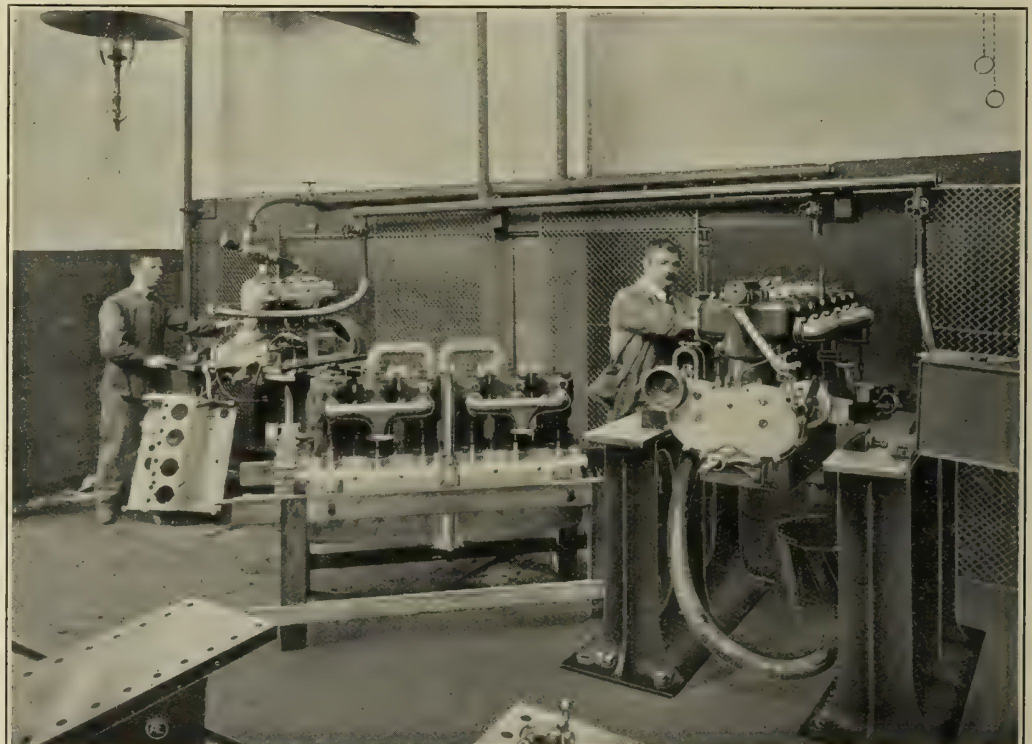


Fig 1.

Mr. Poppe has for the last three years been making an exhaustive series of tests with a single four cylinder engine, which is run and tested for power, several times each day. It has been found that the carburettor requires widely different adjustments at different seasons and at different hours of the day, to obtain approximately the same b.h.p. at the same number of revolutions per minute, but the effect of atmospheric variations has not yet been deduced satisfactorily, though sufficient has been done to prove that they have some effect. The special carburettor fitted to this engine has a series of air intake sleeves of different apertures, which are interchangeable and are numbered. At each run, these are tried in succession till the nearest approach to the average performance (b.h.p. and speed) has been obtained.

It has been found that it is possible to obtain greater power at definite speeds of revolution under certain atmospheric conditions than under others, and it is by careful observation of temperature and weather conditions while all tests are being made that Mr. Poppe hopes eventually to be able to deduce their quantitative effect upon carburation.

It is not too much to say that such laboratory proceedings are likely to be of great value when the summary of years of observation come to be made, and if more manufacturers would take the trouble and go to the comparatively small expense of gathering such data, it could not fail to help towards the more clear understanding of the internal combustion liquid fuel engine, and so to its still greater improvement.

It has been questioned whether it is advisable to test engines for maximum power only, or for maximum power and for slow running with power at a single low speed, or whether it is better to test for power at a great variety of speeds so as to obtain a power curve from the lowest to the highest speed. White and Poppe engines are tested for power principally at a rate of revolution corresponding to a piston speed of one thousand feet per minute, and another power reading is also taken at a higher speed. If desired, it is

easy to obtain another reading or series of readings at different speeds, because the fan dynamometers used are calculated in accordance with a formula based on experiment. There is no doubt that the fan dynamometer is very convenient for ordinary test shop use, as it reduces the necessary observations to a mere reading of the speed of revolution, it has no moving parts and no wearing parts, it is cheap to make, and it is easy to fit.

The fan type of dynamometer working in air is, of course, used by a number of manufacturers, but its accuracy is often doubted, and it is frequently assumed that it is necessary to calibrate every specimen against some other form of brake, if it is to be at all satisfactory. All the fans used by Messrs. White and Poppe are supplied with a standard length of arm, but the blades are detachable and interchangeable. The area of blade used varies with the nominal power of the engine, and is calculated so as to absorb the amount of power which it is desired to obtain from the engine at the rate of revolution corresponding to a piston speed of one thousand feet per minute.

The formulæ given below have been found by Mr. Poppe by experiment and have been tested extensively.

To follow the method of calculation it is necessary to refer to Fig. III. Here it will be seen that :—

R = radius to tip of blade, in decimetres

r = radius to centre of blade in decimetres = 5·5 dem.

a = side of square blade in centimetres

Also let

A = area of blade in square centimetres

$$N = \frac{\text{Revolutions per minute}}{1{,}000}$$

Then $$\text{HP} = \frac{\text{A} \times \text{R}^3 \times \text{N}^3}{4010}$$

And $$\text{A} = \frac{\text{HP} \times 4010}{\text{N}^3} \times \frac{1}{\text{R}^3}$$

Or $$\text{A} = \frac{\text{K}}{\text{R}^3} = \frac{\text{K}}{(\text{r} + \frac{\text{a}}{2})^3} \quad \dots \quad \dots \quad (1)$$

Where $$\text{K} = \frac{\text{HP} \quad 4010}{\text{N}^3}$$

Also $$\text{A} = \text{a}^2 \quad \dots \quad \dots \quad \dots \quad \dots \quad (2)$$

Now to obtain K we require to know the horse power desired at a particular speed. Suppose that we want 45 H.P. at 1,110 revolutions. Then :—

$$\text{K} = \frac{45 \times 4010}{(1 \cdot 11)^3} = \frac{45 \times 4010}{1 \cdot 368} = 131{,}900$$

Having obtained K we can obtain the value of A, from the equations (1) and (2), by giving " a" such a value that the area " A " will work out the same from both formula.

As an example we give the method of calculation of the size of blade suitable for testing an engine with a bore of 115 mm. and a stroke of 135 mm. according to the above formulæ.

Normal revolutions* =

$$\frac{15{,}000}{\text{Stroke in cm.}} = \frac{15{,}000}{13 \cdot 5} = 1{,}100$$

*[These equations apply to White and Poppe engines, but not necessarily to any others.—Ed.]

Revolutions where constant torque collapses * =

$$\frac{15{,}000}{\text{Bore in cm.}} = \frac{15{,}000}{11 \cdot 5} = 1{,}300$$

b h.p. at normal revs. $= \dfrac{(\text{D}-2)^2\text{n}}{8}$

Where D = bore in cms. and n = number of cylinders

Then $$\frac{(9 \cdot 5)^2 \times 4}{8} = 45 \text{ PH}$$

B HP at maximum revs. =

$$\frac{(\text{D} - 2)^2\text{n}}{8} \times \frac{\text{S}}{\text{D}}$$

Where S = stroke in cm.

Then $$45 \times \frac{13 \cdot 5}{11 \cdot 5} = 53 \text{ HP}$$

for a base line, and H.P. for the vertical line of Fig. II., then the H.P. diagram for a series of blades comes out as shown.

The engines, besides being tested for power, are also tested for fuel consumption, as indicated by the reproduction of a test book sheet. To obtain this value the engine is run at a constant speed, while the contents of a one pint petrol tank are exhausted, the exact time occupied being taken. Also the calculation of the power lost in the cooling water is found to show at once if there is anything abnormal in the ratio $\dfrac{\text{petrol used}}{\text{power given out}}$

The principal tests to which every engine is subjected are :

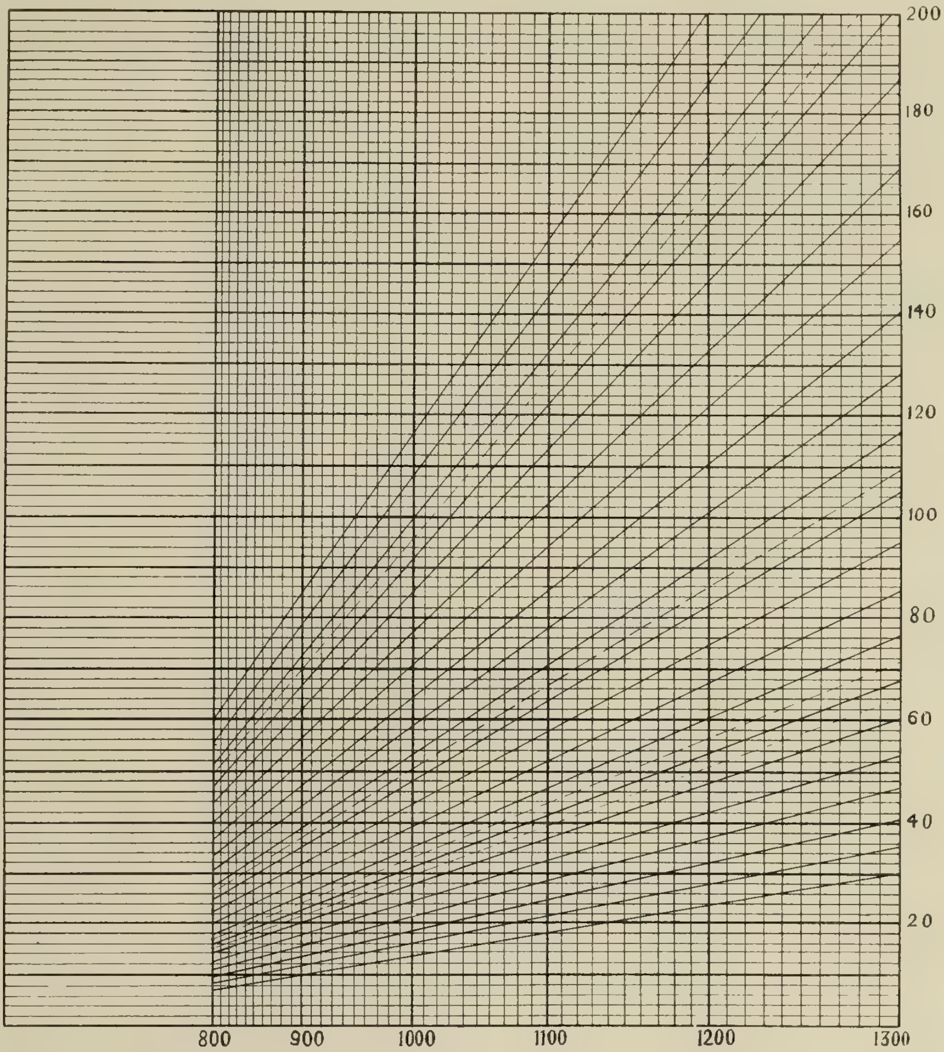


Fig. II.

From these two desired values two corresponding values of K, namely, 131,900 and 96,737, can be obtained. Then taking r=5.5 decimetres, a=21.6 for normal revolutions, and 19 for maximum revolutions. These two values of a give the dimensions of the blades, which will absorb the requisite power at the desired number of revolutions per minute, but it is possible to test an engine over a small range of varying speeds with one set of blades only. The power absorbed by any given pair of blades can be obtained from the equation

$$\text{HP} = \frac{\text{A} \times \text{R}^3 \times \text{N}^3}{4010}$$

and if we then take

$$\text{N}^3 = \left(\frac{\text{Revs. per min.}}{1{,}000}\right)^3$$

1. Two power tests at different speeds, with different fan blades.

2. Consumption test at constant speed.

3. Power lost in water at constant speed.

The H.P. lost in water is, of course, found by passing the contents of a tank of known capacity through the jackets while the engine runs at an observed constant speed; measuring the temperature of the water in the tank, and the temperature of the water as it leaves the engine, and taking the time occupied in emptying the tank.

This system of testing has been in use by Messrs. White and Poppe for several years, and has been found to give extremely satisfactory results, as it enables engines to be dispatched from the works in guaranteed tuned up condition. It should be noticed that there is no obser-

vation to be made that is not easy, the most intricate being the taking of the

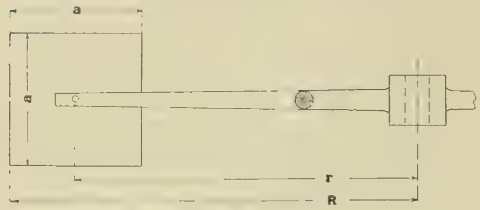


Fig. III.

temperature of the water in the last test. Reading the speed in making H.P. tests, observing the time taken to consume the pint of petrol at constant speed and fitting the fan blades suitable to the type of engine being all the simplest of simple operations. Much more simple than the taking of H.P. by the usual Prony brake with a rope and a spring balance and a revolution counter.

The method of calculation of the dimensions of the fan blades may not be quite clear, as they are not mathematically deduced, but are entirely the result of experiment. All who have been concerned with the design of air propellers for aeronautical work are aware of the impossibility of calculating the exact shape and exact dimensions of a propeller to behave in any particular way at any particular speed. This is due to our lack of knowledge concerning the physical nature of air disturbances. Mr. Poppe has, however, found that by the use of a single pair of blades of thin sheet metal set at right angles to their path, the power absorbed varies in accordance with the formulæ given, within fairly wide limits of speed, and with a total diameter of propeller not far removed from the dimensions given. At present a mathematical explanation of the formulæ is wanting.

THE COMMER CAR POWER SYSTEM.

A description of the engine and transmission gear used on the industrial chassis for from four to seven ton loads.

BY the courtesy of Mr. C. M. Linley, who is mainly responsible for the design of the various types of industrial chassis made by Commercial Cars, Ltd., we are able to describe some of the features common to the majority of them. It is scarcely possible to select a single complete model, as variations have to be introduced to suit the requirements of the different users wherefore wheel-bases, sizes of wheels and tyres, and even more vital dimensions than these have to be capable of variation. The engine, with all its accessories, and the transmission gear, however, are standard for all models within fairly wide limits, and the chassis for a load of seven tons only differs from one for a four ton load in that the final speed reduction on the chains is different, and the frame and other parts are, of course, likewise proportioned to the weight they have to sustain. There are many small details throughout the chassis which exhibit considerable ingenuity, but they are mostly contained in either the engine or transmission, so in confining our attention to these parts but little of interest will be lost.

The engine is shown in Fig. I., and doubtless the observer will at once notice the ball bearing crankshaft, with, perhaps, a certain amount of surprise. As its use affects the design of the whole engine, it will be well to commence with it. The principal reason for the use of ball bearings is to reduce frictional losses, but this has not had much influence on their selection for the engine in question. It must be remembered that a heavy industrial vehicle is often run as fast as it can be made to travel, and being geared very low, this means that the engine speed may be extremely high, for long periods of time. Thus in ascending a hill a waggon may be driven on a low gear for a mile or more with the crankshaft turning at as much as two thousand revolutions per minute. This means that if plain bearings are used they must either be very large or exceptionally well lubricated. Large size necessitates a long engine, and oil fed under high pressure is likely to give rise to over-lubrication of the cylinders, with high oil consumption. Also, we are informed that many owners of Commer Car chassis do their own repairs and overhauling, and that as this work is often performed by unskilled workmen, there is considerable danger that they will take up plain bearings too much, and so bring about a seizure. On the other hand the ball bearings used are claimed to perform satisfactorily, and their replacement when worn is neither difficult or capable of being done in a wrong way.

That ball bearings of the size used make considerable noise cannot be denied, but this is a matter of small moment on a traction vehicle. The cost of such large bearings is one excellent reason against their use, and it is not easy to believe that they can be more durable than a plain bearing of sufficient dimensions, but if everything is taken into consideration there is probably not much to choose between the advantages and disadvantages of both types of bearing for the particular purpose of an engine which must be able to run at very high speed without seizure, and also the fact that ball bearings use but little oil enables the running cost of a waggon to be reduced; very slightly perhaps per mile, but perceptibly per year.

That the durability of the crankshaft bearings is satisfactory is due to three things—their size, the method of their fitting, and the provision of thrust washers which prevent the smallest end stress being taken by the main races. The largest bearing, at the flywheel end, is $7\frac{3}{4}$ins. in diameter, the centre bearing is the same size, but the forward end bearing is 5ins. The balls are $\frac{3}{4}$in. and $\frac{5}{8}$in. respectively. Various makes of bearing are used, and the chief point of failure seems to be the cages. The longest life is to be obtained from bearings with the largest possible number of balls, but as this is increased the strength of the cage decreases. For this reason some special bearings have been made with solid cages. These perform well, but owing to the number of slots which have to be cut in the side of the race to enable the bearing to be assembled, they are somewhat more noisy than the other type with a single point of entry and a two-piece cage.

It will be noticed that the bearings are not secured to the shaft, being merely a very light driving fit on it. The outer races are each fitted into malleable iron housings, the centre one simply fitting tightly into the aluminium, and the two end ones being attached by bolts. These malleable housings must have a beneficial effect on the durability of the bearings, as they have sufficient area of contact with the aluminium to sustain the load upon them without danger of their working slack, whereas if the ball bearings were set directly in the aluminium, they would certainly lack the rigidity which is essential.

The big end bearings are lined with white metal, and are lubricated by a trough splash system. The large dimensions of the dippers is noteworthy, and these are made solid with the bearing caps, being drilled out and connected to a hole leading to the bush. All other parts of the engine depend upon the splash from the dippers, and the cylinders are not protected. The oil pump is of the ordinary gear pattern, and delivers to a glass-fronted tank on the dashboard, whence it runs by gravity to the troughs. This tank is used to form a sight feed, which cannot escape the notice of even the most careless driver, as failure of the oil supply is at once obvious. The pump sucks through a coarse filter consisting of a brass cylinder, about four inches long, and an inch in diameter, and this can be withdrawn for cleaning without the loss of more than a few drops of oil. It fits in a chamber bored in the bottom of the sump, and is secured by an external flange and a couple of fly nuts. At the back of the channel in which it fits is a conical valve with a spring behind it, and as the filter is withdrawn the valve follows it and so closes the hole in the sump. The pump is withdrawable in a downward direction, and the division of the driving spindle is shown in Fig. I. There is a skew gear driven off the camshaft and fixed to a short square-ended shaft; the pump has a similar shaft, and the two are joined by a tube with female square ends, very much resembling a small tube spanner.

The camshaft mounting is also made clear by the figure, and it will be seen that it is provided with a central phosphor bronze bearing and a ball bearing at each end. The use of the latter seems somewhat purposeless, as plain bushes would be quite as durable, and considerably cheaper. The tappets are mounted in an unusual way, being carried by the cylinder castings, and not by the crankcase. Each cylinder foot is bored to take the two tappet bushes, and the latter are flanged at the top. A dog secured by a stud, also set in the cylinder foot, secures the tappet bushes, and thus when a pair of cylinders is removed, the whole of the valve gear comes away with them.

The concentricity of the valve guides and the tappets is also ensured by this form of construction. The section of the valve heads showing how the under side is recessed away from the seating should be noticed, as this, doubtless, has a beneficial effect on the life of the valve.

The cylinders themselves exhibit no peculiarity except, perhaps, that they are cast with open tops, which are closed subsequently by cast-iron caps. These caps are provided with a projection on the under side, which prevents the piston being pushed up too far when assembling and thereby allowing the upper piston ring to expand into a valve pocket. The plug caps are hollow, and so proportioned that their thickness at any point is about the same as that of the cylinder walls, this being claimed to ensure even expansion and contraction, and so to remove any tendency for a plug to loosen.

The pistons are cast with a pair of arched webs supporting the heads, and there is left a slight projection at the point where the webs intersect. The purpose of this is to form a point whence oil may drip, and the small end of the connecting rod is drilled in such a position that it may catch the drops. This system has the disadvantage that any burnt oil or encrustation which may break away from the under side of the piston head can easily stop the oil hole.

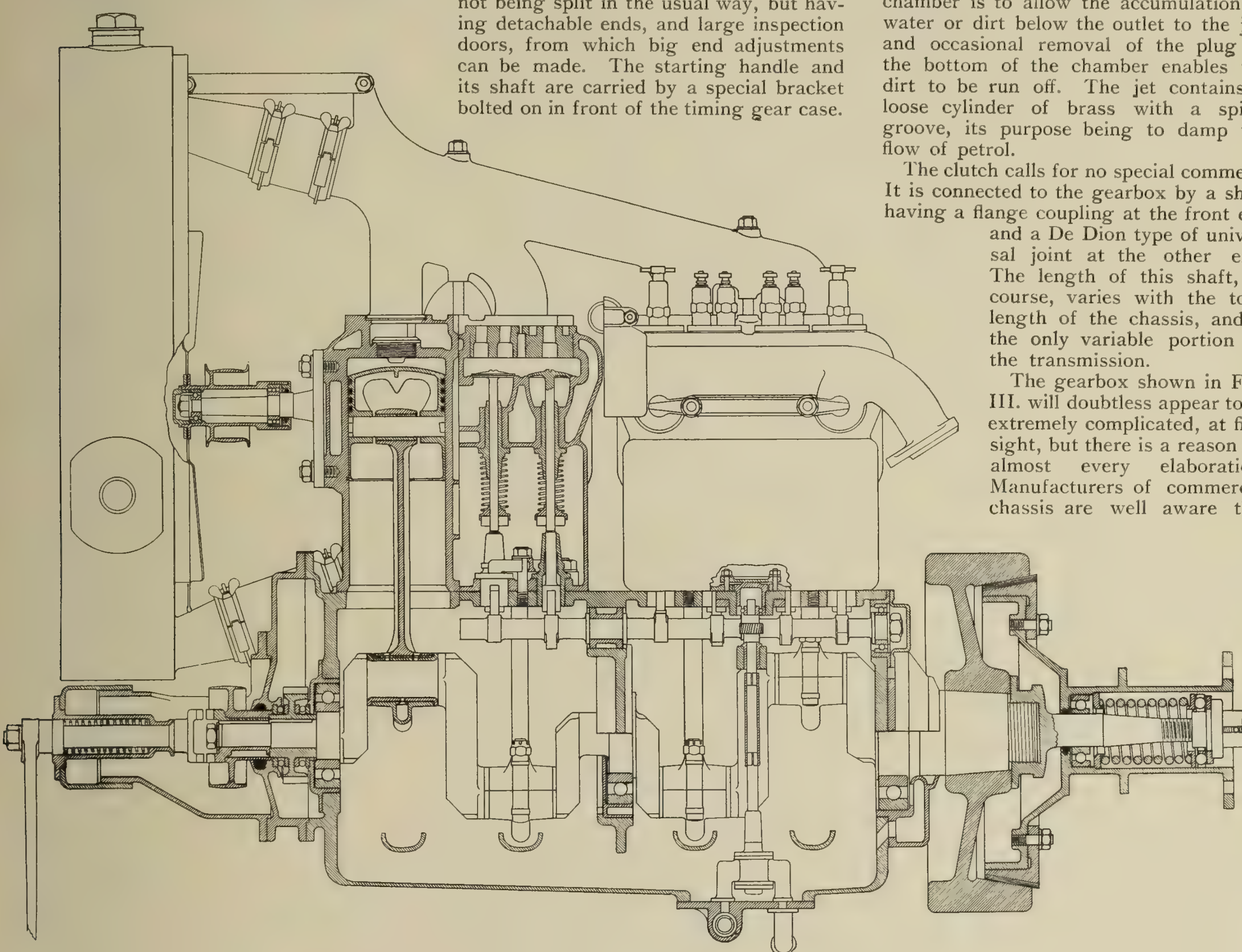


Fig. I. **Scale 1 in. = 7·6 ins. approximately.**

The pistons are comparatively light for their size, weighing a few ounces over four pounds, and they could be lightened still more by dropping one of the four rings, which could be done without any loss of efficiency. The gudgeon pins are secured by special split taper pins, a flat being cut at each side of the piston, where the ends of the gudgeon come through, and the taper pins being driven through and then slightly opened. As the gudgeon itself is a tight fit in the piston, this makes a secure fixing, and is also a cheap form of construction.

The crankcase is a one-piece casting, not being split in the usual way, but having detachable ends, and large inspection doors, from which big end adjustments can be made. The starting handle and its shaft are carried by a special bracket bolted on in front of the timing gear case.

The cooling circulation is natural, and the special clips used to secure the large rubber joints are deserving of special notice; they are made from brass castings, the water pipes being of the same material. The radiator is tubular, with a brass tank, and is trunnion mounted in the frame. The carburettor is carried on the off side of the cylinders, and draws all air from a box on the exhaust pipe, seen in Fig. I. The end of the engine exhaust pipe is connected to the silencer pipe by the flange shown, but the joint is not made with the usual packing. The end of the engine pipe is cup-shaped, and the end of the silencer pipe has a corresponding spherical extension, so that a small amount of relative movement does not affect the tightness of the joint. On a long chassis with solid tyres this construction is found to do away with blowing at this point.

The carburettor, Fig. II., is of a simple type, and is designed for economy rather than for other considerations. All air enters below the jet, and is drawn from the neighbourhood of the exhaust pipe, though there is an oven type of ventilator disc in the intake pipe, and this can be opened in hot weather. By detaching the U-shaped clip, the jet and float chamber can be removed without any other disconnection except that of the petrol pipe union. The great depth of the float chamber is to allow the accumulation of water or dirt below the outlet to the jet, and occasional removal of the plug at the bottom of the chamber enables the dirt to be run off. The jet contains a loose cylinder of brass with a spiral groove, its purpose being to damp the flow of petrol.

The clutch calls for no special comment. It is connected to the gearbox by a shaft having a flange coupling at the front end and a De Dion type of universal joint at the other end. The length of this shaft, of course, varies with the total length of the chassis, and is the only variable portion of the transmission.

The gearbox shown in Fig. III. will doubtless appear to be extremely complicated, at first sight, but there is a reason for almost every elaboration. Manufacturers of commercial chassis are well aware that the ordinary type of sliding gear leaves much more to be desired in their case than it does in the case of pleasure cars, because the drivers are usually more careless and also the frequency with which changes of speed have to be made is much greater. The Commer Car gear has been designed to be reliable, even when handled with extreme roughness, and it has been proved to be satisfactory by several years of use.

The box itself is aluminium and is not divided, being closed simply by the forward end plate and a top inspection door. Fig. III. shows a horizontal section. The bearings of the main shaft are carried in malleable iron housings, flanged and secured to the aluminium by bolts. The shafts are of Ubas and Jessop's nickel

steel, and the gears are all Ubas. The dog clutches are undercut so that considerable force is needed to disengage them when loaded (the purpose of this being partly to ensure secure grip, and partly for a reason to be explained later).

The main driving shaft has a square portion at its inner end, and on this slides a clutch which

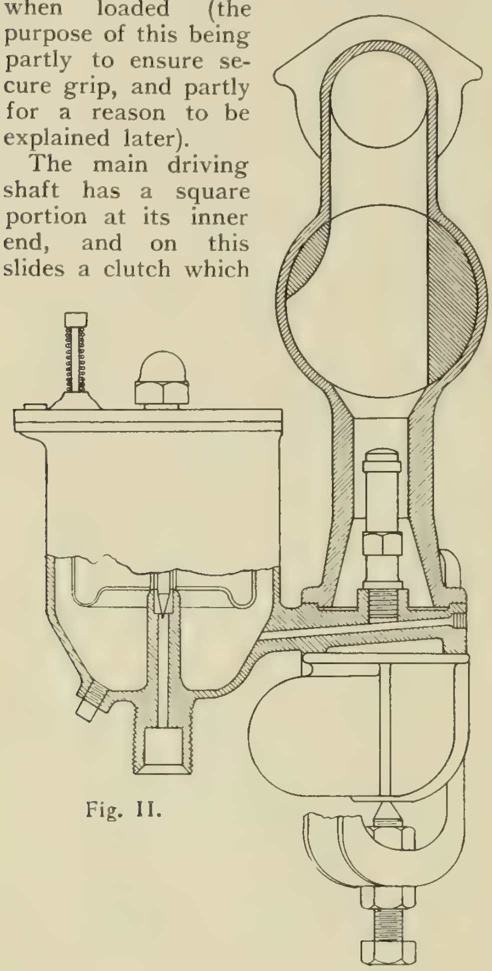


Fig. II.

enables the shaft to be secured to either the secondary shaft driving pinion, or the end of the main shaft. Looking at the rear end of the camshaft the direction of revolution is clockwise in going from neutral to reverse, and anti-clockwise for the other speeds. The clutch at the front end, of course, gives the direct drive, and also makes connection to the secondary shaft for the two lower speeds. The central clutch gives the first and second speeds, and the rearmost clutch the reverse gear only. The secondary shaft is disengaged between each speed, and the rotation of the camshaft is controlled by a lever working in a quadrant beneath the steering wheel which turns a large skew gear at the foot of the steering column, the skew pinion being on the end of the camshaft, which is provided with the necessary joints.

The gear is extremely easy to handle owing to the spring construction of the striking levers, which is shown in Fig. IV. To change speed the hand lever can be moved over with no other effect than to compress one of the springs, the undercut of the clutches enabling the latter to hold in engagement as long as any power is being transmitted. With the lever already moved over, a touch of the clutch pedal enables the dog clutch to release, and the spring throws it over into engagement on the other side. This means that it is impossible to disengage a clutch without releasing the main clutch, and the springs not only enable changes to be made very easily, but reduce the danger of the clutches being forced against each other while they are running, with any greater force than that which the spring can exert. The use of ball bearings for the camshaft is a somewhat debatable point, as is their use for the engine camshaft. The saving in friction can scarcely be considerable in either case; still, the ball bearings are only objectionable on the score of their cost, and not for mechanical reasons.

From the gearbox the drive passes through a cushion coupling to the countershaft, which is a separate unit. This cushion drive is explained by Fig. V., and it is only necessary to add, that the cushions are pieces of ordinary rubber tyre. The normal points of contact between the two portions are at the inner ends of these tyre sections, and as the power is applied the areas of contact increase radially outwards. This enables the utmost advantage to be taken of the elasticity of the rubber and the increasing radius also helps to reduce the shock of engagement. The pads are claimed to be very durable unless out of line.

The countershaft and the final drive are shown in Figs. VI. and VII. The casing of the countershaft is all malleable iron, the revolving differential case is also malleable, and the part carrying the differential pinion shafts is of cast steel. This part is not the usual form of spider, the outer ring being cast in one piece with the centre and bored to take the steel pinion shafts. The cap nuts which secure the inner races of the ball bearings to the differential box are interesting because their shape is accounted for by the fact that the phosphor bronze bushes between the differential case and the driving shafts exhibited a tendency to move endways at a time when the cap was not provided.

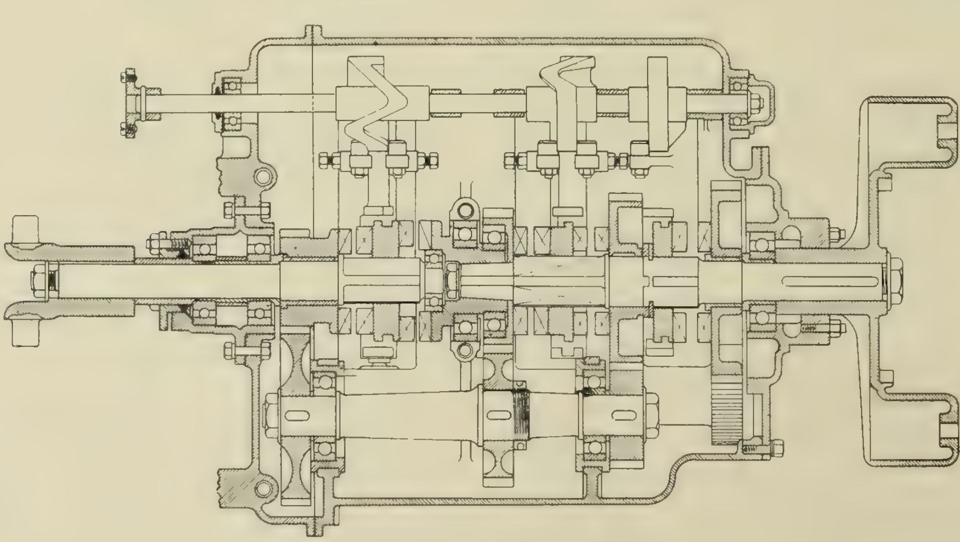


Fig. III. Scale: 1 in. = 7·5 ins. approximately.

The thrust washers which back the bevel pinion and crown wheel are peculiar

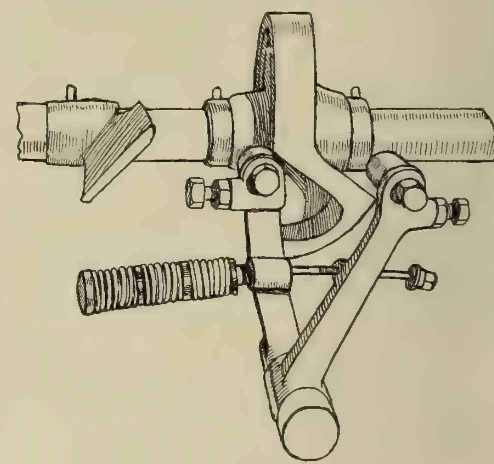


Fig. IV.

in that the races are made of high speed steel, and very exceptional durability is claimed for them on this account, in fact we are informed that no wear whatever has taken place after more than a year's use under ordinary conditions, while other thrust washers of similar dimensions were almost worn out in a similar period of time. The large spigot bearing on the pinion shaft should be noticed, as it is an excellent point.

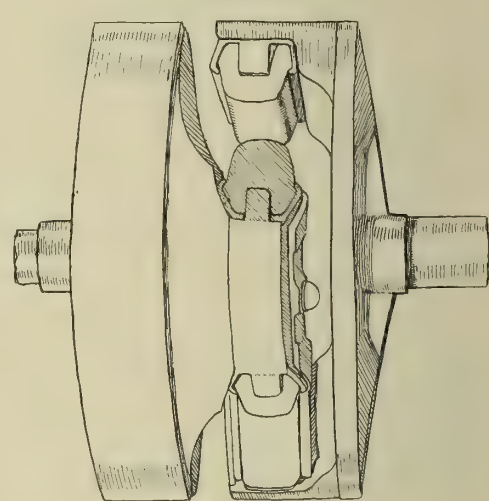


Fig. V.

The final drive is almost fully explained by Fig. VII. The chain case is aluminium with centre plates of malleable iron, and the method by which these are secured is noteworthy. The rear plate is rivetted to the case, but the rivets are set in an inside steel ring so that the aluminium is pinched between the malleable and the steel. The front end plate is attached by studs and clips, it being eccentric in the aluminium for chain adjustment purposes: there are no radius rods other than the chain cases. The axle is, of course, solid, and detachable wheels are used to enable tyre changes to be made without affecting the hubs. To assist the dismounting of the outer parts of the wheels the long cheese-headed screws are provided, tapped ordinary Whitworth thread, so that they can be removed and a long bolt be used to force the outer shell and inner hub apart. To prevent egress of oil from the cases, and also to prevent ingress of water, there are thrower rings formed in the case, and these are said to be entirely effective. In the face of the cases there are inspection

doors secured by wing nuts, to enable the condition of the chains to be observed.

The brakes are operated as usual, the foot brake being on the countershaft, and

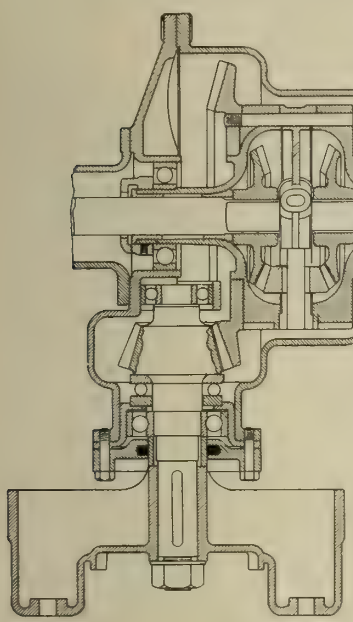


Fig. VI.

the hand brakes in the rear hubs. There is no peculiar point about the brakes themselves, but the compensation gear and adjustment of the hand brakes is distinctly interesting, and is shown in Fig. VIII. The effort from the hand lever is transmitted to the central case by the tube, and movement of this case causes movement of the worm wheel, the worm being secured to the case and incapable of rotation. The length of shaft between the worm wheel and either brake arm being equal, the pull on both brake rods is equalised, and also by turning the worm, an easy adjustment of the brakes is made. There is only one objection to this arrangement, and that is, again, its cost, but there can be little doubt about its effectiveness, though it cannot com-

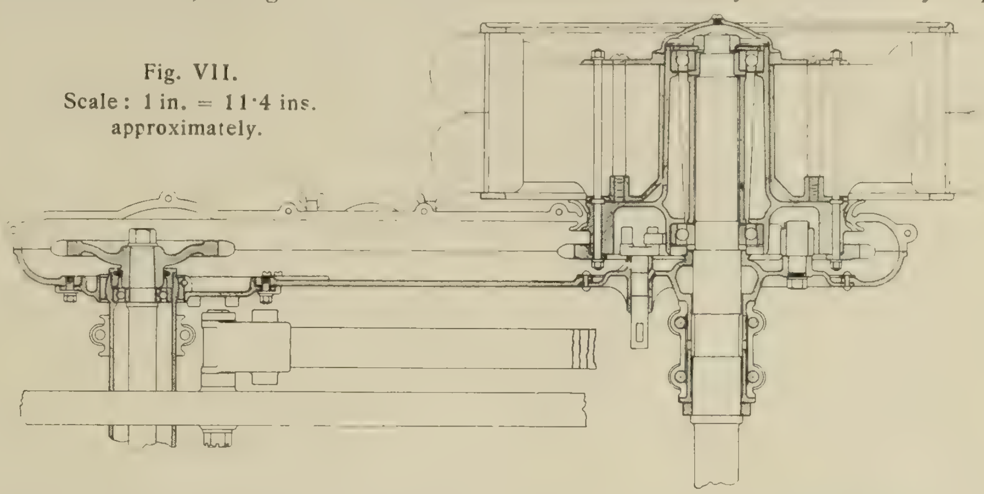


Fig. VII.
Scale: 1 in. = 11·4 ins. approximately.

pensate for unequal adjustment of the driving chains, which may set the back axle out of parallel with the brake shaft.

The designs we have described are difficult to criticise (although there is a good deal about them that is fit subject for debate), because most of their peculiarities are due to the special conditions under which the chassis have to work. Their outstanding feature is the number of the ball bearings used, and, expense apart, this is only to be avoided on account of noise, which is a small matter on an industrial vehicle. The ball bearing crankshaft we have already dealt with, and the only other point which is likely to be noticed is the large number of parts, especially in the gearbox. It is not too much to say that but few designs exhibit so little regard for cost of production, for it must be remembered that the quantities are not exceptionally great, although the same parts are used for many different types of vehicles. Effort appears to have been directed entirely to improvement of durability as indicated by experience, and alterations to have been made as often as they appeared advisable. Throughout the chassis the attention to

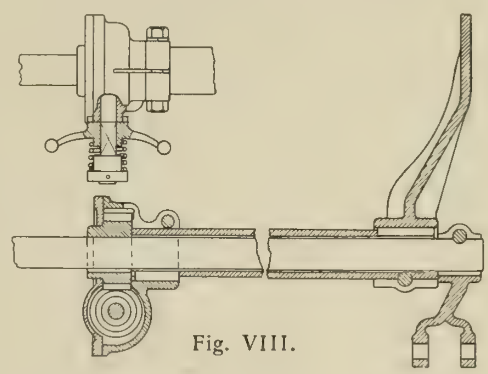


Fig. VIII.

detail is commendable, and is as noticeable on other parts as on the power system with which we have dealt.

A PLEA FOR MONOBLOC CYLINDERS.

Some special points which should be observed in their design, and a description of a particular method of manufacture.

By Arthur W. Reeves.

Contrary to the opinions expressed regarding the monobloc system of cylinder casting in the article on cylinder machining published in the July issue of *The Automobile Engineer*, the writer considers that the monobloc system of cylinder construction constitutes a real advance in automobile engineering. At the same time it must be frankly admitted there is both a debit and credit side to the account, and in the following article consideration will be given equally to both bad and good features, in the hope that a fairly wide experience of the problem may be of some assistance to those readers who may still be undecided in their minds.

Probably the first question to be settled when setting out to design a monobloc engine is as to the employment of a two, three, or five bearing crankshaft. This is, of course, entirely dependent on the type of the car to be produced and the selling price. The two bearing and three bearing crankshaft arrangement is quite satisfactory (as far as the cylinder is concerned) when the limits of the cylinder bore does not exceed about 65 mm., but it does not lend itself happily to the attainment of uniform temperature of the walls of the cylinders and jacket, which is a very important point. Figs. I. and II., representing designs in actual practice, will make this obvious. Also it will be noticed that the valve areas and the jacketting spaces surrounding the valves are restricted in size. There are considerable modifications from the designs illustrated, but the same drawbacks are apparent in each instance which has come under the author's notice. There is little doubt that to obtain maximum satisfaction from the thermo-syphon system of circulation the cooling water should have contact all round each cylinder barrel, thus obtaining, as nearly as possible, uniform transference of heat. It may be observed that in one particularly satisfactory arrangement of a two bearing crankshaft with monobloc cylinders (the 8-10 h.p. De Dion) pump circulation has been employed, and the writer's experience confirms the wisdom of the practice, so far as this particular design of cylinder is concerned.

Coming to what is really the ideal, but necessarily the most expensive, arrangement for monobloc cylinders, i.e., a crankshaft with five bearings, we have a combination of sound engineering practice, as far as the crankshaft is concerned, with the very best possible arrangement of cylinder cross section, resulting in a highly efficient thermo-syphon cooling arrangement, ample valve and jacket areas, and freedom from foundry and machine shop troubles. Considering Fig. III. as a whole, the absence of any sudden changes in section will be at once apparent, and the observance of this point is as conducive to satisfactory cooling by the thermo-syphon system as it is to the production of sound castings. It may be argued that the arrangement produces a longer cylinder casting, and that consequently expansion troubles may be increased both in the mould and when the cylinder is in service. In practice the reverse is actually the case, and it would appear that the essential feature to guard against is not so much expansion in itself as *unequal* expansion: it is in this respect that the designs shown in Figs. I. and II. fall short of the ideal.

Before going further we ought now to consider the advisability or otherwise of passing the gas intake ports through the

water jacket. Of course, in the case of Figs. I. and II. it is impossible to do this, and therefore the induction and exhaust piping will all be on the same side of the engine, in which case full advantage of the possibilities of the monobloc system is certainly not taken.

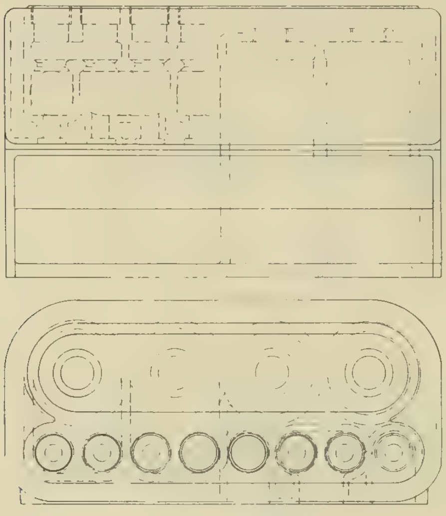
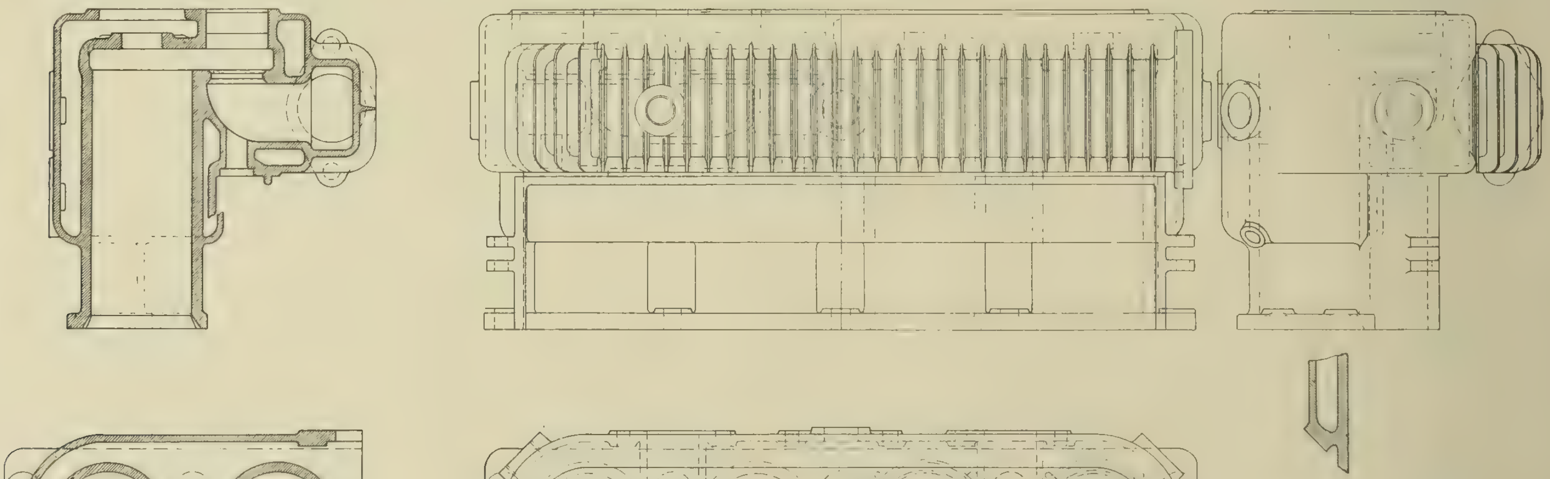


Fig. I.
Cylinders for two-bearing crankshaft.

In view of the difficulty of preventing condensation in induction pipes, it is certainly a very great advantage to have them water jacketted, always providing that the heating is not overdone, and that the location of the pipes or passages is fixed with discretion. The author is aware of an instance in which the induction pipe was carried through the water jacket but, owing to its position being practically on a level with the top of the cylinder, and therefore nearly at the point



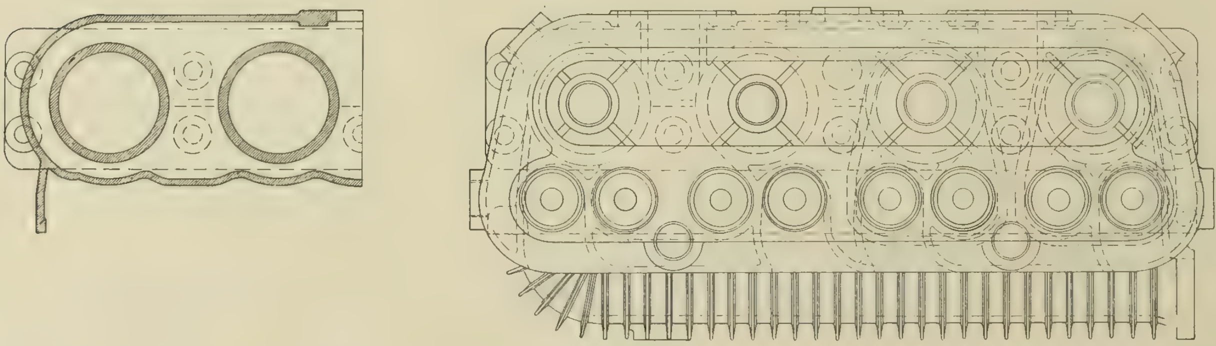


Fig. III.
Cylinders for five-bearing crankshaft.

of maximum temperature, the results were very poor, petrol consumption being extraordinarily high and efficiency correspondingly low. It is impossible to fix any hard and fast rule, as so much depends on the efficiency and type of the cooling system, but the author's practice is to position the intake ports on a slightly lower plane than the valve seats, as shown at A in Fig. IV. This, in conjunction with a satisfactory thermo-syphon system of cooling, is found to give all-round good results.

As nine-tenths of monobloc cylinders are arranged with all the valves on one side, necessitating but a single camshaft, the succeeding remarks will be relative to

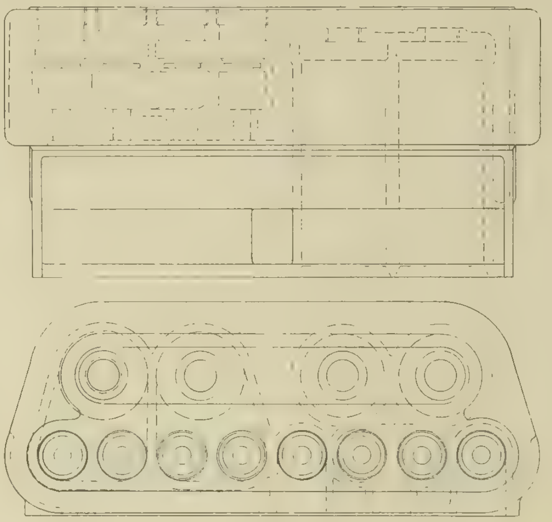


Fig. II.
Cylinders for three-bearing crankshaft.

this method of construction. A few makers employ the system with the valves in the cylinder head operated by an overhead camshaft, but there appears nothing specially difficult in this arrangement to warrant our giving it more than passing attention.

As to whether the exhaust outlet pipe should or should not be cast *en bloc* with the main casting, it is without doubt a fact that very many designs have been rank failures owing to the practice of casting these parts in one, the failure being due to the greater expansion of this pipe or passage when in use, as compared with the jacket of which it is a part. The remedy is obviously to limit this difference in temperature, and this is the reason for the vertical cooling fins shown in Fig. III. The total cooling area of these fins is so large and their action so efficient, that the temperature of the exhaust pipe as a whole never approaches a degree likely to give trouble, even at the highest speeds and loads, and in practice not one single waster has been due to this cause. It will be observed the cooling webs take a good hold of the water jacket, thus assisting the radiation of heat and conducing to a balance of temperatures. The contention is, therefore, that if properly arranged, of generous area and suitably cooled, there is no difficulty in making the exhaust pipe in one with the cylinder, thereby saving the extra cost of making it loose and attaching it by suitable fixings.

A point worthy of mention is, that if at all possible, the usual ribs tieing the water jacket to the cylinder barrel should be omitted. From the section of Fig. IV. their absence will be noted, and it prevents the formation of hard spots in the cylinder, which are well known to be a prime cause of untrue boring, and their absence has not caused the author any trouble. As examples, the Vauxhall and Crossley firms each employ the system with five bearings to the crankshaft, but the former company prefer to make the exhaust pipe as a separate piece.

It may be asked what is the limitation in size of monobloc cylinders, and the reply is that it is by no means a question of manufacture, but entirely one of ease of handling. The author's opinion is that when five crank bearings are required the limit is reached with four cylinders of about 85mm. cylinder bore. Above this dimension there is no doubt that the question of handling becomes a serious one—outside the manufacturer's works, and as far as the car owner is concerned. To carry the system as far as six cylinders has absolutely nothing to recommend it, it being from every point of view preferable to have two blocks of three cylinders.

Let us now consider the question from the standpoint of the user. His whole complaint is wrapped up in the magnitude of the task in front of him, if for any purpose he requires to expose his

pistons. In a small engine of, say, about 60 mm. bore such as the 10 h.p. De Dion, there is really little to complain of, and a man of sufficient ability to under-

Fig. IV.

take the job at all, can replace his cylinders with trifling assistance to guide the piston rings into their places. The whole operation is almost, if not quite, as quick as if the cylinders were cast in pairs, and the weight being but small the physical effort required is trifling. When the larger sizes are reached, there is more in the argument, for to take off and replace a monobloc cylinder casting of, say, 80 or 90 mm. bore, does unquestionably call forth reasonable complaint on this score. But one rarely gets something for nothing, and it must be asked if the advantages to the customer of great rigidity, absence of detail parts, and improved cooling, accessibility, and neatness are not more than worth the slight disadvantages involved, as mentioned above.

The well-known statement that in the event of frost the whole cylinder may be scrapped, is obviously such a weak argument as not to be worthy of very serious attention.

From the manufacturer's standpoint, what are the advantages and disadvantages? The main advantages obviously lie in the decreased cost of production. With a suitable design it can be shown positively that the loss due to wasters is by no means increased. It is, however, perfectly useless to attempt false economy in regard to either design or pattern. The value of the extra rigidity imparted to the engine as a whole cannot be over-estimated, and its reality is made evident by the life of the crankshaft and big end bearings.

Treating entirely of the manufacturer's side of the question, the practice observed

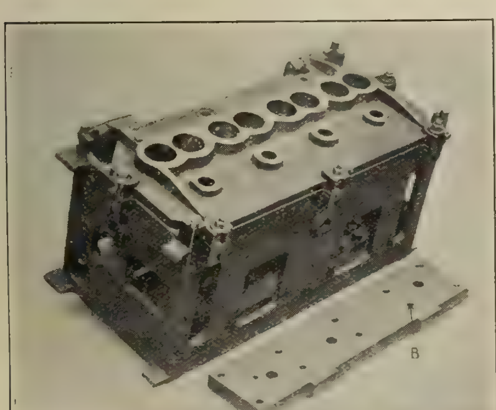


Fig. V.

by the firm of Messrs. Crossley Bros. will be cited as a fairly representative one where monobloc cylinders are a standard. Presuming that the designs have been carried through while bearing in mind the previous portion of this article, the next step is pattern making, and it is impossible to dwell too strongly on the necessity for workmanship and material to be always the very best which can be obtained. Thoroughly well seasoned Cuba mahogany only should be used, and in regard to loose parts the very greatest care should be taken with the locating arrangements—in fact, it is far preferable that these loose pieces should, in every case, be aluminium castings, properly dowled to ensure uniformity of results, and the same remarks obviously apply with even greater force to the core boxes, as they get the roughest handling. It can be assumed that, if proper care be taken, a set of patterns and core boxes will make fifteen hundred cylinders without showing any deviation from standard. Of course, even good patterns in themselves do not ensure good

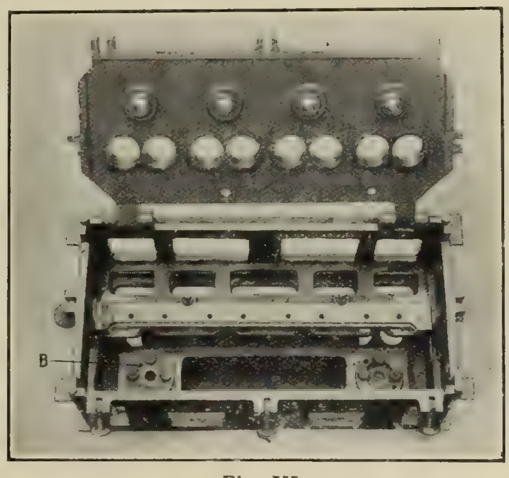


Fig. VI.

castings unless there is equivalent skill and care on the part of the moulder. Assuming this to exist, we still have the quality of the metal to consider, and it is absolutely essential that the latter is as free from sulphur as possible.

The following table is an analysis which should be worked upon, and which gives uniform all round results. It implies the use of exceptionally expensive foundry coke, which raises the cost of the castings proportionally, but unless some such analysis be insisted upon it is more than likely that the thinner sections of the casting will be found to be defective.

ANALYSIS.

Graphite Carbon ...	2.850% to 3.090%
Combined Carbon...	0.460% to 0.550%
Silicon	1.620% to 1.650%
Sulphur	0.058% to 0.070%
Phosphorous ...	0.690% to 1.270%
Manganese ...	0.990% to 1.530%

It is preferable to arrange for a central plug hole in the cylinder head for core supporting purposes, though as most cylinders are now bored on the "snout" boring machine, this is not necessarily of

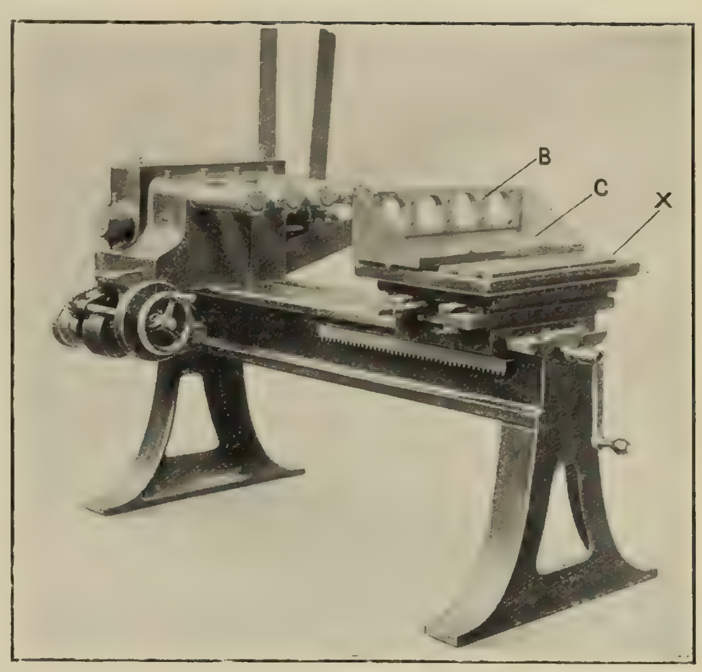


Fig. VIII.

assistance for boring purposes, though it may be so taken advantage of.

When the first sample casting is made it is advisable to saw it up into sections, for the purpose of examination as to the core settings, etc., and care must be taken to see that there are no "fins" of metal likely to cross the water jacket space—a point which often gives rise to much circulation trouble. Regarding relative cost, a monobloc cylinder of the design in question costs approximately 58s. per cwt., as against 50s. per cwt. for the twin casting type, and this difference might be saved in the machining if full advantage is taken of the possibilities of the monobloc system, in this respect.

Assuming the castings have been successfully produced, the machining next demands attention. The outfit of jigs necessary is quite small and simple, as shown by Figs. V., VI., and VII., which represent the whole outfit. (The boring jig is shown in position on the boring machine saddle, Fig. VIII.

After the casting has been milled on its top and bottom and side faces on the

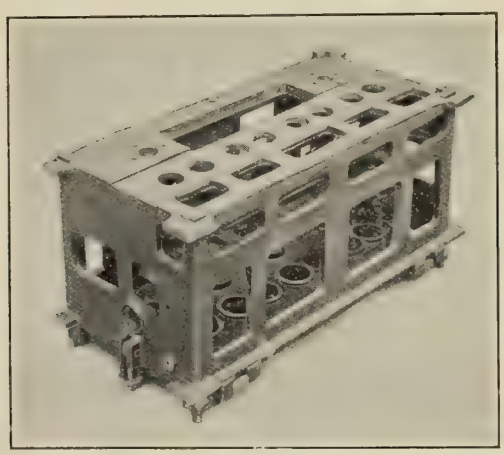


Fig. VII

horizontal mill, the jig shown in Fig. V. is used for the purpose of drilling the holding-down-bolt holes. These holes are bound to be in exact positions relative to the cylinder bores, and are used for locating the casting on the boring jig, Fig. VIII., the bolt holes B on this jig being arranged to correspond. The

bosses C on the jig relieve the holding-down-bolts of all weight. In setting jig, Fig. V., the locating point used is the cored hole of the cylinder bore, thus ensuring equal thickness of

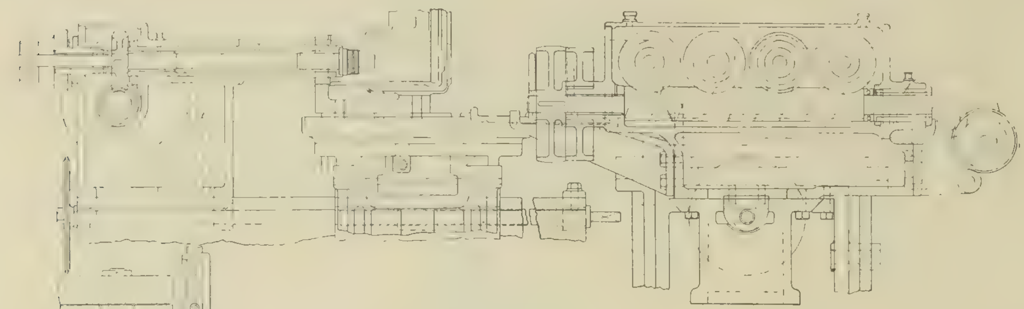
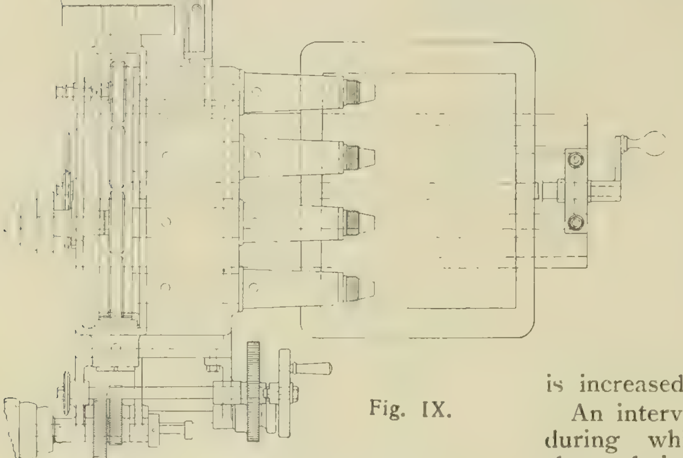



Fig. IX.

metal entirely round the finished bore.

The next operation is that of boring. In this again the monobloc cylinder lends itself to cheapness of production by simultaneous machining of all four bores. The four spindle machine illustrated by Figs. VIII. and IX. was designed and produced under the direction of the author, and fulfils its purpose admirably.

The four spindles are driven by worm gears running in an oil bath, this form of gear being decided upon on account of its steadiness and freedom from vibration. Although several speeds are available for traversing the saddle, the feed in use for roughing is about .095 ins. per rev. of spindle, and for finishing about .014 ins. The cutters operate at approximately 30 ft. per minute, under which conditions there is a minimum generation of heat, and although the machine is of light construction, there is no vibration. The heads are not made adjustable in regard to distance between centres, it being argued that this is an instance in which, for many reasons, a special machine for a special job is preferable to a machine capable of a wider range of duty.

Two traverses are taken through each bore; the first a roughing cut, leaving about .012 to be taken out by the following reamer traverse. The type of inserted cutter used will be readily understood by reference to the illustration Fig. X., and in regard to the final finish it is extraordinarily good if the cutters are kept to a first-class edge. Whilst the author is a firm advocate of grinding the bores, it has to be admitted that the above process leaves them sufficiently good for all ordinary purposes, the engine giving its maximum power within a few hours of its test commencement, but naturally, after grinding, the finish and accuracy is increased very considerably.

An interval of a quarter of an hour, during which time the cutters are changed, is found sufficient to allow between finishing the roughing cut and commencing the finishing cut. If, however, the cutters are kept in condition, the heat generated is very slight. It would be preferable, however, to duplicate the carrier and mount it on a revolving base, to enable another set of cylinders to be set up for roughing out, whilst the preceding set were cooling down, and a considerable saving could be made in this manner.

After leaving the boring machine the cylinder is inserted in the box jig, Fig. VI., for machining the valve seats, valve spindle guides, valve plug holes and the holes and facings generally where necessary, the locating point being the blocks B, which fit in the finished bore. It is probably unnecessary to describe this jig in detail, but a point worthy of note is the fact that it is found advisable to drill the valve spindle guide from the underside, the jig being turned over for the purpose. If drilled from the top side, a bushing being inserted in the top of the jig to steady the drill, the latter will tend to run out of truth in starting on the rough surface of the boss, on account of the long distance between the boss to be drilled and the steady bush. If drilled as suggested, and a bar with steady bushes formed within the jig close up against the work, the result is far superior, and as the finishing of the valve seat is performed by means of a peg cutter, when the jig has been reversed, it follows that the valve seat and spindle are bound to be concentric. A four spindle drill is used in connection with this jig, and consequently only two settings on the drilling machine table are necessary.

Regarding the initial cost of jigs for monobloc cylinders this is approximately 30 per cent. greater than for a simple twin cylinder casting, but, on the other hand, the reduction in the number of "settings up" makes the block system far and away more economical, and soon saves the difference.

It is found that the labour cost of the complete operation of boring all four bores is considerably under 1s., and the inclusive cost for all machining works out well under 12s.

After the final machining operation is

Fig. X.

completed, the jackets should be subjected to a water test of not less than 40lbs. per square inch and the cylinder barrels to 300 lbs. per square inch.

It may be thought necessary to carry out a water test preparatory to any machine work being put on the castings, but the author's experience is, that if due care is taken, the castings are so uniform as to make this procedure unnecessary.

It may be said that the production of monobloc cylinders is, to a certain extent, a specialised branch of the founder's art, in which connection it may be safely said that the productions of many British firms are now fully equal to, and probably in advance of, anything that is done in America or on the Continent.

It is the opinion of the author that the monobloc cylinder will find its place in motor engineering for all motors of less than 80 mm. bore, and that the disadvantage of inaccessibility of pistons will eventually be overcome by the adoption of some type of detachable head which will be proof against leakage and joint trouble. In conclusion, the thanks of the author are due to Mr. Kenneth Crossley, of Messrs. Crossley Bros., for permission to furnish much of the foregoing.

EDITORIAL NOTE.—A COMPARISON OF CLAIMS.

The article above is perhaps of greater interest on account of the somewhat contrary opinions which were expressed in the article on cylinder machining which we published last month.

While it is obvious that Mr. Reeves is sincere in his belief that the monobloc system is the most satisfactory in almost all respects within reasonable limits of size, there are many other engineers who will be found to disagree with him. There is no doubt but that the monobloc system of cylinder casting requires special study in almost every department if it is to be successful, and it is certain that some of those who have tried it, but now condemn it, have encountered failure because they have not given sufficient attention to the design of either the cylinder castings or the tools.

Whether the monobloc system becomes the standard system for engines of small and moderate size, or whether it does not do so, must depend principally upon whether the system is found to be an economical one in all respects, from the point of view of the manufacturer. It is not the sort of question in which the private owner exercises much influence, because the trouble of handling and so forth, as mentioned by Mr. Reeves, does not appeal to more than one per cent. of purchasers. Appearance is all in favour of the monobloc, and, given the essential special tool equipment, the monobloc is in some respects a better engineering job, for it ensures correct alignment. But the matter is one which is not likely to be settled finally for some time to come yet.

THE DETAIL DESIGN OF STEERING GEAR.

FOLLOWING the article on the laying out of steering connection by Mr. J. L. Napier, which appeared in our last issue, a comparison of the different mechanical details of a selection of actual car-steering mechanisms will be instructive. As a general rule the durability of the several portions of steering gear is not very satisfactory by comparison with the other parts of a chassis, and this can only be due to undue loading. If this is so, then it remains to be discovered whether it is possible to reduce the load, or whether it is easier to increase the dimensions, and perhaps the arrangement of the parts which appear to wear badly. Our contributor has suggested one or two ways whereby the stresses may be reduced, but apart from this it is well to consider how wear may

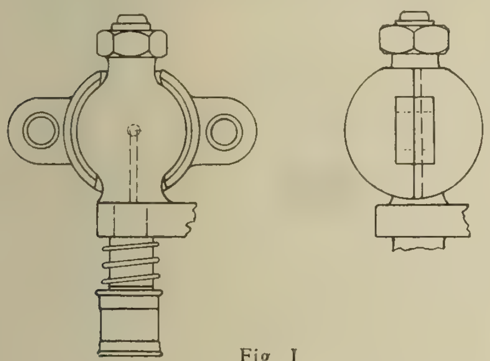


Fig. I.

be lessened, assuming the arrangement to be unchanged.

Slackness at any steering joint is unpleasant, both because it affects the ease of the steering and because it is liable to give rise to noises of a rattling nature, and, generally speaking, the former disadvantage is most patent when the lost motion is between the wheel and the steering arm, and the latter when it is the pins at the ends of the tie rod that are at fault.

Ball joints have been much improved during recent years, and their adjustment is often automatic, while they are frequently so designed that there is practically no danger of their coming apart completely, but there are still some truly dangerous joints to be found on chassis of otherwise excellent nature. It is only necessary to have seen one accident due to the dissolution of the steering gear to appreciate the immense importance of its absolute safety. After the ball joints there is only one other point where steering gears are often dangerously weak, and that is at the stub axle steering arm. In cases where the outside member of the steering pivot, the stub axle, the tie rod arm, and the steering arm are all made in one piece, it is not unusual to find that the last-named part is of small section, and is further weakened by bending into a double curve after the stamping is made, it being easier to stamp the arm straight and to set it afterwards than to stamp it in its final position relatively to the other parts. Also many breakages have taken place of tie rod arms which bear the steering arm as a branch from them, because this puts the stress of steering both the wheels on a single tie rod arm. There is no reason why the steering rod and the right-hand tie rod arm should not be made in one piece, but when this is done it should not be forgotten that the right-hand tie rod arm needs to be stronger than does the corresponding left-hand arm.

The small size of many steering gear parts has already been referred to. Probably it is due to lack of appreciation of the stresses which steering joints are called upon to bear, and even if the stresses are appreciated, the fact that the motion is small and not continuous has tempted designers to cut down dimensions. However this may be, there is great need either for joints of sufficient size to last for a car's life without wearing enough to cause slackness, or for an arrangement which will make the removal of slackness easy by the replacement of a bush, or the actual adjustment of a joint. Ball joints are usually self-adjusting by means of springs which keep the ball cups in contact with the ball. As regards the worm and sector gear, this is now often made with a complete worm wheel, so that as teeth wear away fresh ones can be brought into mesh with the worm. Tie rod joints, on the other hand, are seldom any more than plain pins in plain unbushed holes, and when they are worn it is necessary to reamer out the hole and use a larger pin. On a few cars ball joints have been used on the tie rod with self-adjusting springs, but this has the disadvantage that the correctness of the steering depends upon the exact length of the tie rod, and if this can vary, even be it ever so slightly, the effect on the tyres and the car's steadiness is likely to be highly detrimental. One method of adjusting these joints is to have ball ends to the tie rod and split cups, which can be taken up like an ordinary split brass, on the tie rod arms. This is done on Mercedes chassis, as illustrated in Fig. I. Taper pins can be used, but they are usually no better than plain parallel ones, as wear at once forms a ridge at the end of the effective area, and so prevents adjustment. Many firms use hardened pins, but omit to harden the ends of the tie rod arms in which they work, partly because of the danger that they may break if too hard, and also because it is better that the pin should wear away than that the hole should become enlarged. Where cost is an object probably the best method is to harden the pin and fit the tie rod arm with a phosphor bronze bush that can be replaced easily when worn, and, where cost is of no particular object, it would appear to be worth while to make experiment with small roller bearings, and it is more than likely that an adjustable ball bearing of the cup and cone, or cycle, type would prove more satisfactory than any other arrangement.

The great advantage of ball or roller bearings for such joints is not their low frictional resistance, though this is an advantage, as much as the fact that they require much less lubricant, and one good greasing would last them many times as long as it would a pin joint. Also a ball or roller journal has free internal space wherein grease can be retained, whereas a pin joint can only be provided with quite a small oil groove, and the motion is of such a nature that fresh lubricant has difficulty in finding its way between the surfaces. Another point against the pin joint is the difficulty of enclosing it effectively. It is in a position where it reaps the benefit of all mud and water splashed by the front wheels, and the customary leather wrapping is none too satisfactory as a water excluder or a grease retainer.

Turning to the joints which are necessary on the steering connecting rod, the ball joint is more common than other forms of universal coupling, for two reasons: it is much cheaper to make, and it is much easier to adjust. A universal joint with pins forming part of it suffers from the same disadvantages as do the tie rod pin joints. The chief aim of designers has been to make ball joints secure against involuntary separation, and the methods of carrying this into effect are many and various. The joints on the Mercedes gear, as shown in Fig. I., are all similar. Slackness can be taken up by a fairly simple mechanical process, and if the nuts on the bolts which secure the caps are provided with split pins, the risk of accidental separation of the parts is negligible. The disadvantage is the rather high cost of manufacture, and the fact that the taking up of slack is a job of such a nature that it could scarcely be performed by the average driver. Renewal of a bush or adjustment of a simple ball bearing could be made by much less skilled hands.

A safe rule for designing ball joints is to determine that there shall be two separate parts, which both require to be removed before the joint can come apart. Thus if there is first a lock nut and then a split pin of reasonable dimensions, there is double security, because the nut has to loosen and the pin shear before

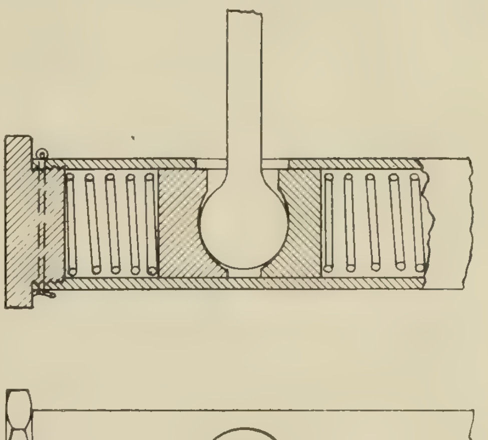


Fig. II.

the joint comes apart. Similarly, where the cups are kept pressed against the ball by a spring, the breakage of the spring should not free the ball completely. There is a type of joint used by a few manufacturers which is trebly secured. A section of it is shown in Fig. II. The ball is inserted through the hole in the tube, and to do this it is necessary to force back the inner cup by means of a piece of tube cut away at the end to allow the

ball to pass. Then the second cup and spring are placed in position and finally the cap is put on and locked. If one spring breaks, the other moves the whole tube relative to the ball, and nips the stem of the latter between its edge and the edge of the hole in the tube, and so the joint is still safe. For accidental separation it would be necessary for both springs to break, and even then the ball

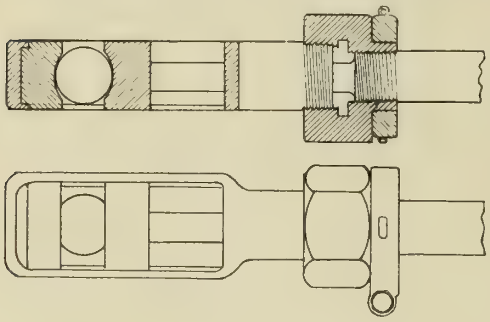


Fig. III.

might be unable to escape without loosening of the cap, while loss of the cap would be equivalent simply to breakage of the front spring. This is just as secure as the type in which the ball has an extension passing through a slot in the far side of the tube, and having a nut on the end, and it is far neater.

The Napier joint is shown in Fig. III. The ends of both the tie rod and the connecting rod bear these joints, which are leather wrapped. The body of the joint is a forging, and the hardened cups slide in it. The end of the tube (tie or connecting rod) has brazed to it a piece of solid steel bearing a fine thread at the larger end, which is of the same diameter as the outside of the tube. The inner end of the forging slides on the smaller portion of this piece of steel, and it also has a thread of the same diameter as that on the tube end, but of coarser pitch. The two parts are connected by the differential nut, and the latter is locked by a split clamping ring, which encircles a split cylindrical extension of the nut. A series of holes are also drilled through the tube end, and a single split pin hole through the nut and locking clip. To adjust the joint the pin is removed, the nut is taken off the coarser thread, and is given a part turn on the fine thread before en-

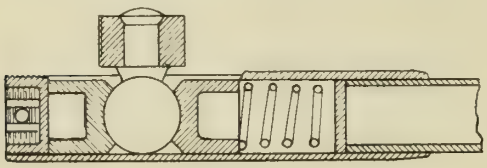


Fig. IV.

tering the coarse thread again. When done up it will then be found that the split pin occupies a different hole. The security is obvious, but the joint is expensive to make, and none too easy to handle; in fact, it would be almost as troublesome to adjust as the Mercedes type, and suffers from the same disadvantage, that it would require great care to be taken lest the ball should be held too tightly between the cups.

The De Dion joint, Fig. IV., has the old type of slotted tube at each end of the connecting rod, but it is none the less a secure arrangement. The rear cup is held against the ball by a spring, and the other is secured by a castellated nut, which screws inside the tube. This nut is locked by a cotter pin secured by a nut which is itself locked by a split pin.

The Straker Squire connecting rod has a pin type of universal joint at the rear end, and a ball joint of customary pattern at the front end. The universal joint is peculiar in that it uses the end of the steering arm for one bearing surface. There is no provision made for adjustment, but a greaser supplies lubricant to both surfaces through a hollow pin, and the whole is wrapped with leather. It is shown in Fig. V.

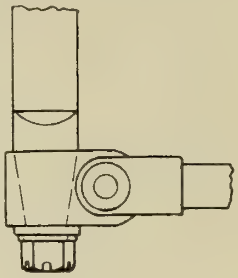


Fig. V.

The Metallurgique cars have a very simple type of joint, shown in Fig. VI. The cups are prevented from rotation by small set screws in the tube, and both cups are backed by springs. The cap which secures the whole is locked by another small set screw, and is itself screwed to the tube. This arrangement is almost exactly the same as that shown in Fig. II.

The N.S.U. has a rather unusual joint, shown in Fig. VII. It is self-explanatory, the clamping screws on the split cap serving to clip the latter to the tube. The front end joint on the same car is precisely similar to the Mercedes, and this last-mentioned fact brings up a curious point, which is the obvious inclination

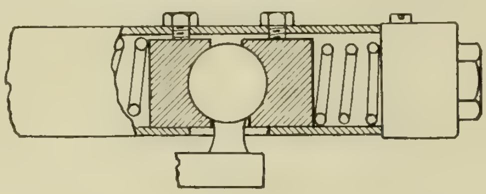


Fig. VI.

of designers to try several different kinds of joint on the same chassis. It certainly gives the impression that the superiority of any one type is a matter of more than usual uncertainty. It is perhaps interesting to refer to an early design for comparison with modern practice. Taking a $4\frac{1}{2}$ h.p. Panhard of 1896, as shown in Fig. VIII., it will be seen that both connecting rod joints were of the pin universal pattern, while the rod was solid and not a tube. The tie rod joints on the same car were almost exactly the same as the majority of those still in use.

The Humber joint is explained by Fig. IX. It is ingenious and secure, even if the spring becomes broken. It depends, however, entirely upon the locking of the cap, and this might be more secure.

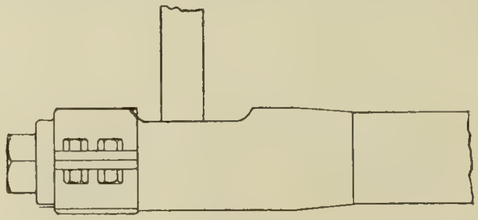


Fig. VII.

Not long ago it appeared likely that the ball joint would disappear in favour of the universal, but at present it seems that the former, as instanced by the best examples, is equally secure, and somewhat less liable to undue wear.

We have considered tie rod and connecting rod joints, and indicated some of the directions in which improvement might advisably be attempted, and may now turn to the steering reducing gear. Here the one-time popular rack and pinion has practically ceased to exist, and the worm and segment is almost always used There are a few worm and nut gears, but these have little to recommend them. It is often claimed that a worm and nut can be so made that they are adjustable for wear, but this calls for a complicated construction, and the fact that wear is localised, by the majority of steering motions being near the middle of the travel, introduces difficulties almost as

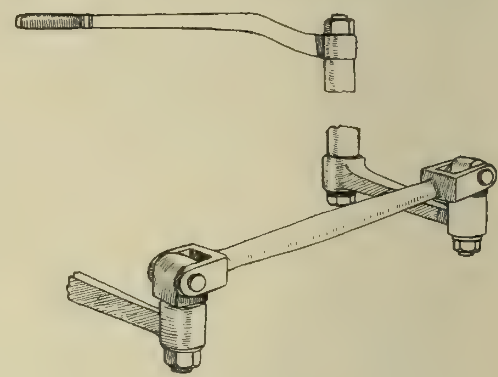


Fig. VIII.

great as it presents in the case of the worm and segment. Probably one of the best methods of adjustment now in vogue is to supply a worm and complete wheel, instead of a segment only, so that the wheel may be turned when worn and bring a fresh portion into engagement. An example of this arrangement is shown in Fig. X., which is the Armstrong-Whitworth gear.

An excellent example of the ordinary segment gear, but with adjustable thrust bearings, is the Maudslay, shown in Fig. XI.

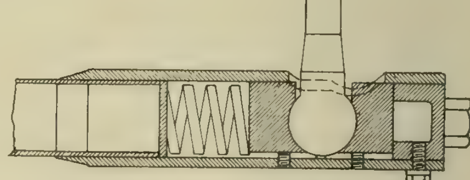


Fig. IX.

A little known gear with some good points was used on Orleans cars, and the idea inherent to it seems worthy of application to other cases. In these cars the worm and segment of a worm wheel were replaced by a bevel pinion on the steering column, and a section of a crown wheel on the steering arm. The pinion was backed by a strong spring, and was free to move slightly on the column; thus whatever wear took place, and however irregular that wear, the pinion was always in full mesh with the wheel segment, and there was no slack. This idea would appear to be capable of application to the worm and segment type of gear, and even more easily, perhaps, to the worm and nut.

The disadvantage of the bevel gear is not its reversibility, for the usual gear is quite as free in this respect, but the large size which the bevel segment must be to give sufficient reduction. Just as the number of points where slack can develop must be remembered and taken ac-

count of when designing the other joints of a steering gear, so ought the same to be kept in mind in making the reduction gear, and it is here that the worm and segment scores, as there are only two places where lost motion can occur,

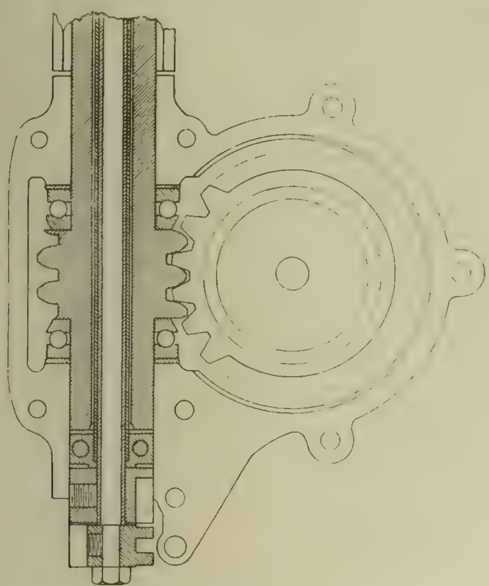


Fig. X.

namely, between the worm and wheel, and at the thrust bearings. The latter can so easily be made adjustable that the last point ought not to be worthy of mention, though there are very many good cars in which adjustment is not so provided. On the other hand, with the worm and nut gear the motion of the nut must pass through a sliding joint between itself and the inner end of the steering arm. This slide is usually unadjustable, and difficult to make adjustable, and it may more than annul any adjustment possible on the worm itself.

However, whatever the pattern of gear employed, the fact remains that on almost every car the durability is far from good. In some cars it is better than in others, but there are scarcely any in which three years' average use will not produce an altogether uncomfortable amount of slack. To double the area of contact between the teeth or threads engaged would not require a steering box of dimensions sufficient to render its attachment to the chassis impossible or even awkward, and the cost of doubling the size of the parts would not be worth considering, while it would at least double the durability, and the greater space would make it easier to find room for an effective adjustment.

There is only one other bearing in the majority of steering gears, and that is at the pivots, which are usually the best designed portion. These also are often made too small, and ought always to be provided with a ball bearing thrust, though there are still some without this refinement. The extra amount of stress thrown on the other parts of the gear, when the thrust is plain, is very considerable, and results in a perceptible increase in the rapidity of their wear, which is again a very obvious fact, but it bears repetition for the advantage of those who have failed to appreciate it. The advisability of mounting the pivots entirely on ball bearings is debatable. It certainly increases their durability slightly, but it increases the cost of renewal as well as the cost of manufacture, and if the swivel pin is large enough, its life, and the life of its bushes, can be regarded as quite satisfactory. The most common fault is defective lubrication, due to the length of the bearing and the smallness of the oilways, which makes it impossible to force grease from a screw-down cap from the top to the lower end. This can be overcome easily by the use of

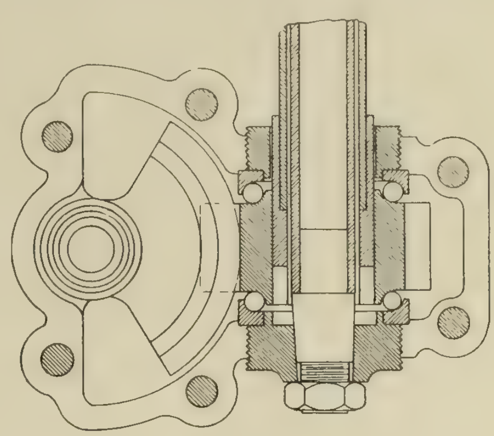


Fig. XI.

a larger groove, or a hollow pin with an additional grease outlet at the centre.

In conclusion, it may be instructive to give a list of the average proportions of the principal parts of the gears which have been mentioned, and this is done in the appended table :—

Wheelbase	2,878 mm.
Track	1,332 mm.
Length of tie rod	1,147.5 mm.
Length of tie rod arm......	153.3 mm.
Length of steering arm on worm wheel shaft	246 mm.
Diameter of pin in tie rod joints	17 mm.
Length of pin bearing ...	25 mm.
Diameter of ball in ball joints	30 mm.

THE DEASY WORKS.

A brief description of their construction and equipment, and the systems employed in the manufacture of Deasy cars.

THE Deasy Company, as may be known, was started in February, 1906, by Capt. H. H. P. Deasy, who had as his engineer Mr. E. W. Lewis. The company purchased from Mr. George Iden the property at Parkside, Coventry, which at that time comprised what are now only the machine shops and general offices, together with a considerable area of land to the south and east. On part of the vacant land to the east the company immediately built a large extension for car erecting, pattern making, body building, etc., and on the accompanying ground plan this extension is shown as the new buildings, and the general arrangement of the works is clearly seen. Last year a further useful extension was added in the shape of a building for finished stores.

The main frontage on Parkside comprises the general offices, and consists of a three-storey building with basement. On the top floor there is a commodious and well-lighted drawing office, with photo printing room, occupying all the space to the right of the centrally-situated staircase, whilst to the left are the accounts department and correspondence office. On the floor below there is a long passage giving access to the board room, the managing director's office, sales department, secretary's office, purchasing department, and works manager's office. On the ground floor there is an entrance hall, and off this an enquiry office, telephone exchange, and waiting room, all the rest of the floor being occupied by the receiving office and rough stores, which have a separate entrance from a lane which forms the western boundary of the property.

A door at the back of the rough stores leads to the steel store, and by a short flight of steps to the machine shop, tool room, axle and gearbox assembling, and tinsmiths' department, all of which are arranged in the old building, extending to a total depth of 220 feet southwards from the back of the general offices, with a width of 100 feet. This building has a ridge-and-furrow roof, lying east and west, on bays of 20ft. span, providing excellent north lighting. Down the centre is a main gangway with passage ways on either side between the rows of machines, to which they afford convenient access. Fig. I. shows a section of the capstan lathes, and gives a good idea how the machines are arranged. Last year a considerable addition was made to the turret lathes, necessitating some little re-arrangement and closer fitting, but, although it will be observed that no space is wasted, there is still plenty of room for transit of material. There are no trolley-ways, or other means of mechanical transport provided, for, as all the material is of a light description, and can be easily carried, they are not required. Beginning at the north end of the shop, and proceeding southward, the machines are arranged on either side of the main gangway in the following order :—On the left hand are the capstan and turret lathes, then the gear-cutting machines, next to them the grinding machines, and at the bottom end the vertical and horizontal milling and keyseating machines. To the right of the gangway there are the slotting, drilling and boring machines, then the crankshaft lathes, while a batch of miscellaneous lathes occupies the rest of the space. Extending down the left-hand wall of the shop, beyond the line of the machines, is arranged the viewing department, so that all work can be viewed after each operation, without the inconvenience of taking it into the stores, as is customary

in some shops, or doing it at the machines, as is the practice in others. The Deasy plan is found to answer very well, and to reduce delay. After final viewing, work is passed into the finished stores, which will be described later.

The capstan lathe section is, as might be expected, a strong feature. Quite seventy-five per cent. of motor work can be done on this type of machine, though it is being ousted by grinders for an increasing number of purposes. Of the makers, Herberts are rather largely represented, but there are also examples of Gisholt, Warner and Swasey, Jones and Lamson, and others—a very useful selection. Amongst the other machines are some Reinecker gear shapers, Landis grinders, and a vertical cylinder grinder by Burton Griffiths, who have also supplied some Woodruff keyseaters, and a Giant vertical miller.

At the end of the machine section proper a cross passage right across the building with double doors at each end, gives access on one hand to the western boundary lane, whilst the other leads into the test department. This passage serves to divide off from the machines the tool room, the axle and gearbox assembling shop, and the tinsmiths' and blacksmiths' section, all of which are located in the south end of the main building, the tool room occupying a central position, and the others surrounding it.

The axle and gearbox assembling shop is shown in Fig. II. Great care is taken to see that workmanship is all that it should be in fitting up the gears, and all the wheels are pressed on their shafts by means of an hydraulic press.

The tinsmiths' shop, which lies along the extreme south end of the building, is shown in part in Fig. III. The principal work carried on here is the making of the bonnets, samples of which are seen in the foreground. In the background will be noticed a large pipe, and it may be explained that this is an air trunk, which conveys warm air to the car-erecting shop in winter, and fresh, cool air in summer. This will be referred to later.

The blacksmiths are arranged along the south-west corner of the building, a position that lends itself to convenience in working, and which appears to be entirely satisfactory.

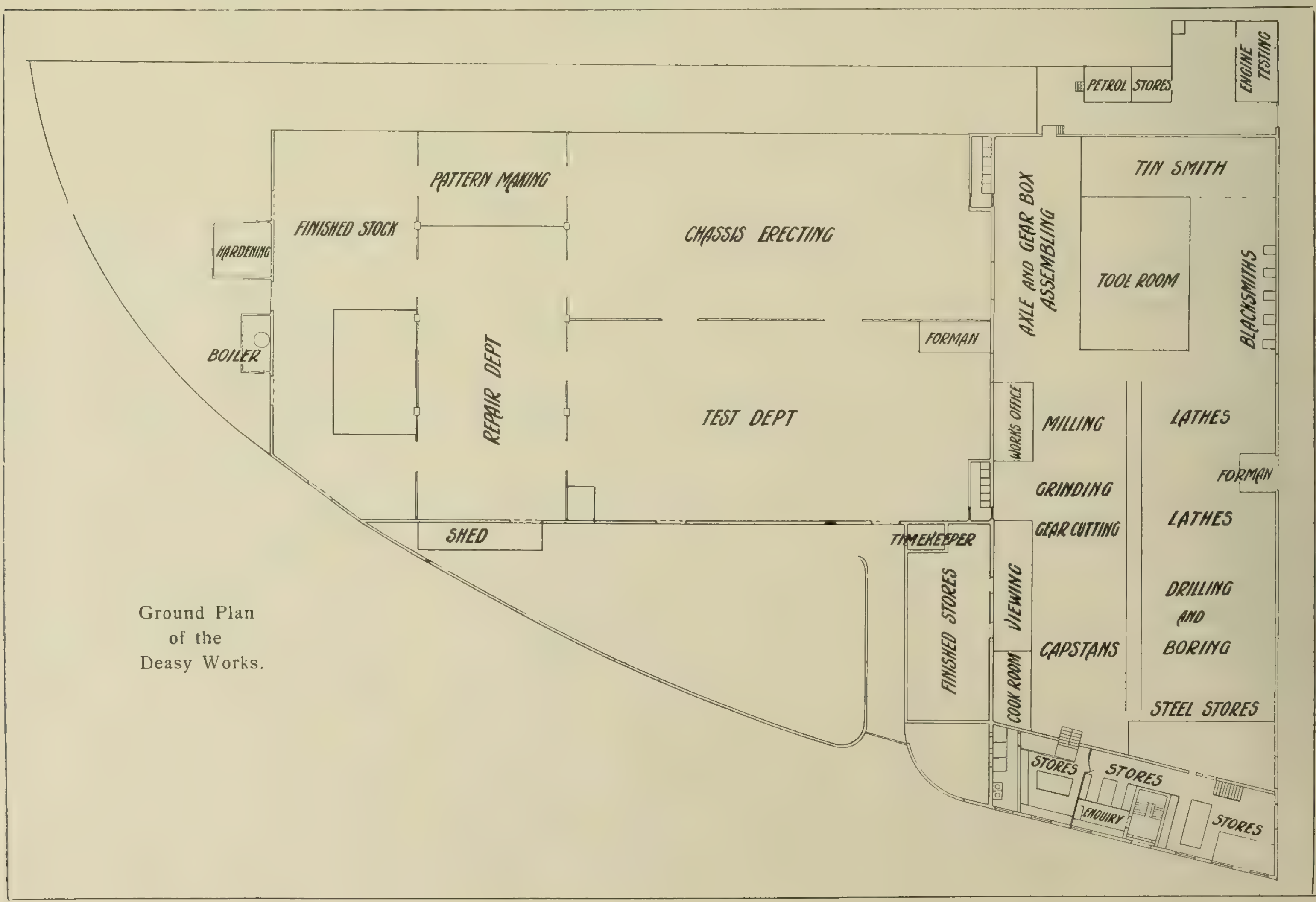
The tool room, as already stated, occupies a central position in the south end of the building. It is about forty feet square, and is completely enclosed by wood and glass partitions, shown clearly in Fig. III. Here a few tools are made, and all the small tools and jigs are kept, but the principal work of the department is to make jigs and to keep tools in good repair—for which it is well equipped with lathes, milling and shaping machines, precision grinders, and gas-fired hardening furnaces. There is a Reinecker backing-off lathe for making special taps and reamers, but, considering the ample facilities which exist nowadays for getting all such tools from firms who specialize in their manufacture, and also the strong desirability of avoiding in designs anything that calls for other than standard tools, the installation of such an expensive machine is hardly necessary.

All tool racks are limited to a height of about 4ft. 6ins., so enabling every man to be seen by the superintendent.

The new building is of loftier section than the old one, and in the car-erecting department, shown in Fig. IV., this fact is very evident. To this department come the frames and the built-up components, the engine, gearbox, axles, steering sets, etc., and here they are assembled into complete chassis. Formerly this and the test and repair department were all one, but during last winter a central partition was erected, and other arrangements made, so that each section is now quite separate. From the erecting department the complete chassis pass to the other side of the partition to the test department, Fig. V., where they are tested and tuned up before being despatched to have bodies fitted, and completed cars are also tested.

Power in the Deasy Works is provided by electric motors of 10 h.p. and 15 h.p., taking current from the Corporation supply. The current is two-phase, alternating on the three-wire system, the working pressure being 200 volts. There are in all about eight motors of 10 h.p., one of 15 h.p., and one of 17 h.p., all except the last being made by the B.T.H. Company, of Rugby. The last one only is fitted with slip rings, and can be started up on its load, but all the others have to be started on loose pulleys. The entire absence of commutators and brushes, and the consistently satisfactory way in which these alternating current motors perform, make them almost ideal for shop work. Each motor operates a short length of overhead line shaft, the machines being suitably grouped for drives from these shafts.

Two large doors, of the roll-up shutter type, provide exits from the test depart-



Ground Plan of the Deasy Works.

ment, one on to a downward sloping paved road, to Parkside road, and the other on to an enclosed yard, which is very convenient for preliminary testing before going on to the road. There is a lean-to roof along the inside of the wall of this yard, providing accommodation for the men's cycles.

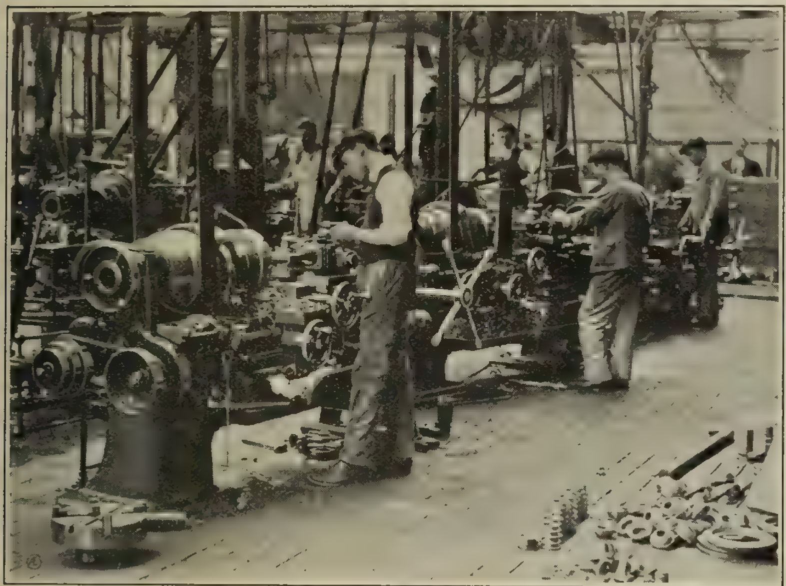


Fig. I.

Referring to the ground plan, it will be noticed that the erecting and test departments occupy more than half of the new building, which has a total length of 240 feet, and a width of 100 feet. The remainder of the building is divided into two equal compartments, of which the one adjoining the erecting and test departments is now used to accommodate the repair department and pattern makers, whilst the other can be utilized, if required, as a stock room.

These two compartments were originally designed for body making and pattern making in the one next the test shop, and for painting and trimming in the other, but as the company have now given up making their own bodies, the arrangements have been re-modelled as above described.

As shown on the plan, there is a small plot of land to the east, on which are erected a hardening shed and a boiler house, the latter being in connection with the steam heating of the new building. The hardening is done by means of a coke-fired furnace, but there is nothing concerning it which calls for special notice.

Fig. III.

The finished stores consist, as already mentioned, of a new building, erected last year, on a vacant space which lay between the sloping approach way to the test shop, and the last wall of the machine shop. It is a fine, lofty, and well-lighted building, 76ft. long by 28ft. wide, and 22ft. high to the eaves. The walls are all covered with storage bins, of which there are over fifteen hundred, with a convenient platform all round giving access to the higher bins. There is besides, a counter surrounding the floor space—leaving a good passage between it and the bins—giving ample table room for parts until they can be examined and stored away. This counter has also bins under it, so it will be seen that the storage accommodation is on a liberal scale, and well arranged. Fig. VI. gives a good idea of the general features of the store, and in numbering the bins, the plan has been followed of making all the horizontal rows start from 1, whilst vertically each row is designated alphabetically; thus large numbers, which are apt to be confusing, are not required.

Two heating systems are in use at the Deasy Works, a hot air system in the old building, and a low pressure steam system in the new building. The hot air system consists of two furnaces, situated in the basement of the office buildings, and using coke or scrap-wood fuel, the hot gases from which heat up a battery of tubes, through which air is forced by means of two large Blackman type fans. The heated air is carried along a brick-built tunnel, down the centre of the shop, under the main gangway, from which it enters the shop through suitable gratings. The system is very satisfactory, and, without any extravagant consumption of fuel, it keeps the shops at a reasonable temperature in the coldest weather. The low pressure steam system has not proved altogether a success. The total cubic capacity of the new buildings is close upon 700,000 cub. ft., and the installation was hardly adequate for this, the grate area being only 9.6 sq. feet. Last winter an extension was taken, by means of the large air trunk referred to in describing the tinsmiths' shop, from the hot air tunnel into the car-erecting shop, making the temperature of the latter much more comfortable. Unless there is already installed a steam boiler, with a large margin of excess power, the use of a steam heating system does not seem to be so satisfactory as one using hot air, or a hot water system with large diameter pipes; but this question can hardly be discussed here.

Practically the whole of the lighting is performed by metallic filament incandescent lamps, which provide a light of excellent quality. Up to last year the works were lighted by means of mercury vapour lamps of the long tube type, which is claimed to be the most economical method of electric lighting, whilst the quality of the light is supposed to resemble daylight, but experience and practical tests proved that a general system of lighting—as that was—did not compare at all favourably with metallic filament lamps placed at each individual machine. The latter system was therefore adopted in the machine shop, and gave very much improved results in output and quality of work. This was followed by the discarding of the mercury lamps in the new building, and the substitution of four-group pendants of metal filament lamps, with enamelled iron shades. The resulting effect was much brighter and more cheerful than that given by the mercury light, and was much preferred by the men.

Fig. II.

To digress for a moment, it is worth while remarking that this aspect of the matter is not always considered as it

ought to be. It stands to reason that in a bright, warm, cheerful room, with plenty of pure air coming in, men must feel better and work far better than they would in a dismal, cold, half-lighted place, where their thoughts are far more

Fig. IV.

intent on keeping warm than on their work, yet there are hundreds of engineers' works to-day still lighted by the old-fashioned, wasteful, bat's-wing burners, using four or five times as much gas as they would do with incandescent burners, vitiating the air breathed by the men, and making the conditions as unfavourable as possible for production of work. To avoid the capital expense of installing an up-to-date system of electric light or pressure gas is not economy at all; it is deliberately putting a premium on reduced output and inferior workmanship.

The timekeeper's office is just outside the large door of the test shop, facing the main entrance from Parkside, but the two Rochester Time Recorder clocks, by which the men record their incoming and outgoing, are situated inside the main building, and the men enter by a small door, which is seen on the ground plan. Time is booked on going out, as well as when coming in, so that the cards form an accurate record of each man's working hours during one week. For the office staff a clock is provided in the entrance hall, which contains a travelling band of paper passing under an opening normally covered by a slide. By pressing a lever, the slide is pushed aside, and the time is printed on the margin, opposite which the person writes his or her name. Releasing the lever again causes the paper to move on a space, ready for the next. A piecework system of payment prevails in the works, and is found to work quite satisfactorily. The hours of the men are 6 to 8 a.m., 9 a.m. to 1 p.m., and 2 p.m. to 5.30 p.m. on Monday, and 6 p.m. on other days. On Saturdays the works close at noon, so the working week is 54 hours.

For the purpose of making up wages, the week ends on Wednesday nights, and the wages are paid after closing time on Friday nights. Half an hour, previous to stopping on Saturdays, is allowed the men to clean their machines.

One of the most important points in connection with any works is the system by which the work is put through, but, as it is bound up so closely with the general costing system, it is not easy to describe within the limits of such an article as this.

Apart from accuracy of work, which, of course, is not dependent on system, the main objects kept in view are to maintain an even flow of the various parts, and obtain accurate costing. All raw or—as it is better termed—manufacturing material is received by the rough stores office, which is also responsible for the despatch of all small goods, spare parts, and the like. This office also makes daily returns to the accounts and purchase departments of all goods received and despatched. It receives from the purchase department a carbon duplicate of every order sent out, and as the goods come to hand, the deliveries are noted on the backs of these order duplicates, which thus form a convenient means of reference, and an extra record in addition to the usual "Goods Inwards" book, which is the official record. "Sanctions" for work, which, in the case of cars generally means batches of ten or twenty, are issued to the works office by the drawing office under instructions of the manager of the sales department. The works office thereupon issue complete sets of cards for each batch. Of course, every item in a car has a "part number" either cast or stamped upon it, and for each batch of cars the full quantity of each individual part is dealt with on one set of cards. A set of cards comprises (1) Requisition for material, (2) Process card, which is a complete record of the batch in its progress through the machine shop, (3) Cost card, which goes to the costing department, and (4) Operator tickets, of which there must be a separate one for every operation performed.

No material may be issued unless the whole complete lot for a "Sanction" is in stock. This is an important point, because it ensures that when the "assembling" cards are issued there can be no troublesome delays or enquiries due to missing parts. It will no doubt occur to readers that for bar work, or other capstan work, in which a considerable amount of time has to be expended in setting up the tools, the putting through of such small batches as ten, twenty, or even fifty or a hundred, of certain small parts, would not be found convenient in practice; but this difficulty is easily disposed of. Such articles may be put through in several batches consecutively, and in order that this may be done entirely without confusion, the plan is adopted of distinguishing each "Sanction" by a colour as well as a "Sanction Number," the colour being stated in each case on the cards. Hence, in his progress through the works, the visitor will notice heaps of castings at the various machines, all marked with blue, red or yellow paint, by which they can always be readily distinguished. These colours are arranged by schedule in a certain consecutive order, and it is the duty of the process clerk to see that sanctions go through in their proper rotation. The process department keeps track of all work in progress, and can always tell exactly the position of any batch and all about it.

In order to facilitate the work of this department, the sanction numbers are terminally classified in "Sections," so as to indicate the various components of a car, e.g., gear box parts will end with one certain figure, and back axle parts with another, and so on. The process cards being filed accordingly, it is a simple matter to ascertain at once the

Fig. V.

whereabouts of any batch of parts, and what remains to be done to it, each operation being entered on the process card as soon as the work has been passed by the viewer.

On completion, all parts go to the finished stores, whence they are again issued in lots, as required for assembling. An assembling card is made out for so many built up components, such as gear boxes, back axles, steering sets, and so

on, and it specifies exactly the parts necessary to make up any such complete component, even to a split pin. Similarly the cards for final erection specify the various complete components required for each chassis. The process department takes no cognizance of, and has nothing to do with costs. Duplicates of material requisitions and operator's tickets—after completion of work—are sent in to the costs office, which enters up particulars on the "cost card" issued with every set of tickets.

All piecework rates having been fixed beforehand, it will be seen that the system in its outline is effective and simple, involving no more clerical work than is absolutely necessary.

The men know their piecework rates, and as, on completion of each batch of parts, they get back their operation ticket with the parts signed for, they are enabled to check the wages allotted to them by the works and costs departments.

To carry out such a system as that first described necessitates having a good deal of room, and particularly plenty of room in the finished stores. It would be futile to try to put through separate batches of parts unless there was plenty of space to store them away separately when they are finished, and when we remember the great number of different parts that go to the making up of a car the necessity for storage room is apparent. This might be difficult to find in some factories, but it suffices to say that in the Deasy Works the facilities exist, and the system works smoothly and satisfactorily.

The company have this year made a new departure in buying their engines outside, instead of making them. This change of policy makes possible a much larger output, without a large increase of plant, for which there is no accommodation, and as a matter of fact, the output of cars now is more than double as great as what could be done previously.

Fig. VI.

As already mentioned, the company have also discontinued the body building department, Mr. Siddeley's experience having led him to the conclusion that this is a branch of the trade that is better left to specialists. This is a point on which opinions are not all alike, and no doubt a good deal can be said in favour of having the body work under direct control and supervision. Whether this compensates for the extra work and trouble involved is a question outside present consideration, but there can be no doubt that such a department requires the provision of a great deal of additional shop room, and involves the locking up of a lot of extra capital.

In conclusion, we have to express our thanks to Mr. Siddeley and the Deasy staff for giving facilities for inspection of the works, and would congratulate them upon the high state of order and efficiency, and general prosperity which was apparent.

THE 20 H.P. LANCHESTER.

A description of the chassis which also applies to the 28 h.p. in many respects.

It may be assumed that the general arrangement of the Lanchester chassis and the position of its various components is a matter of common knowledge, so the details only will be described in the following, the principal unit being that of the engine, gear, and clutch mechanism, the rest of the chassis being adapted to it rather than it to the chassis. Although the engine and transmission form a single part, they are capable of separate consideration, as they are merely bolted together for convenience in assembling and in use. Taking the engine first, it will be noticed that the stroke is extremely short by comparison with the bore, the actual dimensions being, bore 4ins. and stroke 3ins. This unusual feature is largely due to the position of the engine between the two front seats, as the short stroke enables the crank case to be made very narrow.

The normal speed of revolution is high, the gear ratios being arranged on the assumption that the crankshaft can make over two thousand revolutions per minute. To give good bearing durability the crankshaft bearings are of greater area than the average for a four inch engine, and the oil is fed to them under the unusually high pressure of forty pounds per square inch. The material from which the crankshaft is made is three per cent. carbon steel, having a strength of 50 tons elastic, and 62 tons ultimate. The nature of the oil way drilling is such as to weaken the shaft by the minimum amount, as it creates no sharp corners inside the shaft, and large radii are noticeable at the points of junction between the webs and the shaft and between the webs and the crank pins. The shaft is cut from the solid, and is ground after turning; as it is in one piece with the gear shaft it is a rather unusually elaborate part for automobile work. To appreciate this, reference should be made to both the figures I. and II., which show the engine and gear box separately. The connecting rods are five per cent. nickel steel and are circular in section, being drilled through for the passage of oil to the gudgeon pins; the outside diameter is 1in. and the inside 5-8in., this bore being narrowed at the top end and closed by a plug with a $\frac{1}{4}$in. hole at the bottom end, as seen under cylinder number two in Fig. I. Although, weight for weight, it is theoretically not quite as strong as a connecting rod of the more customary H section, the tubular rod certainly has advantages for an engine with an entirely forced system of lubrication, and it is also a simple machine shop job, even though it requires more machining than the common type.

Steel is used for the pistons, which are pressed considerably thicker than their finished size, every part of the pistons being machined with the object of obtaining unvarying weight. The gudgeon pins are $\frac{3}{4}$in. diameter and are solid, the old hollow pattern having been discarded; they are secured to the pistons by split pins passing through the bosses, and they are bushed with phosphor bronze. It is not quite clear why there should be four piston rings in addition to the scraper ring at the bottom of the piston, but if nothing is gained by the use of the extra ring, so similarly there is little lost. The big end bearings are all formed by running white metal directly into the connecting rod end and cap, and the crankshaft bearings are phosphor bronze lined with white metal. It should be noticed that bolts instead of studs are used to support the crankshaft bearings.

Aluminium is, of course, used for the crankcase, which is a simple casting with a very clean exterior. It is not provided

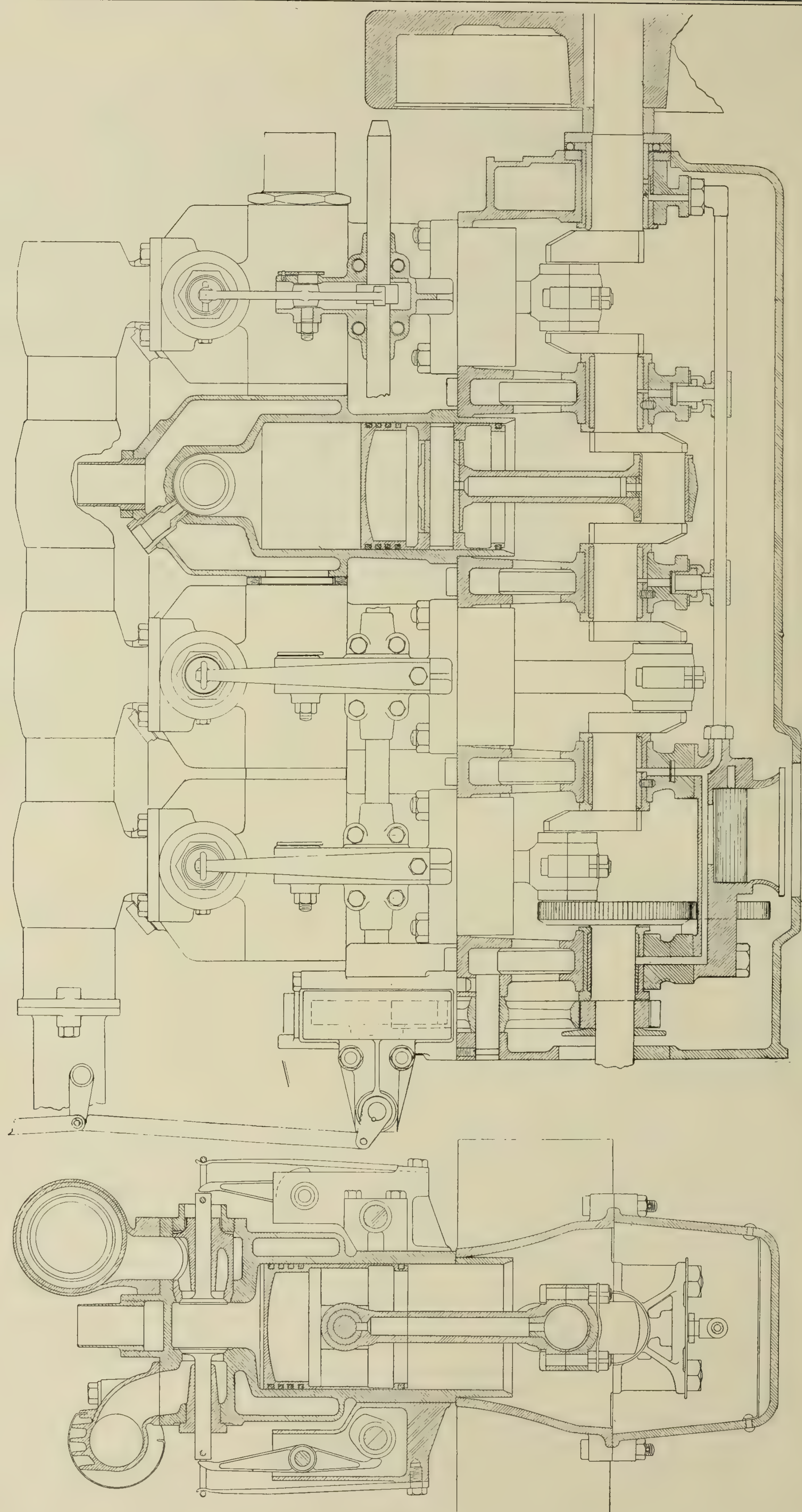
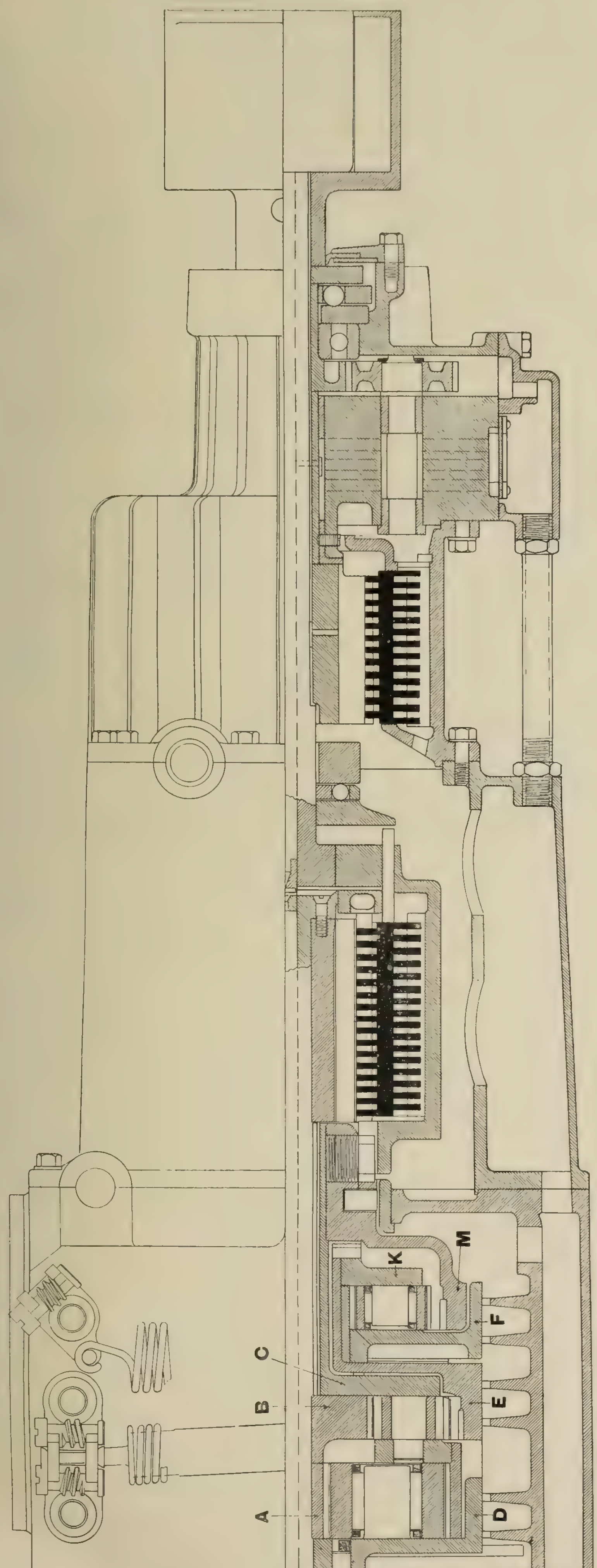


Fig. 1.

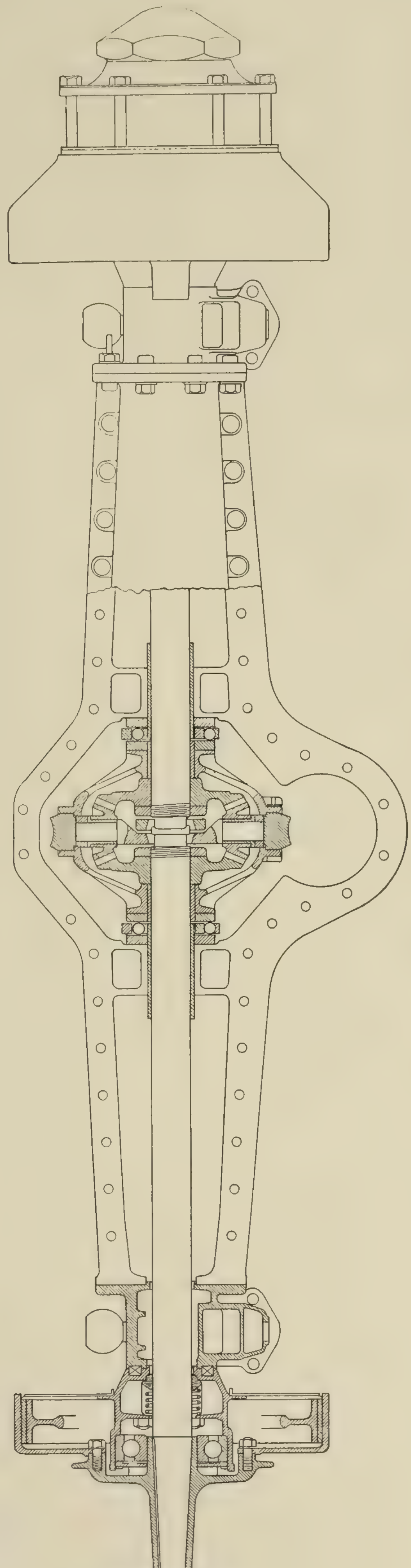
Scale: 1 in. = approximately 4.2½ ins.

SIDE ELEVATION AND SECTIONAL END ELEVATION OF THE LANCHESTER ENGINE.



Scale: 1 in. = approximately 5.9 ins.

Fig. II.



Scale: 1 in. = approximately 6.3 ins.

Fig. IV.

LANCHESTER TRANSMISSION AND REAR LIVE AXLE.

with supporting arms, but is attached to the inner frame by means of malleable iron brackets secured to the case by bolts and nuts. There are four of the brackets, and they support the whole power unit, the gear and clutch cases having no separate attachments to the frame. Separate cylinders are used with horizontal valves, as is shown in Fig. I. These valves are 5 per cent. nickel steel, and have half-inch stems, the heads being $1\frac{1}{2}$ins. in diameter, and the total lift $\frac{5}{8}$in. A simple faced joint is used for the inlet valve seating, and the outer end is rounded, as seen in Fig. I., in order that a slight amount of movement may take place between the cage and the retaining nut, so enabling the cage to set true in its seating. A minor trouble which suggests itself is that with the valves arranged horizontally, and only one valve in a cage, there would be danger of grit entering the cylinder when grinding in the other valve, but this is really a trivial matter, as the entry port may easily be stopped up temporarily. Such an arrangement gives the gases very free entry and exit, the large size of the valve stems guarding against the possibility of one of the valve heads falling into the cylinder, and the horizontal entry ports enabling the inlet and exhaust pipes to be carried along the tops of the cylinders in a position where they are quite out of the way and do not interfere with any other part.

The inlet valve is set to open 7° before the top dead centre is reached and remains open while the crankshaft turns through 202°. The exhaust valve opens 142° past the top dead centre and remains open for 225° of crankshaft revolution, so it will be seen that the inlet and exhaust valves are open together for a period equivalent to 14° movement of the shaft. The camshafts are driven by spur gearing from the rear end of the crankshaft, and are carried in bearings bolted to the sides of the cylinders, as seen in the transverse section in Fig. I. The whole valve arrangement and the flat springs all tend to save breadth of engine, but are somewhat detrimental to accessibility. This quality has received but scant attention from the designers of the Lanchester, as instance the quite considerable number of operations necessary before it is possible to remove a cylinder. But if the larger tasks are made difficult the smaller adjustments are often ingeniously provided for. In the case of the valves the tappet arms rock on pin bolts, with eccentric central portions, to which there are attached small quadrants of sheet steel with a row of holes near their outer edge. Each tappet case has a pin which can be fitted through any of the holes, so to adjust the clearance between the end of a valve stem and a tappet all that is necessary is to loosen the clamping nut and quadrant, give the bolt a part turn, replace the quadrant with the pin through a fresh hole, and lock the nut again, an operation which takes but little longer to perform than to describe.

The lubrication system was described fully in the July issue of *The Automobile Engineer*, so it is only necessary to refer briefly to it here. A large gear pump sucks oil from the sump and forces it to all the main bearings, whence it passes to the big ends and the gudgeon pins. In the old type of Lanchester the same pump supplied oil to the gear and clutch as well as to the engine, but entirely separate oil circulation systems are now used. The engine oil is filtered twice, by a gauze tray covering the whole of the sump and by a gauze covered pump intake.

Ignition is accomplished by a Bosch magneto driven from the rear end of the inlet camshaft by a skew gear, and mounted on a special platform above the gear case.

Cooling of the engine is performed by natural circulation with a parallel vertical tube radiator, through which the air draught is maintained by two fans

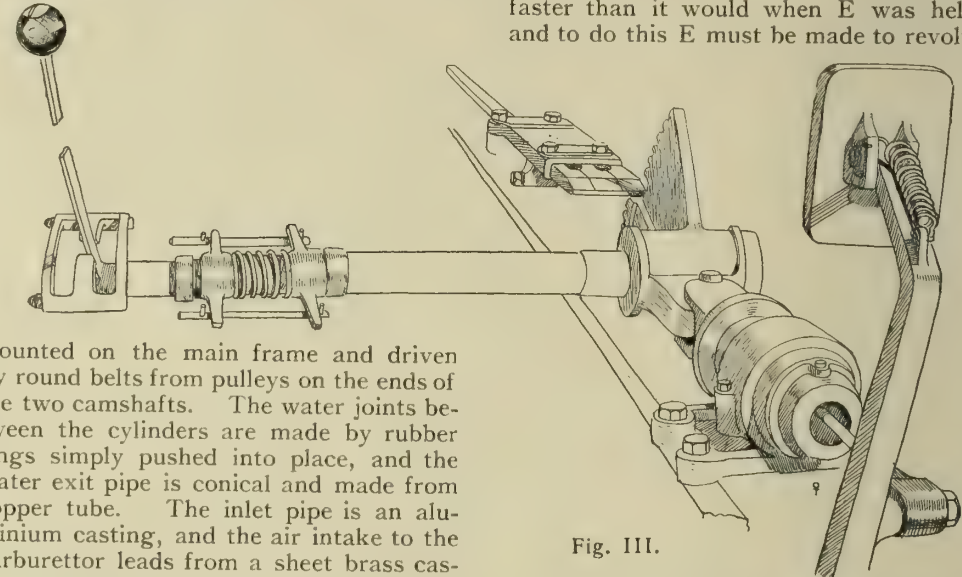


Fig. III.

mounted on the main frame and driven by round belts from pulleys on the ends of the two camshafts. The water joints between the cylinders are made by rubber rings simply pushed into place, and the water exit pipe is conical and made from copper tube. The inlet pipe is an aluminium casting, and the air intake to the carburettor leads from a sheet brass casing round the exhaust pipe.

The carburettor is, we believe, too well known to need description, and it is practically the same now as it was on the original Lanchester chassis. It is situated on top of the petrol tank, between the front seats, and directly behind the engine, the fuel being raised to it by exhaust pressure, and the mixture control being situated on the top, convenient to the left hand of the driver.

The transmission gear is somewhat complicated, in common with all epicyclic speed changing mechanism (it is the number of parts in this form of gear that has prevented its more general use on automobiles) and its operation can only be followed by reference to Fig. II. The crankshaft is continued right through the gear box proper, and the driven shaft will be seen to carry the male portions of the high speed clutch and the foot brake. When the crankshaft is revolving the following parts, being keyed to it, must also revolve at the same speed: the sun wheel A, the sun wheel B, and the inner or male part of the clutch. To the driven shaft is attached the female part of the clutch and the pinion carrier C. The drum D carries the pinions of the reverse train. The part E is the annulus of the middle train, and is secured to the piece K, which carries the pinions of the right-hand train. The drum F is integral with the sun wheel of the right-hand train. When any of the brakes are in operation the clutch plates are held apart by external mechanism, which will be described later, so it must be remembered that the drive only passes from the inner to the outer clutch plates on the direct drive.

If the drum D is held, the annulus of the left hand train revolves in a direction opposite to that of the crankshaft, and this annulus is part of the piece C, so it drives the female part of the clutch, giving the reverse gear.

If the drum E is held, the sun wheel B, being driven by the engine, causes the pinions to revolve and the pinion carrier C to revolve as a whole in the same direction as the sun wheel, giving the first or lowest speed.

For the second speed two trains are in operation, and their action can be best followed by remembering that in order to obtain a faster speed of final drive we must move the part C faster. Now the drive of the second speed still originates from the sun wheel B, so we must obviously make the pinion carrier C revolve faster than it would when E was held, and to do this E must be made to revolve in the same direction as C, but at a lower speed. E is connected to the pinion carrier K of the last or right hand train, and we can hold the sun wheel of this train by means of the drum F. C is, however, connected to the annulus M of this same train.

Now, assume F to be held and also assume that E remains stationary for an instant, then C revolves at the low speed rate and M also revolves. But F is being held, so K (and therefore E) must revolve forward in the same direction as the sun wheel B, so giving the desired effect.

For the top speed the clutch is let in, and this locks all parts of the gear together, as can be seen easily from Fig. II. The clutch plates are made from 14 gauge sheet steel, and are retained in the customary manner. The spring is external and operates through a large thrust washer seen in section in Fig. II.

The same spring as that operating the clutch is also used for the purpose of contracting the gear brakes, and the manner in which this is attained can be followed from Fig. III. The clutch spring is contained in the box, and the gate lever controls the selectors in the usual way. There are four plungers connected to the four portions of the gear (the clutch, and the low, second, and reverse brakes) and the selector cams connect the clutch spring with either plunger at will. In the neutral position the spring expands to its full extent, within the limits of the box, and no pressure is put upon any part of the gear, which revolves idly, while to allow for free movement of the selectors, when in use, the change lever is carried on a swivel bearing on the frame side.

The brakes themselves are locomotive type, and they are contracted by a right and left hand screw at each end of the actuating shaft, and are provided with an adjustment for taking up wear of the

shoes. The foot brake is very similar to the clutch, and is actuated by a separate lever working through a plain thrust. It is held out of engagement normally by an external spring and the thrust, when it is in operation, is taken by the thrust bearing immediately behind the driving pot. For the lubrication of the whole of the gears the clutch and the brake, there is an oil pump, similar to that used for the engine, situated at the rear end of the brake chamber. This sucks oil from the sump at the bottom of the case and supplies it to the inside of the driven shaft, whence it passes out to the plates of the brake by radial holes. From the end of the driven shaft the oil is led into the end of the crankshaft by means of a jet connection, seen in Fig. II., and thence it passes through radial holes to the plates of the clutch and to all parts of the gear.

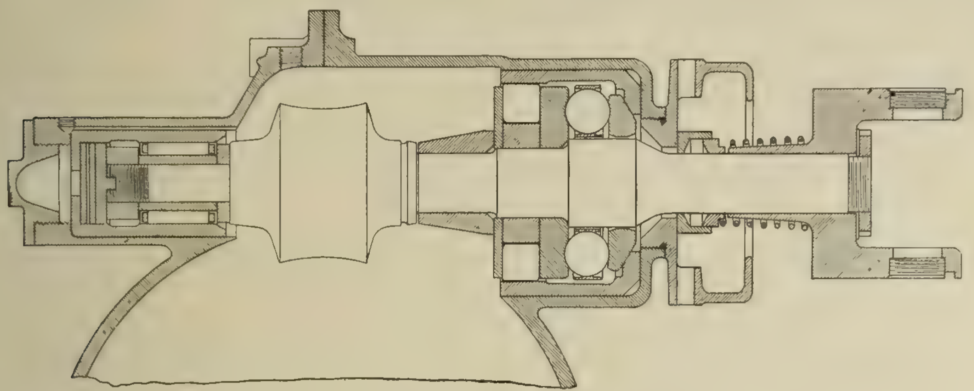


Fig. V. Scale: 1in. = 3·8ins. approximately.

It is not our present purpose to discuss the broad question of the advantages and disadvantages of an epicyclic transmission, and therefore it would be unfair to criticise the gear adversely on account of the number of parts in it. The epicyclic portion is no more complicated than are most similar mechanisms, and its principle is such that it is efficient on all gears, including the reverse, as in no case are any parts attempting to make impossible speeds of revolution. The gears are all case-hardened mild steel, and all the pinions run on roller bearings, this pattern of journal being used also to support the whole gear in its casing. Malleable cast iron is used for the revolving clutch box, or driven member. Aluminium is employed for the outer casing, which is deeply ribbed to increase its strength, and phosphor bronze is used for the long bushes which separate the different gear sleeves, and for the tailshaft bearing. This last-named shaft is also provided with a ball journal, and the purpose of the plain bearing is partly to provide a ready means of passing oil to the inside of the shaft.

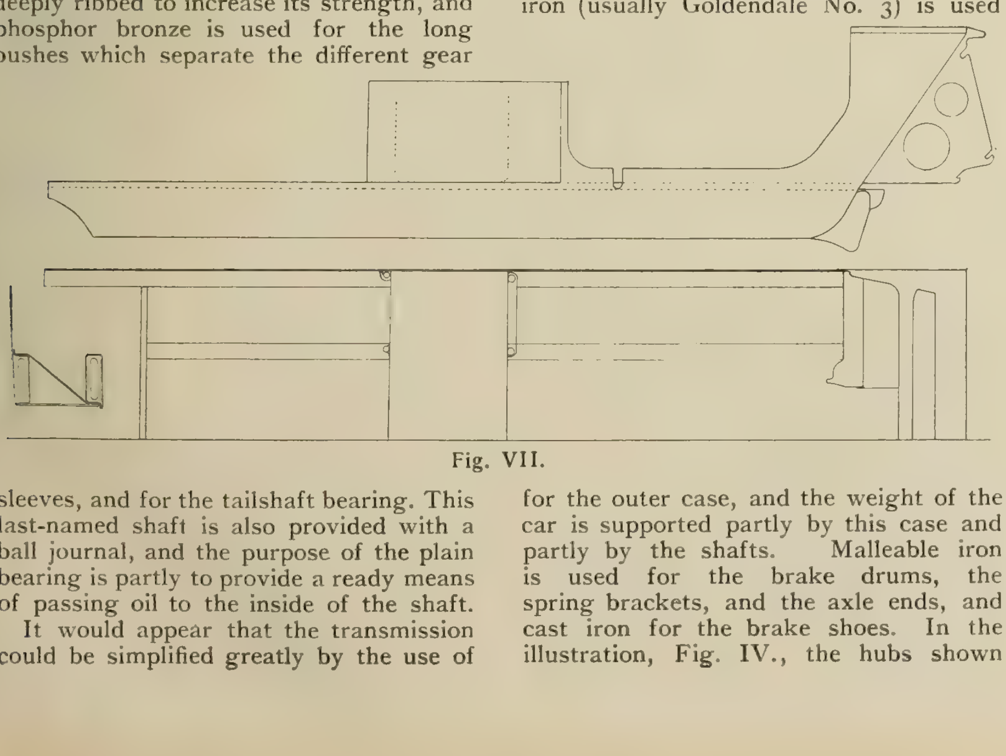


Fig. VII.

It would appear that the transmission could be simplified greatly by the use of a different pattern of clutch and a less unusual type of foot brake. With an epicyclic gear there is not the same need for a delicate clutch that there is with the sliding gear, and a metal-to-metal single plate, or cone clutch, should answer all requirements, with many fewer parts and a considerable shortening of the box, to say nothing of reduced cost of production. As regards the brake, this is certainly very sweet in action, and wear is easily taken up by the simple expedient of putting in an additional plate, but a locomotive type of brake, run in oil, ought to be almost as delicate, would be far easier to adjust, and would be cheaper.

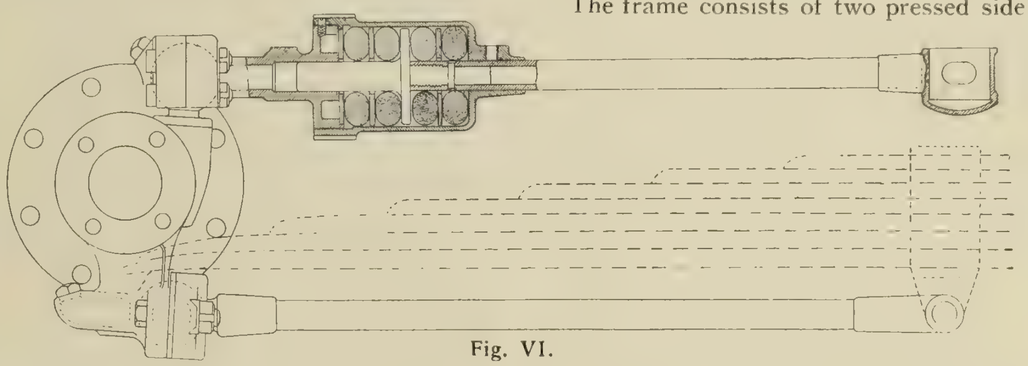


Fig. VI.

The forward end universal joint is a modified De Dion type, having a large cross pin, on the end of the shaft, bearing directly upon the slides of the pot without any intermediate square nuts, and the rear end joint is of a cross-pin type with easily renewable bushes, as seen in Fig. V. The back axle is seen in Fig. IV., and calls for but little explanation. Cast iron (usually Goldendale No. 3) is used for the outer case, and the weight of the car is supported partly by this case and partly by the shafts. Malleable iron is used for the brake drums, the spring brackets, and the axle ends, and cast iron for the brake shoes. In the illustration, Fig. IV., the hubs shown are for Lanchester wire wheels and for artillery wheels respectively, though the latter are not often supplied. The large roller bearing for the worm (Fig. V.) is interesting, as is the abnormal size of the balls in the worm thrust. The tail end thrust is plain, as it is adjudged unnecessary to use a ball washer for a point where so little working contact occurs.

As regards attachment, the front and rear axles are precisely alike, the ends of the springs being passed through the slots in the pads, but not otherwise secured. Retention of the axles is performed entirely by the radius rods, which are shown in Fig. VI. They have spherical cups at the axle ends enclosing steel balls fixed above and below the spring pads. The springs are half elliptic, and are partly enclosed by the frame members, the rear springs being inside the main frame, and the rear ends of the front springs inside the sub-frame. Each spring is clipped at its centre, and the clips rock on pins in the same axial line as the pins of the lower radius rods. All these rocking joints are provided with grease caps, and the frame ends of the upper rods have grease-retaining pots in addition, as shown in Fig. VI. A division is made in the upper rear radius rods, and a rubber cushion is inserted to lessen the shock of too sudden an engagement of gears or foot brake.

The frame consists of two pressed side members and a similar pair of inner frame members, which are also pressed, and are straight; they are connected to the outer members by a number of light malleable brackets and by the petrol tank, which is the principal cross member of the whole frame. This tank is, of course, made of stout gauge sheet steel, and is attached by bolts. The other cross member is the dashboard, which is also pressed, forming the front end of the frame. A most unusual section is used for the frame pressings, it being a complete rectangle, as shown in Fig. VII., the edges being riveted together. It should be noticed that the outer members have the riveted flange on the top and on the outside, this leaving a ledge wherein to lodge the footboards. Exceptional rigidity is obtained by means of this peculiar frame construction, it being possible to lift one corner of a complete frame without any observable sag at the opposite corner. It is, of course, far from a cheap method of manufacture, but its efficiency can scarcely be questioned.

In dealing with the gear we have confined our description to the gate lever control, and have not mentioned the old pattern, which is still made. As the gate control always goes with wheel steering, we do not propose to do more than mention the fact that the tiller-steered type is still frequently supplied. In the wheel-steered chassis a nut and lever reduction

gear is used of similar design to the well-known Malicet et Blin. Forked stub axles are used, the tubular front axle carrying stationary pivots, which are provided with cup and cone ball bearings at the bottoms and an ordinary single row self-contained journals at the tops. The front hubs are only fitted with two single-row journals, there being no provision for resisting thrust.

Taking the whole chassis into consideration, the most striking point in connection with it is the almost total disregard for cost of production which it exhibits. It would appear that in the original design the aim had been to have everything except one or two parts, in exact accordance with the designer's ideas as to what they should be for the perfection of performance. No doubt popular prejudice has an influence on the choice of the vertical engine, the wheel steering, the gate gear control, and one or two other parts, but common practice (as distinct from popular prejudice) seems to have affected only a very few items, the most noticeable being the absence of a thrust bearing in the front hubs. Perhaps the fact that only a few Lanchester features have been copied by other makers is owing to the great difficulty of copying any principal part without also copying the remainder, as almost every main portion exercises a great effect upon the adjacent members. Comfort, from the points of view of both passengers and driver, has been the prime consideration in the evolution of the Lanchester chassis, and the different aims of different manufacturers tend to confuse comparisons. It would, however, be very interesting to see what type of car would be created by the Lanchester designers in the event of their deciding to make entry for the honours of the racing track or hill climb.

THE POSSIBILITIES OF AIR COOLING.

By "Anglo-American."

A REGULAR reader of contemporary automobile literature can scarcely have failed to notice that the attitude of engineers towards the substitution of air for water cooling for automobile engines has undergone a considerable change. Not long ago it was scouted as an altogether visionary idea, but now the suggestion of its possibility but seldom excites derision, and more usually meets with acquiesence. The reason for this change is probably concerned with the success of the air-cooled aviation engines, which do not appear to suffer from overheating at all, and in fact, it has been made obvious that quite large engines can perform satisfactorily with a moderate air draught and no special air jacketing.

It has been hinted that the rotary cylinder engine may come to be used for car work when some of its peculiar drawbacks have been overcome, as they doubtless will be. However, there are double difficulties in the way of using a rotary cylinder engine for a road vehicle, because its inherent bad points, such as its enormous oil consumption and its free exhaust, have first to be overcome, and then there still remain some formidable obstacles connected with the design of the chassis. Therefore it appears likely, to the writer, that there will be cars with air-cooled engines in more or less common use before the rotary engine has reached a stage of development sufficiently advanced to make it worth while to give serious consideration to its adaptability, or otherwise, for road work.

Air cooling, from the point of view of the car user, is an improvement on water cooling, because it saves weight, and because it saves a certain small amount of trouble. These two points of advantage also appeal to the manufacturer, who has also the probable attraction of lowered cost of production. There is no particular disadvantage for the user, providing that the air-cooling system is as reliable as the usual water circulation pump, and the manufacturer has only to consider whether an air-cooled engine will meet with popular disfavour or not. Several recent inventions have demonstrated more or less conclusively that good results may be obtained by radical departures from standard practice, and there would probably be no difficulty in selling a car with an air-cooled engine, if it was put upon the market by a manufacturer of repute, and if it could be shown by means of an R.A.C. test, or otherwise, that it did not overheat.

It would not, however, be practicable to convert the design of a water-cooled engine to one for air cooling by the mere substitution of different cylinders, and the addition of a forced air draught. The part of an engine which needs to be kept most cool is the valve gear, and this is not easy to jacket with water, let alone with air. Efforts are made by most engine builders to obtain as free and rapid a circulation of water as possible under the valve seating, and sometimes even round the valve stem guides. Cooling only one side of the valve seating by water has been found to be insufficient if the engine is required to develop a high horse-power by comparison with its size, owing to unequal expansion causing the seating to become distorted, and it is obvious that with the higher temperatures which must prevail in an air-cooled engine, still greater precautions will need to be taken to keep the necessary even heat. It is more than likely that, to obtain complete success with air cooling, the valves should be set otherwise than in the usual side pocket. The obvious alternative is to place them in the cylinder head, but this means that they must be either in cages or some other more or less complicated structure, if the necessity for removing the cylinder every time it is desired to grind in the valves is to be avoided. There would appear to be possibilities in the Lanchester arrangement of horizontal valves placed on opposite sides of the cylinder, as this enables the exhaust valve to have its seating formed directly in the cylinder wall, and the incoming gases are of assistance in keeping down the temperature of the inlet valve cage, but the passages should be kept as close as possible to the seating in order to allow the valve stem guides to be exposed to air draught. Such an arrangement of valve gear presents one particular disadvantage, which is that the tappet cannot act so directly upon the valve, and longer links with more joints, and therefore more tendency to noise, are required.

Although it may sound unlikely, at first hearing, the author considers that the piston valve engine lends itself more to air cooling than the poppet type. Of course, this belief does not include the Knight engine, or any engine with sliding sleeves surrounding the main piston, but refers to the ordinary piston valve as used in the Hewitt engine. There is no difficulty in cooling the comparatively large cylinders of the present day motor cycles and the small pistons of a valve gear are working under much less severe conditions than the working piston of any air-cooled cycle motor, and it would be possible to construct an engine in such a way that the piston valves and their cylinders were entirely separate from the main cylinder except for the junction round the ports. There is no doubt but that were it not for the valves and gas passages water cooling would never have been thought of. That is to say, there would be no trouble in air cooling an engine so far as its piston and main cylinder are alone concerned. Therefore it appears reasonable to assume that it would be an advantage if the valves and cylinder were made separate, as far as is possible, and were cooled separately. To obtain this desired end the valves might be designed as separate parts with complete air space all round them, and then combined with the main cylinder with as little interference as possible with the cooling arrangements of either.

It is being realised that in a water-cooled engine it is advantageous to so proportion the various walls of the cylinders and jackets that their expansion will be equal all over, and this is still more important with air cooling. Every part should be as thin as it can be without sacrifice of strength, and all parts should be of the same thickness. It would be possible to construct an engine to meet these varying requirements by making the cylinder of steel tube and the piston, or poppet valve, chambers of a similar material. The heads of both main and subsidiary cylinders could be cast iron or cast steel, with or without radiating fins as might be found best, while the valve chambers and the cylinder could be connected by steel pressings welded in place, either electrically or by the acetylene process. Air should be admitted at the bottom of roomy sheet metal jackets enclosing both cylinder and valves, and having a central outlet over the middle of the main cylinder, with an additional pair of smaller outlets, one over each valve chamber head, leading up to join the main outlet an inch or two above its inner end.

As has already been said, it is the opinion of the author that the piston valves would be superior, because they offer a much larger surface to the comparatively cool walls which surround them than do the poppet type, and should

therefore part with excess of heat more rapidly. In either case it might perhaps assist cooling if the valves rotated as well as reciprocated, in order that any tendency to distort should be neutralised. Such rotation is not difficult to provide for, and in the case of a cam-operated valve is easily obtained by the use of a mushroom-ended tappet with the cam out of centre.

Some makers of air-cooled engines have fallen into the perhaps natural error of directing the draught upon the cylinder heads, the supposition being that as they were the hottest part, so they required the most cooling. This is a strictly accurate statement of the case, but the deduction, though correct, is not best followed by attempting to pass the air in a direction opposite to that of natural flow. Experiment has shown that an engine *can* be made to run with natural air circulation only, that is to say, with air admitted at the bases of the cylinder jackets and exhausted at the tops. The most probable trouble which suggests itself, if this procedure is followed, is that the centre of the cylinder head would overheat owing to the air passing over it (so to speak) without contact. The arrangement of outlets just suggested would combat this tendency. The central outlet should be the principal one, the supplementary outlets being for the purpose of increasing the total outlet area without making the central pipe so big that the middle of the cylinder head could fail to receive its necessary share of draught. Also, the more rapidly the air passed out through the smaller orifices the stronger would the induced draught be in the central one, and vice versa, tending to maintain the correct proportions between the quantities which should pass through each outlet.

It might be well to mention here that the natural inclination to put a high polish upon the outside of the air jackets would be distinctly detrimental to their efficiency. The inside and outside of the jackets and the outside of the cylinders should be coated with some dead black deposit, preferably a metallic oxide formed directly upon the surfaces.

An argument which is frequently resorted to by opponents of air cooling is that the power absorbed by the circulating fan would be a large percentage of the total power of the engine, but this is not actually the case. Considering an engine of moderate size developing, say twenty brake-horse-power, if it were cooled by water the power lost in the water would be about eighteen horse-power. Assuming that it would be as much as twenty horse-power, sufficient air would require to be passed through the jacket to absorb this quantity. Twenty horse-power is $33{,}000 \times 20$ ft. lbs per minute, and if the air in passing through the jacket is increased in temperature from 15 degrees to 115 degrees centigrade, then the quantity of air must be sufficient to absorb about 900 B.T.U. for 100 centigrade rise in temperature. This is equivalent to a little less than forty cubic feet of air, and this could be supplied by a Blackman type of fan of a quite moderate size. It is hardly possible to calculate the exact size of fan, because so much depends upon the resistance to the passage of the air offered by the jackets and pipes, but sufficient variation in fan output for experiment could be obtained by altering the speed at which the fan was driven.

This leads to another point, which is the necessity for the curve of quantity of air passing through the jackets to correspond with the best obtainable horse-power curve from the engine. The output of a fan is not in proportion to its speed of rotation, and if it is sufficiently powerful to keep the engine cool at low speeds, when the throttle is wide open and the engine giving its maximum power for the speed, it will be needlessly powerful at high speeds of engine revolution. Therefore some type of variable gear is required which will reduce the ratio of crankshaft to fan revolutions as the speed rises. It is not easy to see how this could best be accomplished, but there are many methods which will suggest themselves to readers, on a little reflection, and it is not necessary to enlarge upon them here.

It has been proposed that a waste product of the engine, that is the exhaust gases, might be used to create an induced draught through air jackets, and it might be possible to keep an engine cool by such means at high speeds, but as the size of the passages affects the strength of induction, it is most likely that the air current would not be rapid enough at slow engine speeds.

The experienced automobilist has become so accustomed to water cooling that he usually fails to appreciate its disadvantages, and air cooling is scoffed at by many, because they say it is not worth while. As a matter of fact, there is no simplification that is not worth while if no efficiency is lost by it, and air cooling, if properly carried out, would be more likely to increase efficiency than decrease it, while the fan and air jackets should be cheaper to construct and even less trouble to maintain than the best natural water system. To say nothing of the considerable saving in weight.

FITTING CAR-LIGHTING DYNAMOS.

THE great convenience of electricity for lighting both the interior and exterior of large cars, together with the great improvements recently made in the design and construction of powerful electric head lamps, has caused several dynamos to be put on the market which can be fitted to a car in conjunction with a small battery of accumulators, which they can keep fully charged as long as the engine is running, and these small dynamos have sold in fair quantities, notwithstanding their comparatively high price. In most cases the attachment of an accessory which requires to be driven from the engine is an awkward job, difficult to execute in such a manner as to be satisfactory under all conditions of weather, and also the fitting often costs quite a large sum, owing to the special parts which have to be made. As there is great likelihood of a dynamo lighting equipment becoming one of the fittings of the majority of large covered cars, it has been suggested that car manufacturers should make provision for the attachment of a dynamo on a platform similar to that usually provided for the magneto, with some form of positive drive which could be easily connected to the armature shaft.

The task before designers would be greatly simplified if the various dynamo makers could agree to make their machines to fit a standard platform, but, as it is, the dimensions differ widely, and two courses are open, there being but little to choose between them. One is to make the platform and drive to suit one particular dynamo, and the other to make accommodation for the largest machine. In the first case it would be possible to drill the platform for the holding down bolts, and to supply a wheel for the dynamo shaft which would render the fitting a mere spanner job. In the second case any dynamo could be fitted, but the platform would have to be drilled at the time of fitting, and a special part might have to be made for the drive.

Double the crankshaft speed is usually recommended as the best ratio of drive. This means that it is not sufficient to arrange a duplication of the usual magneto drive, and in fact, it makes the fitting of a gear drive rather awkward, as the dynamo pinion would require to be unduly small if it was to be driven off the distribution gear. A "silent" chain should be satisfactory, if properly enclosed, and it would usually be possible to accommodate the large chain sprocket on the crankshaft inside the crankcase, and to mount the small sprocket in some such position as the inside of a crankcase arm, so that the final drive to the dynamo could be made by a form of universal coupling. A gearing-up worm drive might also be used, and could be made to occupy rather less space than the chain. However, so much depends upon the engine, and upon the disposition of its other attachments, that it is only possible to indicate vaguely how a dynamo may be accommodated, the main points to be remembered being that the drive should be short, and entirely enclosed, to guard against the possibility of noise, which can hardly be avoided if any form of unprotected gearing is used. As a guide to designers we give hereunder the approximate overall dimensions of several of the dynamos most frequently fitted with the size of platform required to fit the base of each.

The C.A.V. machine, which has been one of the most successful, is made in three sizes. The smallest needs a space $8\frac{1}{2}$ins. long, $5\frac{1}{4}$ins. broad, and $6\frac{1}{4}$ins. deep, and the platform for it should be 4ins. by 3ins. The second size is nearly 10ins. long, with the same width and depth as the smaller size, and requires a platform 5ins. by 3in. The largest pattern is 10ins. long, $5\frac{7}{8}$ins. wide, and nearly 7ins. deep, and its platform needs to be 6ins. by 3ins. The centre of the armature spindle is $3\frac{1}{4}$ins. above the platform, in the case of the two smaller machines, and $3\frac{5}{8}$ins. for the largest. This means that if arrangements were made to take the largest dynamo, packing pieces would

be needed if either of the others were fitted. Also, the end of the spindle becomes more distant from the nearest end of the platform as the size of dynamo increases, so if any one of the three is to be attachable, the platform must be made longer than would be necessary to accommodate the largest, or different sizes of couplings would have to be provided. It seems rather a pity that these three machines of one make could not have been made interchangeable as regards platform and drive, for though the differences are not very great, they make the car designer's task considerably less easy than it would otherwise have been.

The Polkey dynamo is quite a different shape, as it is 14½ins. long and about 6ins broad, and 6ins. deep. It needs a platform 5ins. long by 6ins. wide. It is more nearly circular in section than some of these machines, and has a brass case (at the opposite end to the drive attachment) which requires to be taken off occasionally. Clearance for this operation must therefore be left. The makers also recommend that the form of coupling used for the drive should be such that it can be disconnected easily, if it is desired to run the car without using the dynamo.

This last recommendation is also made by United Motor Industries, Ltd., whose "Castle" dynamo is again of a quite different size. Its dimensions are 12⅛ins. long, 6⅛in. wide, 7½ins. deep, and the platform area necessary for fitting it is 9⅞ins. by 4ins. The distance between the top of the platform and the centre of the armature spindle is 3.35/64ins.

It is, of course, the ambition of each dynamo maker for his machine to become the standard pattern to which other makers will have to conform. Eventually, no doubt, the dynamos will become as nearly interchangeable as are magnetos at the present time, but with such considerable differences of form as now exist it will be possible for car manufacturers to exercise great influence over the dimensions of the final type, if they will agree to make standard provision for dynamo fitting on certain of their cars.

A FIVE HUNDRED TON PRESS.

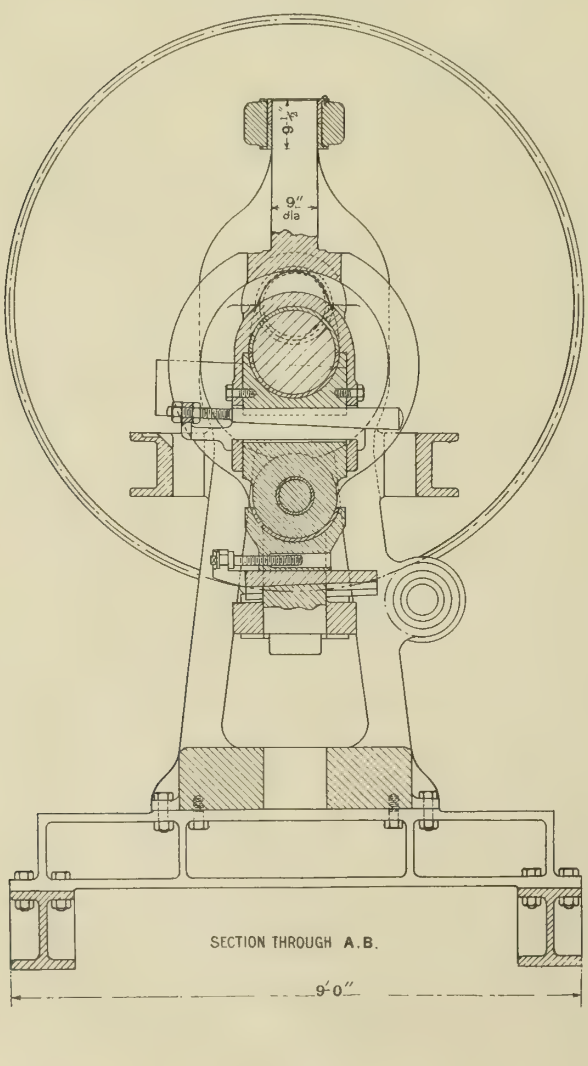
Messrs. Taylor and Challen have lately installed a special five or six hundred ton press in the Rudge-Whitworth works at Coventry for the purpose of finishing the hub shells for Rudge-Whitworth detachable wire wheels. These hubs are first rough drawn, and the press is only used to remove all roughness left in the drawing. The steel used for the hub shells is rather harder than the majority of steels used for pressed parts, which accounts for the great power required, the force which the press can exert being sufficient to ensure the piece leaving the dies quite correct to size.

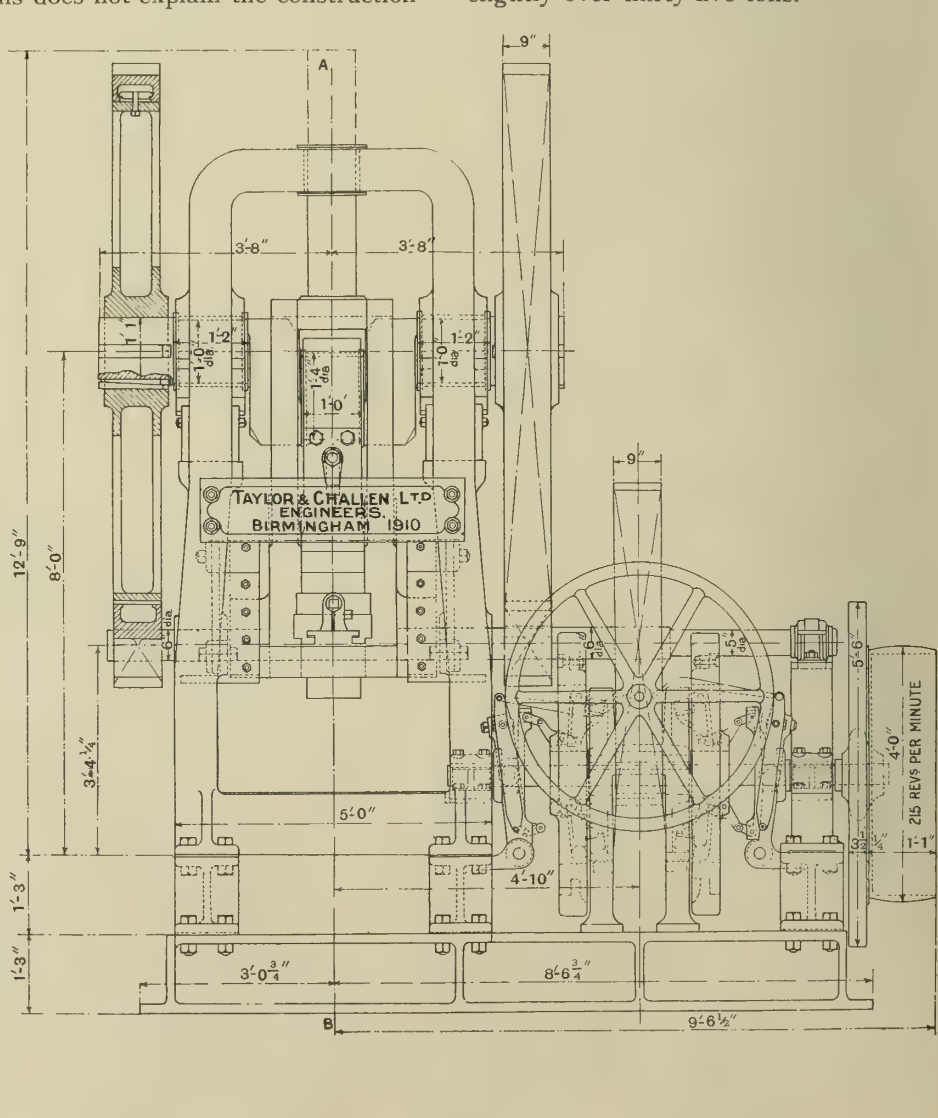
The frame is cast steel, machined for the reception for the phosphor bronze bushes of the crankshaft and intermediate shaft, and also machined on the bed. The main guides of the slide are cast iron, and are provided with an easy adjustment for taking up wear. The slide, with its projecting tail rod, is steel, and the last-named part is guided by phosphor bronze bushes in the top of the frame.

The crankshaft is forged from a single piece of Swedish steel, and the connecting rod is cast steel with bronze bushes, and is provided with a gib and cotter adjustment. The lower end of the rod bears upon a thrust block, immediately beneath which is a wedge, supported by the slide, and carrying the force block to which the tools are attached. This wedge is of importance as regards the work, for it enables accurate adjustment of the tools to be made.

The above-mentioned parts are mostly shown in the accompanying illustrations, but this does not explain the construction of the clutch so well. The latter is double-sided, with six wooden friction blocks on each side. It is slightly conical, and both sides engage simultaneously. The clutch pulleys are keyed to the boss of the pinion, which meshes with the spur wheel on the intermediate shaft, and the pinion is bored to a larger diameter than the shaft, so that it is always clear. On disengaging the clutch the pulleys and pinion drop against brake blocks contained in recesses in the supporting girders, thus bringing the machine to rest promptly. The sets of driving blocks can be adjusted separately by means of the expanding toggles seen in the illustration.

The principal dimensions are shown on the drawings, and the total weight is slightly over thirty-five tons.





CORRESPONDENCE.

No responsibility is accepted by the Editor for the opinions of correspondents, and the right is reserved to omit any portion of a letter if it is considered desirable to do so. Letters to be inserted with a nom de plume must be accompanied by the name and address of the writer.

ARCHED AXLES.

Sir,—There are one or two points in connection with the articles on "Universal Joints" and "Arched Axles," appearing in your June number, which we should like to raise, but before doing so we should like to express our appreciation of the great service to automobile engineering which you are likely to be doing by the bringing out your extremely interesting and valuable paper, which, we feel sure, all automobile engineers and designers will recognise as supplying a very long-felt want.

In the first article, on "Universal Joints," all that we wish to draw your attention to is the fact that in the adoption of the ball and socket type of torque tube end this firm were amongst the very first, if not actually the first, as this arrangement first appeared on our cars at the Olympia Show of 1907, thus preceding the similar arrangement on the car of Messrs. Crossley by more than two years.

The design in which the adjustment is made parallel to the axis of the torque tube is the original design found on our 45 Model, and still obtains on that type. The design with the housing divided in the horizontal direction, which is shown as our arrangement in the article in question, was adopted on our smaller cars, because in those types the gear box is placed at the forward end of the torque tube immediately behind the universal joint. This method of dividing the housing greatly facilitates the taking down of the gear box and rear axle as a complete unit. In the case of the 45 h.p. Model, in which the gear box is at the rear end of the torque tube, the dismantling difficulty encountered with the smaller cars does not obtain, and thus the original design is maintained. We may say with regard to wear and adjustment that though we have had this type of universal joint in operation on our cars for nearly four years past, we have not, up to the present, found the slightest indication of wear in that part.

With regard to the article on the arched back axle, we venture to think that the opinions you express there are somewhat more pronounced than the actual state of the case warrants. We think, for instance, that it is not true that there are absolutely no advantages to be gained by the process of arching the back axle, the fact being that in practice many useful points are brought out in this type of axle. In the first case, the act of tilting the wheels from the vertical position, which is possible with axles of this type, avoids the extremely ugly unmechanical appearance which is presented by a great many fine

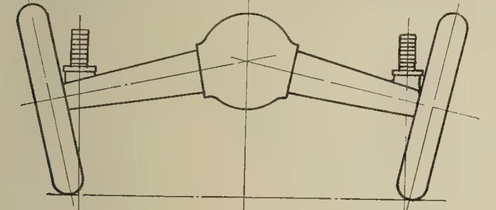


Fig. I.

cars which are to be seen running about the roads with the wheels doing their best to form a very bad letter "A." We think that this advantage alone is sufficient to justify the arching of the axle, but there are others in addition.

There is little doubt, for instance, that the arched axle is actually stronger than the straight axle, though we do not say that such increase of strength is very great. The accompanying diagram (Fig. I.) will, we think, serve to illustrate this point.

It shows that by inclination of the wheels the point of the application of the load is brought nearly under the centre line of the spring tables, thus relieving the axle casing of any load other than that due to its own weight, and while the inclination of the wheels required to accomplish this in actual practice would be too great to be convenient, still, the arched axle, as manufactured by this firm, does substantially reduce the load on the axle case.

The facility for using dished wheels, which is referred to in your article as being fallacious, we venture to think, has a greater advantage than you accredit it with.

In the first case, the tyre is inclined to the road at an angle which can be made to correspond with that of the camber of most of our English roads, and thus side rolling of the tyre is greatly reduced, and its life correspondingly increased, and, while it is true that the dished wheel is correspondingly weak to blows from the inside in proportion as it gains in strength to those from the outside, still, it is generally admitted that the blows and shocks sustained from the outside greatly exceed both in number and degree those from the inside.

There is another point, which, while not of necessity connected with the arched axle, nevertheless has played a greater part in its success than is commonly believed. It is impossible to prevent the strongest axle from deflecting in a greater or a less degree, and this deflection, from the form of structure of the axle, will take the form shown in Fig. II.

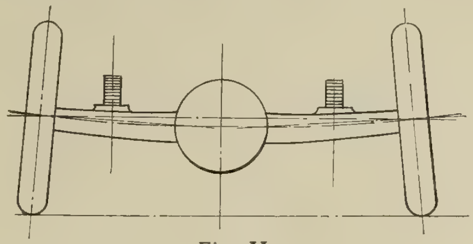


Fig. II.

If then the shafts are fixed at both ends, as is so commonly done, not only are the shafts themselves subjected to considerable bending moment, but the wheels of the differential gear itself are liable to be subjected to a very undue load.

We think it will be admitted from this that the provision of universal joints between the differential and the road wheels is likely to materially prolong the life of the whole of the axle driving gear. This in itself, we think, constitutes justification for the universal joint in the axle, and thus, since this is really the only additional expense incurred in the manufacture of the arched axle over the straight axle, we think this removes any objection that can be found to the arched axle.

We may say that there is little doubt where this universal joint should be, viz., whether at the differential wheels or in the hub, as we think consideration of the diagram of the axle will indicate where the greatest deflection of the shaft is likely to take place.

It may be urged that the presence of the universal joint in the shaft will cause an irregular angular velocity of the road wheels, and thus produce scrubbing of the tyres, but when we consider that the average camber of English roads is approximately three per cent., or rather under two degrees, it will be seen that no greater inclination than this need be put on the shafts, and that then the irregular angular velocity put on the wheels will be extremely slight.

We thus venture to think, in respect of your article, in the first case, that there are no marked disadvantages against the arched back axle, since practically the only additional expense is the universal joints, which we believe to be necessary in either case, while the above substantial advantages can fairly be claimed for this type.

THE SHEFFIELD SIMPLEX MOTOR WORKS, LTD.

MAKERS AND INVENTORS.

Sir,—The article on carburation in your June issue has led me to ask whether you would draw attention to the great difficulty which inventors of carburettors encounter when endeavouring to bring them to the notice of manufacturers. I know several instances where there has been no question concerning the good qualities of the particular carburettor, and no question concerning its superiority over the one used by the makers to whom it was submitted, yet it has never been given a fair trial.

Of course, for every inventor with a sound idea there are hundreds with unsound ones, and it would be impossible for manufacturers to make experiment with every device which is pressed upon them, but when a man comes to them with proofs of the claims he makes it is extraordinary that he should be unable to obtain proper consideration.

In this particular matter of carburettors it is being proved every day that the great majority of standard pattern cars can have their petrol consumption reduced, and often their pulling power improved, by the substitution of a better carburettor, and there are several firms who do considerable business in fitting new carburettors to cars both new and old. Recently much improvement has taken place with regard to the carburettors fitted as standard by the leading manufacturing firms, but the fact remains that there are several excellent inventions the adoption of which would make a real improvement in the performance of most standard cars, but their patentees have tried unsuccessfully for years to get them taken up by any firm.

Often, of course, it is a matter of price, but it is a short-sighted policy to save a few shillings on a part when the expenditure would be returned in a few weeks at most, through the economy in fuel effected. This question of fuel consumption, and oil consumption also, is taken much more notice of by makers of industrial vehicles than by pleasure car manufacturers, because the user of a van or waggon takes an extremely keen interest in anything which will tend to reduce running costs. Thus the inventor has more scope on this side of the industry, but even here he is far too often met with conservatism of that most bigoted type which cannot believe in any new thing.

To make proper tests of really promising inventions could not fail to be of assistance to any firm, and it would require neither much time nor much acumen to pick the grain from the chaff.

ECONOMIST.

REPAIRS.

Sir,—The *Automobile Engineer* will, I am sure, be widely appreciated by everyone connected with the motor industry; it fills a long-felt want. I think I shall not be alone in hoping that while the primary object of your paper is the design and construction of automobiles, you will not at the same time exclude articles on repair work. The repair man is often confronted with difficulties greater than the manufacturer has to contend with. In the course of his work, he has to deal with many different makes of cars; he has not the special tools and jigs of the maker, and is often called upon to repair, when for various reasons, replacement is not feasible, or perhaps possible. As you remark in your introduction, automobile engineering is a distinct branch, and the general engineer, however good he may be, should recognise that there is always much to learn, and there are many jobs which he may do well in one way, which others may do a little better by other methods. Short articles written by practical men on repairs and methods they have actually used would be very welcome, and also, I think, be exceedingly helpful to the manufacturer, for it is in the repair shop more than anywhere that faults and inconveniences of various designs become evident, and it is to the manufacturers' interest that repairs when they become necessary may be executed as speedily as possible.

Wishing your paper every success,

J. COOPER.

[We should appreciate the opinions of other readers with regard to this matter.—Ed.]

MAGNETO SYSTEMS.

Sir,—I notice in Mr. Walford's synopsis of Automobile Patents, in dealing with the Brooks and Alston Patent 15803/09, he says:—"Another method not so commonly used is to suddenly establish the current in the primary winding, this being until recently the case with the Eisemann magneto." Please note that the words "until recently" are not correct. We are still manufacturing and selling the Eisemann magneto of this construction, that is to say, it is a low tension magneto with separate induction coil, the high tension current being developed in the manner named by Mr. Walford. For instance, Mr. Rolls used an Eisemann magneto of this form of construction in his recent cross-Channel flight. Moreover, the Packard Co., of America, use nothing but this form of construction, and many other firms on the Continent, which it would take too much of your space to mention. The high tension current obtained from this form of construction is of much greater voltage in magnetos of self-contained pattern, which we also manufacture, and is especially useful for high compression engines.

STANLEY J. WATSON.

CARBURETTORS.

Sir,—I am naturally much interested in the comments and criticisms on my article on carburettors, which appeared in your issue for June, and in Mr. Bourne Dale's remarks. I am afraid, however, he has misread some of my statements, and in several cases has not quite grasped what I meant. Perhaps I was scarcely explicit enough.

At the commencement of his letter he suggests that I referred principally to older types of carburettors working at low differences of pressure, and throughout his letter he leads one to believe that a desirable state of affairs is to maintain a constant pressure head operating the jet. I really fail to see how such a system could be regulated, as a constant flow of petrol would result, and naturally this is not to be desired. At slow engine speeds, and consequently when the demand for carburetted air is low, many modern carburettors are so arranged that the incoming air is more or less concentrated round the jet, and thus the negative pressure, if we may so term it, in the vicinity of the jet orifice is artificially increased above its proportional value to ensure a steady flow.

In discussing the effect of air velocity perhaps I have been scarcely clear. What I mean in every case is that "as the flow of petrol from a jet orifice bears a definite ratio to the difference of pressure between that in the float chamber and that in the vicinity of the orifice, the factor controlling this petrol flow is primarily the air velocity in the vicinity of the jet orifice."

Of course, the magnitude of this air velocity depends on the volume of air entering the engine in unit time and the area of the air passage; and in cases where the whole of the air passes round the jet, when the size of this air passage is of fixed dimensions, the velocity of air bears a direct relation to engine speed and throttle opening.

It appears therefore that a fixed value for the partial vacuum as suggested by Mr. Bourne Dale would have to be supplemented by some arrangement forming an ejector, to induce a sufficient petrol flow at high engine speeds. In my article I did not consider any ejector arrangement at all, but simply reckoned the petrol flow as being governed by the differences in the two pressures as before explained.

It follows therefore that Mr. Bourne Dale's criticism of my remarks about the air slots does not hold good, for by "properly proportioned" I naturally mean that, as in the ordinary way, the petrol flow is not directly proportional to the differences of pressure acting, but increases in a greater ratio at higher pressures, so should the openings for air inlet be arranged to admit a larger volume of air at such times, as the petrol flow increases. For instance, when the throttle is full open and the engine speed still rises beyond the point of normal petrol flow, a further movement of the throttle will open the inlet air slots and admit a proportionately greater volume of air. This argument, of course, can be applied at any position of throttle opening.

From the last remark at the bottom of page 60 it is evident that one of my chief points has gone astray, as I endeavoured to show that throttle opening and petrol required did *not* bear any definite relation, as the effect of road resistance will always counteract any possible relation of this kind.

It appears to me that any carburettor designed to give a difference of pressure such as is indicated (14in. of water) must unduly throttle the incoming charge, and for this reason alone will be undesirable. In a well-designed system, such as I advocate, I am sure Mr. Bourne Dale will find that at 20 miles an hour car speed the cylinders will receive a charge as dense as his cylinders do at the lowest engine speed, whilst at the high speeds now so usual my system will be infinitely preferable.

All I can say is that, in my carburettor experiments, the results I obtain in practice agree very well with the theoretical calculations made by means of my curves, and that I have always found that the velocities of air which I advocate through the carburettor are a very good average for general work. The small curve reproduced in my article was really of interest if well understood, as it shows that the particular carburettor gives an equally efficient mixture under widely varying conditions. I do not see any vagueness about it, and hope Mr. Bourne Dale will give it further study.

In any carburettor problem certain approximations must be made—not guesses—depending upon the general design of the system, size of valves, shape of inlet pipe, and position of the carburettor, but these are not difficult to arrive at with a fair amount of accuracy.

ROBERT W. A. BREWER.

AUTOMATIC ENGINE STARTING.

Sir,—Your correspondent, Mr. Schofield, points out one of the very many omissions in my article on the above subject. Omissions were conditioned by the fact that it was an article on self-starters and not a book on them.

I agree that shunt-wound machines are used for charging accumulators, but I have also known of compound-wound machines being used. In the case under consideration, a machine is required for another purpose besides that of charging, viz.: starting the engine, and an electric motor, if nothing else than a large starting effort were required of it, should be series-wound. In the compound-wound machine, which is a combination of both shunt and series, a machine suitable for the double purpose might be found provided the difficulty of preventing the reversal of polarity could be overcome.

Mr. Schofield says that permanent magnet machines are never used for this purpose, but in this he is mistaken as there is at least one such machine on the market, and in my humble opinion, the principle of using permanent magnet machines for the special purpose of charging accumulators from the engines of motor cars is, in view of the varying speed at which the dynamo is driven, a correct principle to adopt, because of the less rapid increase of output with the speed.

J. DALRYMPLE BELL

INCHES AND MILLIMETRES.

Sir,—Mr. Toby asks you to give all measurements in metric and not in decimals of an inch. May I ask you to respect the widely used and most valuable and convenient thousandth of an inch, and if you must restrict yourself to one system only, the necessity for which is not obvious, to let it be the decimal of an inch system? Mr. Toby's letter implies that only metric would relieve his brain effort. To me, and doubtless to many others, to give both would be quite unobjectionable, leaving us to use which we prefer. But, in any case, do not mix them up in the way in which, as an example, Centigrade temperatures are so often mixed up with British thermal units.

HENRY LEA.

CONSULTING ENGINEERS.

We have been requested to give publicity to the following report of a meeting held recently in London, and we do so with all the greater pleasure because the title of consulting engineer, with respect to automobiles, has been assumed by a number of men who are quite unqualified. The formation of an association such as the one suggested would be a protection to many who are true consulting engineers, and would probably raise the status of automobile specialists very considerably.

The meeting was presided over by Sir William Preece, K.C.B., F.R.S., the past President of the Institution of Civil Engineers, and was held to consider the desirability of forming an association of Consulting Engineers, the object of the association being (*a*) to form a recognised group of bona fide independent consultants who would constitute a body for the protection of their interests and the interests of the public generally; (*b*) to improve their status and professional position, following the examples of the Council of the Bar, of the Medical Council, and of the Chartered Accountants.

Mr. Midgley Taylor, who presided in the absence of Sir William Preece at the beginning of the meeting, said that the subject had been before several consulting engineers for the past two years or more, and as some initial step had to be taken the members who originally took the subject up formed themselves into a Provisional committee with the result that the present meeting was called to confirm (or otherwise) the action taken by the committee in the past. The object of the meeting was to see whether the general idea of the formation of the association met with the approval of the consulting engineers of this country. It was necessary, in the interests of the public as well as in their own, that the genuine consulting engineer should be distinguishable from those who were not genuine consulting engineeers, but had trade interests. He felt that the public also wanted protecting. In matters of health or law, or finance, the safeguard was that the doctor or solicitor or accountant who was consulted was a member of a recognised body, and it was felt that the public should have the same protection in engineering matters.

Under those circumstances he felt they should endeavour to form an association such as would not only protect the bona fide consulting engineer, but would also protect the various persons in this country who were anxious to get a properly qualified man to advise them on engineering works. The committee hoped that the meeting would result in the formation of the association, but at the same time they wanted to have an open expression of opinion from everyone whether they were favourable or otherwise. He then called upon Mr. Swinburne to propose the formation of the association.

Mr. Swinburne formally proposed that the association should be formed, and asked the meeting to look at the matter as broadly as possible, and all work together for the one end.

After some discussion, Sir William Preece, as Chairman, said that he was in entire sympathy with the movement, and remarked that what was wanted was a very strong committee with a strong chairman to carry the matter to a successful conclusion. He put the resolution to the meeting that the association be formed, and this was carried unanimously. It was then suggested that the following gentlemen should be asked to form a committee:—Messrs. Robert Hammond, C. Hunt, B. M. Jenkin, Baldwin Latham, S. R. Lowcock, E. L. Mansergh, W. M. Mordey, W. H. Patchell, Sir Wm. Preece, Henry Rofe, J. F. C. Snell, E. H. Stevenson, James Swinburne, Midgley Taylor, Henry Woodall.

This committee was elected unanimously, and Mr. A. H. Dykes, of 1 Victoria Street, Westminster, S.W., was elected as honorary secretary.

PUBLICATIONS.

THE DESIGN OF PETROL MOTOR VANS.—This reprint of the paper read by Mr. T. B. Browne, before the I.I.A.E., on February 9th of this year, has been published by the Institute at a shilling, with a paper cover, and may be obtained from the secretary. In its present form the paper should be of assistance to all designers who are engaged on light van work, for besides describing some of the most salient features of several well-known vehicles, the author makes some suggestions worthy of careful consideration. Also, Mr. Browne has not been afraid to state obvious facts, in direct language, and though this gives the paper a somewhat elementary flavour, it is useful, as it emphasises the difference between the conditions of working of vans and pleasure cars. (The Incorporated Institute of Automobile Engineers, Caxton House, Westminster, S.W.)

THE ART OF AVIATION.—This volume, by Robert W. A. Brewer, is a rather curious medley of information concerning the various types of aeroplanes now in use, their engines and parts, and the actual art of handling the machines. The author has endeavoured to explain the theory of flight with aeroplanes without the use of high mathematics, and, on the whole, he has succeeded very well. The principal engines of the day are described fully, and as the illustrations of their parts are very numerous it is probable that this section of the work will be of real assistance to engineers. The Gnome engine receives perhaps rather more than its due share of space compared with that given to the other engines, but this may be excused on the ground that it has more points of difference from the standard type of automobile prime mover. The actual instructions for learning to fly, with either monoplanes or biplanes, we do not feel qualified to criticise as regards accuracy, but they are certainly clearly expressed, and Mr. Brewer gives a good reason for every one of the actions he recommends. The book should be useful to non-technical men who are anxious to fly with as little trouble as possible, and it will be read by engineers for the valuable information in the section on engines, if not for other parts as well. (Crosby, Lockwood and Son.)

A NEW BALL BEARING.

It is well known that the strength of a ball bearing is increased by increasing the number of balls in it, but that it is difficult to make a strong cage if the balls are very close together. In the Skefko bearing the difficulty is got over by using a double row of balls, staggered in a central solid cage. Fig. I. shows a journal bearing made on this principle and makes the con-

struction quite clear. The outer race is so shaped that all parts of it lie on an imaginary sphere having the centre of the whole bearing for its centre, and this means that the bearing is quite unaffected in working if the inner and outer races are not true with each other. Fig. II. shows a double thrust bearing of the same make, and this is contained in a mounting box for the purpose of enabling the shaft to run with slight angularity to the housing. It will be seen that the thrust washers proper are rounded at the

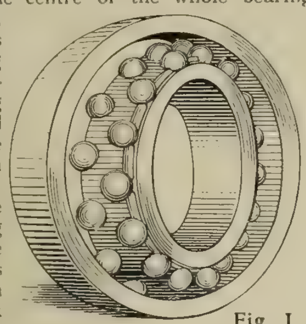


Fig. I.

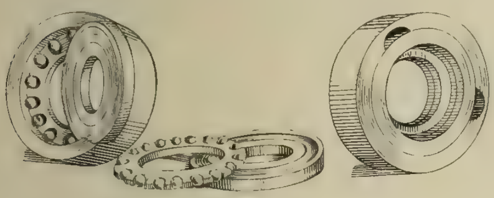
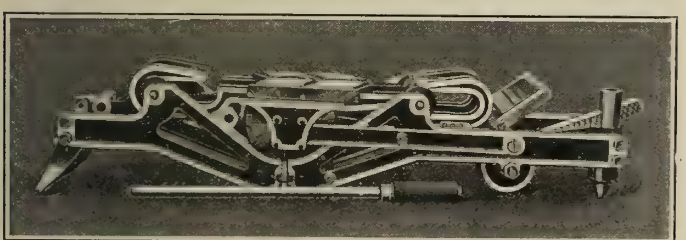


Fig. II.

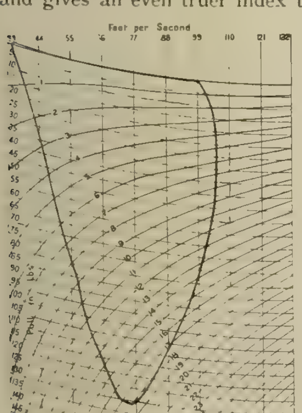
outer edges, and the inside of the housing is shaped similarly. All Skefko bearings are made from Swedish steel and are not case-hardened, while special care is said to be taken to ensure equal hardness of balls and races. We understand that these bearings are now undergoing tests at the hands of several automobile manufacturers, and if the promise of their design is supported by experience we should think the new bearing will be in considerable demand.

A NEW PATTERN DYNAMOMETER.

Being quite reasonably accurate, the Acer dynamometer is slowly being recognised as a useful accessory to any repair shop, as the ability to take a horse-power reading at any speed by simply placing the instrument beneath the fly-wheel saves a perceptible amount of road testing, and gives an even truer index to the condition of the engine. The old design has lately been improved in detail, and the latest pattern is shown below. An aluminium cradle is now employed (to carry the cork pads, which are brought into contact with the fly-wheel), instead of the chains used on the first model. The light chain which transmitted motion to the recording drum has been replaced by a rack and pinion mechanism, and the speed indicator, which controls the pencil, has also been materially simplified by a reduction in the number of its component parts.



The new pattern Acer dynamometer.

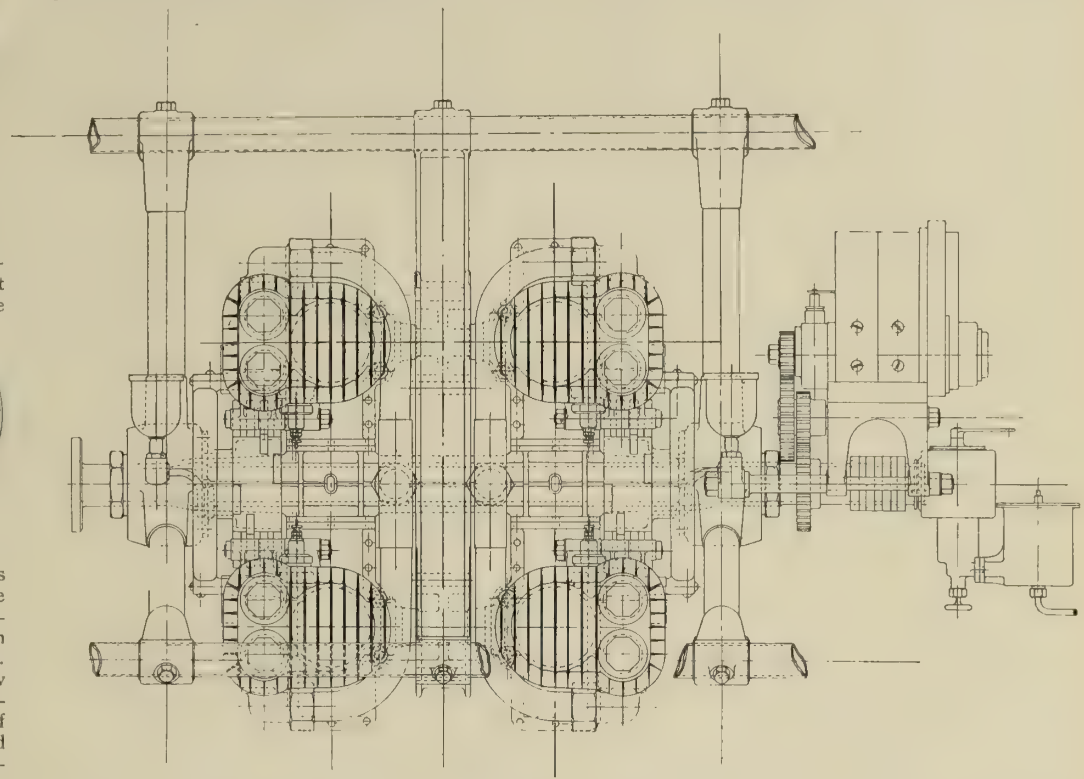


Portion of a chart taken by the Acer dynamometer.

The whole dynamometer is distinctly lighter and less bulky than the design by which it was preceded, though improvement in this respect was perhaps scarcely necessary for the average methods usually employed in commercial testing of horse-power.

A SLOW-SPEED AERO ENGINE.

The engine described hereunder is the invention of Mr. D. Keizer, and has been designed with a view to producing a rotary motor combined with a speed reduction gear in a compact and convenient form. The fixed portion of the engine is a tubular frame, which carries the main bearings, the carburettor, the magneto and the oil tank. This frame also has a large annulus secured to it. There are eight cylinders arranged on two crankshafts, with a pair of opposed pistons connected to each crank pin. At the centre of each crankshaft there is a pinion meshing with the annulus.



The Keizer engine.

Thus as the crankshafts revolve the pinions on them roll on the annulus and draw the crankcases and cylinders round at a speed depending upon the number of teeth on the annulus and the pinions. If the pinions have half the number of teeth in the annulus, then the whole engine will revolve at half the crankshaft speed. The crankcases are solid with the main driving shaft, which runs on the main bearings in the frame, and it is to this shaft that the propeller is supposed to be attached.

The valves are very little different from the usual pattern, and are operated by cams cut on the outside of a pair of sleeves, secured to the frame, and situated at each end of the main shaft, just inside the brackets. The valves are not balanced, the springs being relied upon to close them against centrifugal force. The Gnome idea of passing mixture up the inside of a hollow shaft has been adopted, and the ignition leads are also taken in the same way from high tension slipper rings beneath the stationary magneto. The mixture is taken to the valve pockets through pipes leading from the hollow shaft, and the ignition wires are similarly led out through holes.

Lubrication is performed by gravity feeding oil to the inside of the crankcases in similar manner to the Gnome, so it is to be assumed that castor oil would have to be used, but the oil consumption might reasonably be less than that of the latter engine, on account of the slower speed of revolution. We understand that an engine is at present under construction and that it may be expected to be ready for trial shortly.

MISCELLANEOUS.

THE COVENTRY CHAIN CO. inform us that they are making two special varieties of chain for camshaft drives. Chain transmission for the valve shaft has proved quite satisfactory on Knight engines, and some thousands of Coventry chains are now in use on Daimler cars. Especial care is said to be taken regarding accuracy of pitch, which is guaranteed not to vary more than 0.014ins. per foot of chain. The two pitches are ½in. and ⅝in. respectively, and sprockets of any dimensions can be cut to suit.

AN ADJUSTABLE ENGINE STAND has been introduced by Messrs. Brown Bros. It is sufficiently strong to carry almost any automobile engine, and is intended for use in repair shops. It has roller feet, enabling it to be wheeled about, and it is also provided with holding-down brackets for use when an engine is to be tested under its own power.

CATALOGUES RECEIVED.

MACHINE TOOLS.—James Spencer and Co., Manchester, have recently issued a complete list of their lathes, boring, milling, planing, slotting and drilling machines, and their various accessories. Many of the machines are especially adapted to the needs of an automobile factory, particularly some of the lathes and milling machines. One or two of the radial drills, and a multiple spindle drilling machine would also appear to be very useful tools.

MACHINE TOOLS.—J. and P. Hill, Sheffield, have issued a special catalogue dealing with their high speed lathe designed for general work, and of a capacity equal to handling most of the heavier parts of automobiles.

FRICTION CLUTCHES.—The Saver Clutch Co., Ltd., Manchester, have prepared a circular describing their speciality. The illustrations are good and the matter clear, while the instructions for adjusting the clutch are particularly lucid.

TWIST DRILLS.—E. G. Wrigley and Co., Ltd., Birmingham, have sent us a convenient tabular list of their twist drills and shell drills. It contains a useful set of curves showing the correct speeds and feeds for drills up to 3in. diam. on mild steel.

MACHINE TOOLS.—Webster and Bennett, Ltd., Coventry, have two new lists devoted to their horizontal boring machines and their multiple spindle drilling machines. In the last-named list there are some two-spindle and four-spindle machines, which should be very useful for automobile work, and we understand that several are at present employed for cylinder work, being particularly adapted for boring and facing valve seatings and pockets.

LUBRICANTS AND LUBRICATORS.—W. H. Willcox and Co., Ltd., have recently published a new catalogue intended principally for the retail motor trade, but none the less useful to manufacturers. In addition to ample description of the articles mentioned above which are principally intended for automobile use, we notice numerous repair shop tools and appliances, and also a number of small tools, such as electric hand drilling machines, and similar small grinders. The list should be useful to the drawing office for the particulars of lubricators, and to the works for certain of the tools.

BALL JOURNALS AND END THRUST.

A few points, which are liable to be overlooked, in connection with ball bearings, when employed in engine design.

It is a matter of common knowledge that several motor manufacturers of repute have made experiment with ball-bearing crankshafts, and that they have in many cases returned to the use of plain bearings. It is hardly too much to say that at the present time there is a general opinion that ball journals are not satisfactory for positions where they are likely to be subjected to shock, but whether this is true is by no means proved. Manufacturers of ball bearings are unanimous in saying that there is no reason for the failure of their journals, except unfair treatment, but, though this may be discounted simply because a man cannot be expected to condemn his own productions, it should not be forgotten that the above-mentioned makers are equally interested in the satisfactory service of their bearings, and therefore would not recommend them for a purpose for which they were really unsuited.

Also, ball bearings are maltreated by automobile builders very frequently; for instance, there are still many cars with a pair of ball journals in the hubs and no provision whatever for resisting thrust, and under such circumstances no standard bearing can possibly last for more than a quite small percentage of what its life ought to be.

It might be suggested as more than likely that this matter of axial load on journal bearings has been the cause of the failure of more than one ball-bearing crankshaft, because it is too often assumed that the usual type of bearing can withstand a small amount of end thrust, whereas it is not really and properly constituted to resist the very smallest load in this direction.

In theory, and very nearly in practice too, the balls in any ball bearing of ordinary type make point contact with the inner and outer races: that is to say, there are two lines, one on the inner race and one on the outer, along which the balls are designed to roll. When end thrust is applied to a single or double-row journal the races must move relatively until the balls are in compression in a plane not at right angles to the axis of the shaft. It is easy to see that there can be no endwise resistance until this has taken place, and equally easy to see that the line of compression through any ball cannot alter unless its points of contact with the two races alter also. It is also easy to see that the ball will be in a position of great mechanical disadvantage when end thrust is applied to the bearing.

Thus if there is any end stress, however slight, it follows that the bearing must inevitably become distorted, and such distortion must likewise result in a widening of the ball path on the races, and so in slackness of the bearing as a whole. This is soon followed by uneven wear, and finally by disintegration of the bearings.

That the failure of ball bearings on crankshafts could be due to end thrust may seem a somewhat far-fetched theory, but it can scarcely be so regarded if the matter is investigated carefully. Firstly, there is always danger of thrust arising in the clutch, for even if it is designed to give no thrust when engaged, usually quite considerable force can be applied by the clutch pedal and may have to be resisted by bearings external to the clutch when the latter is held right out, as is frequently the case.

In two cases of well-known car makers, who tried and abandoned ball bearings, the clutches were of the leather cone type, and their thrust was resisted by a ball race placed between the flywheel and the crankcase.

Now this arrangement would be all very well as long as this heavily-loaded washer remained of its full original thickness, but as soon as any wear took place it could not but allow some longitudinal movement which would bring stress on the journal bearings. This may have been the reason why a certain batch of British cars which went through the first Herkomer trial behaved quite well for the first few months of their life, but afterwards broke the balls in the main races. The cars in question had an arrangement of clutch just as described, and the thrust washer had to carry a very heavy load indeed. The symptoms certainly would seem to point to such an explanation of the trouble.

Another factor which may also have contributed to failures is that most ball-bearing crankshafts are of the three-bearing variety, and the bearings are usually housed in aluminium. The crankshaft not supported between every throw must spring a little, and this small amount of spring, while it might not affect a plain bearing at all, might be detrimental to a ball bearing, and the writer says this in full knowledge that designers of ball bearings allow for this action. The housing in aluminium also renders it none too easy to lock the outer race firmly in position, if the design should render this desirable.

All things considered, therefore, it seems not improbable that a good many failures (where end thrust from external sources has not been the cause) may be due to this springing tendency, of comparatively ill-supported cranks, displacing the races to such an extent as to subject them to wear and stress "on the cross," or, in other words, not on their legitimate lines of contact.

It is, of course, a moot point whether ball bearings when used for crankshaft work give sufficient advantage over plain journals to recompense for their considerable extra cost, but that is a matter which requires the assumption that bearings of a reasonable size can be used with success, and to this many will probably refuse to accede.

Certainly the difficulty of designing a bearing of moderate size that will slip over the crank cheeks, for supporting the middle crank shaft journals or for the crank pins, is very great; for no designer now dreams of splitting the ball races into two halves, top and bottom, as was done by some designers in the earlier days of the application of ball bearings for crank shaft work.

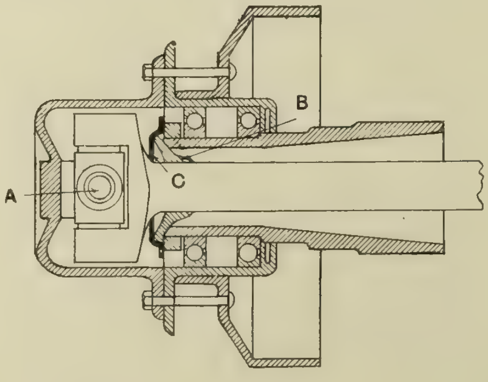
It would be interesting if some of those who have made successful use of balls for engine work would say whether they took peculiar precaution to protect the bearings from end thrust, and to ensure their remaining in their exact original positions, and it may be significant that in several motor cycles where there is no question of thrust, ball bearings have proved extremely satisfactory.

RECENT AUTOMOBILE PATENTS.

By Eric W. Walford, F.C.I.P.A.

Axle Construction.

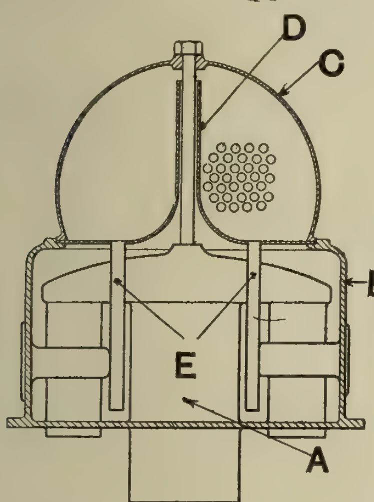
The invention refers to the maintaining of oil-tight joints at the ends of the driving axle lengths, particularly in cases where electric or other rotary motors form part of the axle. The features of the invention are, however, otherwise applicable. The driving axle shaft is attached by means of a universal joint at A to a box bolted to the wheel, and the wheel is mounted upon a tubular axle member in such a way as to obviate the driving shafts being subjected to bending stress, according to the drawing. To maintain the bearings and the universal joint A well lubricated the box and other parts are made oil-tight. On the driving axle length is mounted a spherically-shaped disc B, which is so designed as to butt against a similarly-shaped face on the fixed axle portion on which it moves, and where it is retained by a spring disc C. In this manner freedom of movement for the axle is provided without possibility of the lubricant escaping, while, incidentally, a greater flexibility of transmission is obtained.



Lentz-Getriebe and H. Lentz. No. 17,837/09.

A Radiator System.

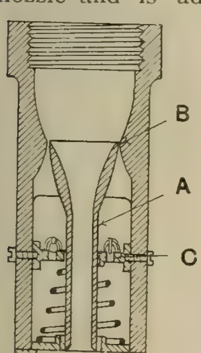
This invention refers to a somewhat novel arrangement of radiator, and it is shown as applied to a piston valve engine of the Hewitt type, such as was exhibited at the last Olympia Show. The cylinder A, which has a piston valve chamber at each side, is, together with the valve chambers, completely enclosed by a water jacket B, which is open at the top. To this is attached a water tank C, which is continuous for the length of the engine, and is fitted with longitudinal cooling tubes as shown. The tank is held down to the engine by means of the central bolt illustrated, the joint being suitably packed. The central bolt is surrounded by the riser pipe D, through which the hot water enters the radiator, falling past the radiator tubes, and issuing cooled through the pipes E into the space between the cylinder and the valve chambers. It will be gathered that external water piping is dispensed with, and a good head of water obtained, so as to ensure thermo-syphon circulation. The invention necessitates the employment of a special form of bonnet, and it would appear that some difficulty might be entailed in maintaining a tight joint between the radiator and the water jacket if a single central bolt only is to be used, as stated in the specification.



J. M. Hewitt and Hewitt Engines, Ltd. No. 13,230/09.

A Carburettor Valve.

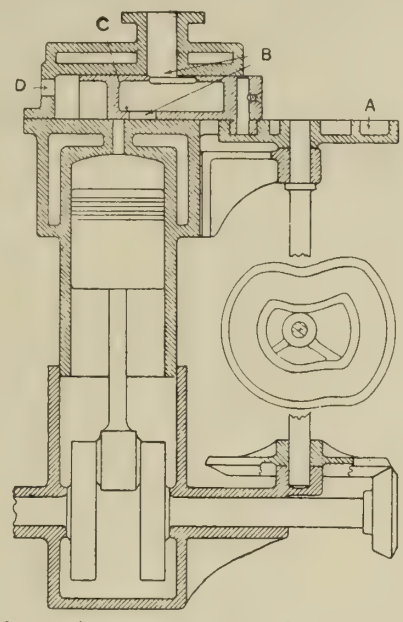
The carburettor is provided with a choke tube A, which surrounds the jet nozzle and is adapted to rise under increased suction to allow a varying quantity of uncarburetted air to pass along the annular passage B around the jet. To prevent the movable choke tube oscillating owing to the variations of pressure, a simple dash-pot arrangement is provided. The retaining spring abuts against a ring C, which practically constitutes one of the end covers of the dash-pot, and is formed with transverse holes, upon which seat ball valves. These permit the air beneath the rings C to escape, but trap it on the return, and acts as a dash-pot preventing chatter of the choke tube.



J. B. McVail. No. 27,064/09.

A Slide Valve.

In the attempts to produce a satisfactory slide valve engine, it is interesting to notice how closely some constructions approximate to the original Otto slide valve. In the case under consideration a cam groove on a half-speed shaft reciprocates the sliding valve, which is formed with exhaust ports B and an enlarged web at C, which is adapted to cut off the inlet port, and also close the cylinder port

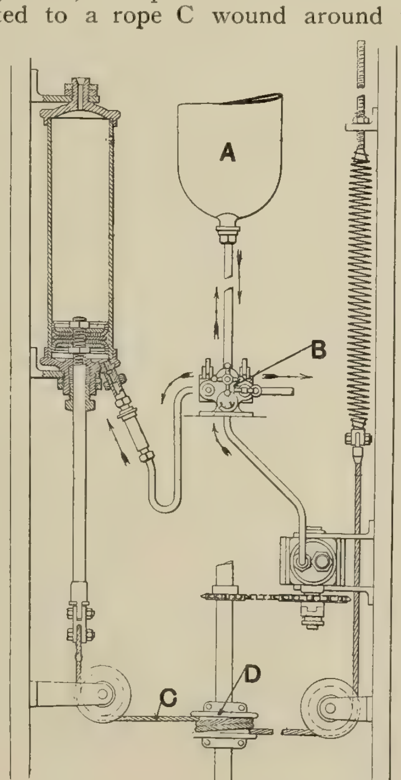


during firing and compression. For the exhaust stroke the valve is moved over to the left-hand side, bringing the ports B opposite the cylinder port and exhaust pipe. The valve is D-shaped or rectangular in cross section, and arranged in a water-cooled valve chest.

W. and W. C. Ward. No. 15,514/09.

Compressed Air-Starting System.

Compressed air in the tank A is supplied through a distributing valve B into a cylinder, the piston of which is connected to a rope C wound around the

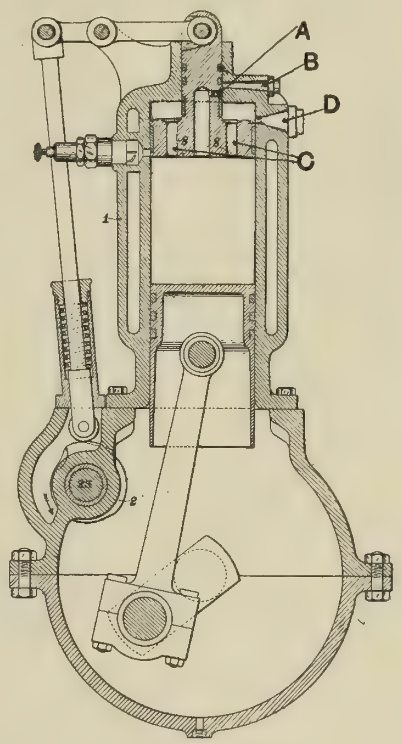


pulley D on the driving shaft, its free end being attached to a spring. The pulley is split so as to be easily applicable to existing shafts, and it has a free-wheel mounting on the engine crank shaft. The crank shaft drives a compressor maintaining the pressure in the tank A, and this compressor is driven through a clutch. To start, the distributer valve is opened, supplying compressed air beneath the piston of the starting cylinder. This pulls on the rope C rotating the pulley D, and turning the engine. At the end of each stroke exhaust takes place through the exhaust port, so that the pressure falls and the spring predominates, drawing back the piston. The gas below the piston passes through the supply tube, and exhausts through a special valve in the distributing chamber. The distribution chamber can be caused to supply air for tyre inflation, etc.

J. A. Ageron and Vicomte A. de Charpin. No. 15,903/09.

A Piston Valve Engine.

The top of the cylinder is provided with a cam-controlled disc, provided with a central passage adapted to communicate by means of a port A with the inlet pipe B when the piston is raised. It is also provided with a number of other passages C communicating with the space above the piston to which leads the exhaust pipe



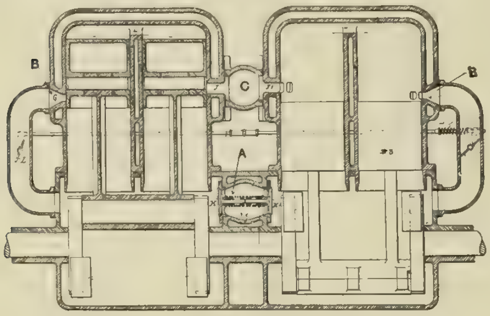
D. The valve is shown in the exhaust position, and it will be understood that it rises till it touches the top of the cylinder during the firing and compression strokes, and drops slightly to bring the ports A and B into line on the suction stroke. The top of the disc is provided with a raised circular rib, so that the pressures of explosion have effect on both sides of the disc in order to more or less balance it, and enable it to be moved in spite of high pressure existing in the cylinder.

J. H. Richards and B. Bellingham. No. 28,767/09.

A Two-Stroke Engine.

This invention relates to two-stroke engines, somewhat of the same type as the Valveless, in which the gas is compressed in the crank chamber, it being drawn into the two-crank chamber ports from a common induction pipe A through non-return valves. The gas is passed through transfer passages and inlet ports B into the

cylinders, and exhaust takes place from the opposite cylinder of each pair into a common exhaust pipe C. By using a single inlet and exhaust pipe a simple construction of engine, having pairs of twin units of this type, is provided. The transfer passages are controlled by

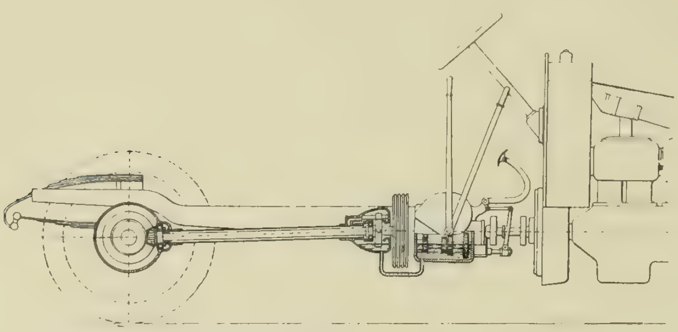


throttle valves, that on the right being adapted to cut off earlier than the left-hand one, a spring arranged in the throttle control permitting the valves to close separately. The engine is preferably arranged with a balancing piston working in a plane at right angles to the ordinary piston.

Albion Motor Car Co., Ltd., and T. B. Murray. No. 15,026/09.

A Lubricating System.

Any invention for reducing the number of lubricating devices which require attention must be welcomed, and for this reason the present invention appears to be a step in the right direction, inasmuch as it enables the lubricant for the gear box to supply the universal joint and back axle. As will be seen, the gear box is connected with the stationary universal joint casing, so that there is a common supply for the two, and the oil splashed up in the universal joint casing finds its

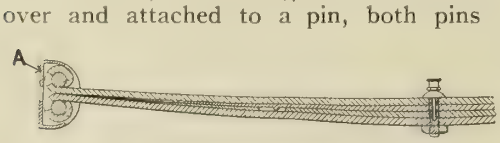


way along the propeller shaft tube to the back axle. This reminds one somewhat of the lubrication in the original 8 h.p. Rover, in which the oil supply to the engine fed the gear box, universal joint, and back axle.

L. Renault. No. 7,856/10.

Motor Car Springs.

The top leaf, as usual, has its ends turned over and mounted upon a pin. The successive leaves are shorter, as usual, but a bottom leaf corresponding to the top one is added, this having its ends turned over and attached to a pin, both pins

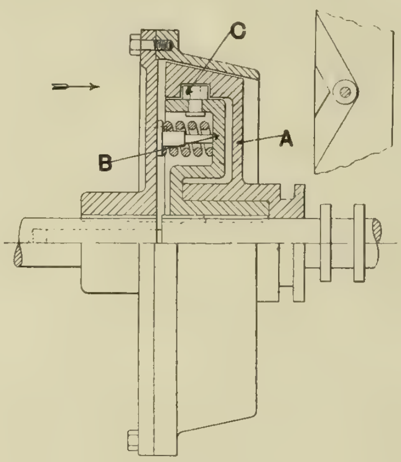


being mounted in a box A. This box is provided with means for lubricating the pins, and the whole spring is connected together at the centre by a transverse bolt, which is hollow, and provided with a lubricator and ducts to lead the lubricant to the faces of the spring leaves. These are provided with suitably-shaped oil ways. This is one of the few instances in which the question of admitting lubricant to the spring faces has been taken in hand. Theoretically the spring faces should not require lubricating, as one of the features of the leaf spring is its automatic "shock-absorbing" or damping effect due to the friction between the spring leaves. In practice relative movement between the spring leaves often is accompanied by a squeaking noise, which is, in many cases, as objectionable as spring oscillation.

W. G. Nelson. No. 20,524/09.

A Friction Clutch.

This friction clutch is of the type in which the engaging pressure varies with

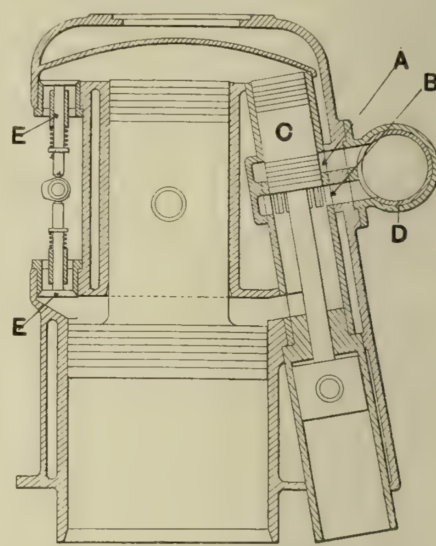


and is dependent upon the load transmitted through the clutch. The female cone is fixed to the driving shaft, and the male cone A is free to slide on the hub of an intermediate disc B, which is pressed to the right by the spring illustrated. The intermediate disc carries a roller C, having slight freedom in inclined slots or grooves formed in the male cone. The middle disc B is a fixture on the driven shaft. To engage the clutch the male cone A is moved to the right into contact with the female cone. As soon as contact takes place the male cone is rotated in relation to the stationary disc B, and this rotation, owing to the engagement of the rollers C with the inclined slots, causes the male cone to be moved endwise, so that the two conical clutch surfaces are forcibly engaged. To disengage the clutch the cone is moved to the left, the intermediate disc B yielding. The pressure required is light, which renders a construction of this type interesting to automobile engineers.

G. Savors. No. 15,101/09.

A Tandem-Cylinder Engine.

This is the invention of the same parties as the radiator already described, and, as will be seen, it refers to an engine with tandem cylinders controlled by piston valves. The exhaust valve is not illustrated in the drawings. The inlet ports A and B for the two-cylinder elements are controlled by a single valve piston C. These ports communicate with the induction pipe, in which is a rotating sleeve valve D, by which either port A or B can be cut off or throttled. The engine operates on the ordinary principles. In the drawing inlet is taking place in the lower cylinder, and on the inlet stroke of the upper cylinder it will be understood that the valve C has to move down to the bottom of its stroke uncovering the port A, but closing the port B. The invention relates chiefly to means for using either

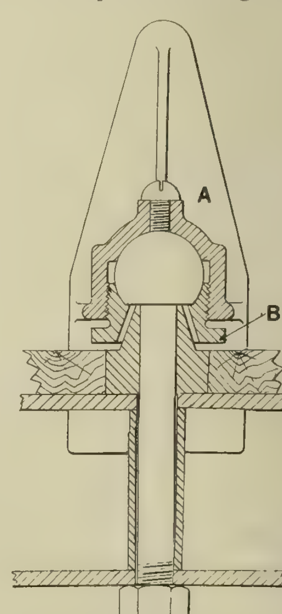


cylinder element alone or the two together. For this purpose each of the cylinders is provided with an air valve E controlled by cams connected up to the rotating throttle D. With the parts in the position illustrated both cylinders are operative, but to cut out either the throttle valve is moved to close the desired port A or B, and this movement at the same time opens the corresponding air valve E, enabling the cylinder cut out to offer as small a resistance as possible.

No. 13,667/09.

A Radiator Attachment.

To each side of the radiator is attached a plate A, which is formed, by casting or otherwise, integrally with a cup-shaped housing. This receives the spherical head of a bolt, which passes through the frame



members, and is gripped by a screwed collar B. The housing is provided with a plug at the top for the admission of lubricant, so as to provide ample freedom and permit the radiator to take up any desired position.

W. E. Lake (Brevetti Enrico). No. 1,442/10.

THE

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CONTENTS.

THE IMPORTANCE OF QUALITY.

It is not too much to say that, as a manufacturing community, the British have always been the leaders of the world with respect to the quality of their productions. Again and again in recent history one may find foreign inventions which British makers have improved and perfected to such an extent that the British copy became eventually better than the original, and even better than the foreign descendant of the original. Solid and durable workmanship, the highest class of raw materials, and elaborate finish are regarded as symbolical of British work all the world over, and the position which the British nation occupies to-day is, in a large measure, traceable to nothing more nor less than thoroughness.

Many reproaches have been hurled at this country on account of the fact that foreign automobile makers were for a long time years ahead of home manufacturers, yet those who had the lead have lost it now, and it is hardly to be believed that the British industry would have been in a better position to-day if it had commenced at an earlier date. The beginning of all things is experiment, and it does not matter much who makes the experiment if we read its lessons aright; so we gained as much from the early work of the old French firms as they gained themselves, and no one could deny that the average excellence of British cars to-day is as great as, or even greater than, that of any foreign ones.

The outstanding disadvantage of British goods has, however, always been their prime cost, but this is no disadvantage if the future is taken into account and durability allowed its true value; also it is a fact that when a British manufacturer commences to try to cut quality, in order to obtain a market for his wares amongst those who will buy things for their cheapness without other considerations, then he almost always begins to lose money. Just as we as a nation can do better mechanical work than the rest of the population of the globe, so do we fail if we try to do "good enough" work. It is in cheap stuff that the severity of foreign competition is felt most keenly.

There is reason to fear that many of our automobile makers are concentrating their energies upon cheapening their product rather than upon improving it; if they see faults in a design they will sometimes delay to remedy them till the action of some competitor compels them to do so. Firms who are acting thus are near a pitfall into which many manufacturers of all classes of goods have made haste to leap, and from which there is no exit, with the solitary exception of the Court of Bankruptcy. This trap is cunningly hidden by a mathematical fallacy, for it may be argued that if a hundred things cost so much to make and can be sold at so much more, yielding so much profit, then if the output is doubled the cost of production will be less and the price may be lowered without loss of profit. Within reasonable limits this is strictly true, but the trouble is that so many forget that there is any limit at all.

Taking the automobile industry alone, for the sake or argument, there are in this country a certain definite number of people who have incomes sufficient to enable them to afford to run a car. It is not only a matter of first cost, for if cars could be sold for five pounds there are still only quite a small percentage of the population who could afford to keep them, and even if overseas markets are taken into account, there is a limit to the total number of cars likely to be demanded of manufacturers in any one year. It is easy to see that if half-a-dozen firms base their calculations of car prices upon the expectation of selling a thousand cars, and if there are only eight hundred buyers, then each maker stands to lose money, and this is what is likely to happen in Great Britain—what must happen if price reduction is carried much further.

So far we are thankful to be able to say that costs of production have been kept down by improvement of manufacturing methods, and by the stoppage of improvements in design, rather than by any retrograde step, but as soon as the possibilities of such methods are used up there is no doubt that some makers will begin to cut the quality, and the effect upon their own reputation and the reputation of the British industry as a whole requires no elaboration. Once quality is cut reputation is gone for ever, for to regain it is impossible.

Of course, one of the many ways in which quality may be reduced is in raw material, and it is probably the most dangerous and, at the same time, the most difficult to guard against. The buyer for a manufacturing firm is usually a special member of the staff, who has nothing else to do, but in small shops material is often selected by the works manager, who is probably better qualified, but whatever the exact status of the man who has the actual giving of orders, his duty is to purchase the best possible stuff for the lowest possible price, and the temptation to allow the latter consideration to weigh the more heavily of the two is very great. It is so easy to make a tiny sacrifice of quality if it involves a fairly large difference of cost, and it may even be advisable to do so in many cases, but there is a danger of repeating the process every time the occasion arises for re-ordering fresh supplies of the same material. The first sacrifice may count for very little, and may result in the production of exactly the best balance between quality and

cost, but the second and third sacrifices may very easily end in a most serious decrease in the quality, even to the extent of making the material absolutely dear.

It is easy to argue that if material and fittings are inspected with proper care by the works staff, there can be no serious falling off without equally serious complaint, but the complaint is liable to be unheard until after the mischief is done. It is bad policy to demand such terms of a supplier that he will be tempted to break faith, and it should be remembered that the cost of testing every batch of raw material that comes in may amount to a most serious total by the end of the year. Therefore it is far better to pay a fair price for a trustworthily good article, and to apply occasional tests, than to pay an unfair price and try to force a hard bargain by perpetual exhaustive checking. This applies to tools and all contract supplies, as well as to raw materials.

There is little doubt that the tendency towards cheap work is not an inherent trait of the engineer, but is merely a mistaken commercial notion which is being forced upon engineers against their will, and so we feel confident that we shall have the support of the bulk of our readers in drawing attention to the underlying facts of the matter. It should always be borne in mind that there is no lasting honour in cheap work, unless it is also good work, and the honour will then arise from the goodness rather than the cheapness. It would be easy to mention a dozen cars that are lacking a few refinements which make just the difference between mediocrity and excellence spoiled for the proverbial ha'porth of tar—and we would urge designers to do the utmost in their power to make those who are responsible realise to the full the damaging nature of their policy, and especially its inevitable effect upon the future of the business.

The position is complicated very considerably by the need for cheap, small cars, since there is a fairly large class of user who can afford the upkeep of quite a small vehicle and whose ability to lay out capital for the initial purchase is very limited. The trouble, of course, arises when it becomes necessary to decide the position of the dividing line. Still, it is better to give "quality value for money," than to attempt to give "quantity value," and it has already been proved that there is a market for very small cars which are made as well as possible, and command a correspondingly fair price. If it is desired to make a car to sell at a fixed price, the size should be chosen so that the best of everything can be given at that price, because small size with satisfactory performance and durability is much more valuable than large size coupled with inefficiency.

If further proof be needed that our contention is the correct one, there is a convincing commercial argument to be found by an intelligent study of the balance sheets of the dozen largest British motor manufacturing concerns for the last five years. It will be seen that those who have made a steady profit are usually those who have made the best articles, and those who have not kept quality as the prime consideration have, in some instances made very large profits for single years, but on the whole showing have not done so well. Here again the situation is greatly complicated by side issues, but it is not really possible to construe the facts otherwise.

Another complication arises from the fact that quality can be worshipped too devoutly. It is possible to carry the desire for excellence above all else to an extent bordering upon the absurd. To give an instance, it would be excellent if all parts of a machine which are liable to rust were heavily electro-gilded or platinised, but it would be absurd to increase the cost so much for so little real benefit. Likewise the habit of making parts from the solid bar has often been allowed to become ridiculous by increasing the cost of making parts tenfold or even more, with only a fractional increase in practical value. This again brings up some knotty points, as, for instance, the need for thrust bearings in road wheel hubs, for while they are probably essential, it may be argued that they are not worth their cost even to the user, as the durability of simple ball journal bearings is sufficiently good, and renewals are not needed sufficiently less often when the thrust bearing is used to recompense for the extra cost. It would be possible to go on multiplying examples at very great length indeed, but to do so would serve no useful purpose, and many of them will doubtless suggest themselves to our readers.

The immediate aim of all industry is towards the amassing of money, and we would not thus urge the importance of quality on ethical grounds alone. The prosperity, as well as the reputation, of the British automobile industry is at stake, and by maintenance of quality alone is the leading position in the automobile world to be secured to Britain.

ON SOME ETHICS OF DESIGN.

THE sternly utilitarian training of the engineer usually leads him to regard mere appearance with something of contempt, but when he comes to design for automobile work and finds himself face to face with the requirements of the individual public, guided as often as not by the foibles or fashion of the moment, it is not to be wondered at that in some cases reaction has led him rather to the opposite extreme, with the result that now and again one meets cars in which much has been sacrificed to appearance.

It is the really practical man who realises that appearance is one of the first essentials to commercial success, for he knows that, from that point of view, the buying public constitute his ultimate fact, and as a general thing they set more value on the appearance of a car than on many other much more essential qualities.

But what constitutes the highly elusive and indefinable "style" so diligently sought, so frequently just missed? What makes one car appear beautiful and another ugly?

The writer would suggest that the answer is to be found in the axiom that utility is the basis of beauty. This statement, trite enough in itself, but insufficiently recognised outside of artistic or philosophic circles, will bear the crucial test of application. If every individual part of a car be so designed as to be the most suitable possible for its purpose, and if those parts are combined in the most effective form, the result is bound to be intrinsically beautiful, no matter how far it be removed from the conventional, but, if the machine exhibits unsuitability for the purpose for which it is designed, no argument can transform, no fashion render it permanently pleasing to the eye.

If, therefore, the designer takes as a guiding principle never to insert a single feature that cannot be justified up to the hilt by sound reasoning, he cannot go far amiss, and he will find it a rule that can be applied to forecast permanent public taste with considerable accuracy. However far removed the idea may be from convention, if it can justify its existence, it will continue despite the natural aversion of most of the public to anything strange, and must in the long run inevitably be regarded as good—and, if it affect the appearance of the car at all, to that extent beautiful—for the human mind is very peculiar in this way, and what at its first introduction may appear ugly may, and often does, by constantly coming before the mind, by its own merits come to be looked on as beautiful.

Take a case or two in point. The writer can well remember the time when bogies were first extensively introduced on locomotives in this country, and the public, used to express engines with a single pair of small leading wheels, for a time remarked on the unpleasing appearance afforded by the then unusual design. Now an express engine, if anything, looks strange without a bogie, and certainly the general opinion from an æsthetic point of view is in favour of the bogie. In fact, the public are unconsciously aware that the bogie is the most serviceable construction for its purpose, and therefore it has become to them subconsciously pleasing.

Let us take another case nearer home—the flush sided body. At its first introduction few had a good word to say for its appearance, which went against preconceived ideals, and even the writer must confess that he found it hard to appreciate its beauty, although quite conscious at the time that ideas of convention were leading his reason astray. By now, however, we have had time to re-adjust our outlook, and the car with the flush sided body looks pre-eminently the "thing beautiful," and this simply because there were sound reasons and justification behind every strange feature.

It may be argued that it is not a wise policy to introduce such radical changes in the appearance of a car or machine—that it is best to leave others to experiment and create the demand for the novelty, and then step in and reap some of the result. If such an argument, however, would stand the light of day there would be an end to all enterprise that could not with fair certainty be protected by law. On the whole, motor users constitute a progressive section of the public, and if a thing be really good there is quite sufficient demand to take up the initial supply from those who like to have the latest thing. Broadly speaking, it will be found that firms that have introduced really good novel features have not had occasion to regret their action, and, although there are exceptions to every rule, we would suggest that in such cases the failure has been caused by lack of management in handling the novelty rather than in the failure of that novelty, and we commend the rule to our readers as a sound and reliable gauge for the estimation of public favour.

THE DESIGN AND CONSTRUCTION OF CAB CHASSIS.

By Hubert C. Clark, A.M.I.Mech.E.

IN commencing the design of a chassis for cab and general hiring purposes, a number of considerations have to be taken into account which naturally do not apply in the case of a vehicle intended solely for private pleasure purposes.

Whereas, in the case of a private vehicle sacrifices in the matter of expense may be made to secure luxury, comfort, or perhaps speed, no effort must be spared in the case of a cab chassis to ensure that the running costs shall be as low as possible.

This is of prime importance, as, obviously, the sole reason for the existence of this class of vehicle is to earn a sufficient profit on its running, and if it cannot do this it fails entirely in its object. Coupled with this is the need for low prime cost, and, while everything possible must be done to keep this at a minimum, it is a very mistaken policy to attempt to achieve this end by means of indifferent or poor workmanship and material.

By careful design the number of parts must be reduced to a minimum, and they must be such that they can be finished, as far as possible throughout on the machines, and assembled with the least possible amount of hand work. Some thought also, should be given by the designer to the class of machine tools to be used, and, wherever possible without impairing the efficiency of the parts in question, the design should be such as to permit the use of those tools which can be installed and operated at the lowest cost. Standardization should be aimed at, and similar parts used in as many places as possible, in order to cheapen the production by requiring larger quantities of the same article, and reducing the amount of stock necessary for repair purposes.

Needless to say the first cost is dependent mainly upon an efficient factory organization; firstly, by the use of only the most suitable tools, and by working these to their maximum output, and secondly, by the minimization of overhead charges.

A rigorous inspection of raw material and finished components is, in the writer's opinion, essential, as by this means only can good quality and accurate workmanship be ensured. By the rejection of all parts not within the prescribed limits of accuracy the work of both assembling and repairing is facilitated, and the cost of such inspection and rejection saved probably many times over.

By such methods only can low cost of production be attained, while at the same time they ensure a high quality of product automatically.

In the matter of running expenses, tyres are probably the most costly item, and while these cannot be considered as a part of the mechanical portion of the car, their wear and tear is a point which must be considered seriously by the designer. The points which mainly affect the life of the tyres are:—(1) Weight of car; (2) Regularity of turning effort; (3) Clutch; (4) Brakes

With regard to weight, this should, of course, be kept as low as possible consistent with a proper degree of strength and stability. In the case of a four-cylinder chassis, such as is now most commonly used for cab purposes, the weight, with wheels and tyres, should not exceed about 1,600 lbs. As far as the writer's experience goes the weight of the average four-cylinder cab complete is 22 to 23 cwts., or somewhere in the neighbourhood of 2,500 lbs., which means that the body and accessories account for nearly 900 lbs. Thus there appears to be ample opening for the coach builder to contribute his share towards the elimination of unnecessary weight.

In designing a chassis for cab purposes, one of the first considerations must be the Police regulations. These for the most part are not difficult to comply with, but two of them tend to hamper the designer to some extent. These are the rules which ordain that the vehicle must turn completely in a 25ft. road, and that the clearance between the lowest point and the road surface must not be less than ten inches.

The first of these restricts the wheelbase definitely, and, while from the traffic point of view the regulation is undoubtedly an excellent one as it facilitates manœuvring to a marked degree, it presents several disadvantages from the designer's standpoint.

In the first place excessive lock of the front wheels must be provided, which necessitates the provision of a very strong front axle, and one capable of resisting considerable torque, as if power is applied when the wheels are on the extreme lock the bulk of the effort tends to push them in a direction almost parallel with their axes, and so twist the axle.

The reduction of the wheelbase to some eight feet, which is the maximum that will give the required turning circle, crowds the driver very much forward, and necessitates the use of a very short side steering link, which naturally has a big angular movement in the horizontal plane. This gives rise to a big side thrust on the lever which depends from the steering gear box —a not very desirable feature.

Also the chassis rides much less easily than would be the case could the wheelbase be made, say 9 feet or so, while the diameter of the road wheels is more or less restricted, or otherwise the room necessary for a clear entrance to the body is encroached upon, so that the required road clearance cannot be obtained by the use of large diameter wheels.

The question of road clearance needs careful attention, mainly in connection with the front portion of the chassis. For some reason the Scotland Yard authorities are not quite so particular with regard to the rear axle clearance, as to the clearance under the front axle, engine and steering connections. Apart, however, from this point, it is advisable to allow a good margin over or under, as the case may be, the dimensions set forth in the regulations, as the examination is very strict.

As the above-mentioned regulations are largely concerned with the frame, it will perhaps be as well to consider this part of the chassis first. Here we experience one of the difficulties introduced by the need for an extreme lock of the front wheels, namely, the narrowing of the front portion of the frame. This, in the ordinary type of pressed frame, involves a very considerable offset or joggle. Now the presence of this offset produces in the side member, when loaded, a torsional stress which the frame is most ill-calculated to resist, and as this stress is naturally proportional to the offset, its effects are most marked in a cab frame, where this is large.

To avoid the excessive sagging, which is apt to take place, some means of stiffening the frame at this point must be adopted.

Two methods may be suggested. The first of these is somewhat on the lines of the Lanchester frame (see Fig. I.), and

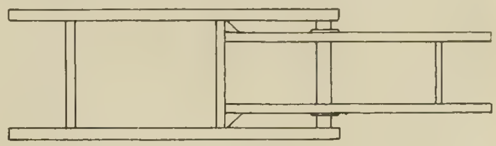


Fig. I.

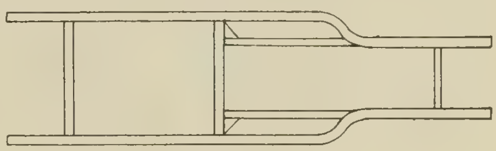


Fig. II.

the second a modification of this, which makes use of the ordinary joggled side member (see Fig. II.). In both cases the rigidity of the cross members is relied upon to eliminate the torsional stress in the side member, a much more effective method than deepening the flanges of the side member and relying on that alone.

While on the subject of frames a word may be said for the tubular frame. This type has practically dropped out of existence, mainly because of the difficulty of making satisfactory joints, and of attaching the various brackets in a simple manner and at a reasonable cost—a difficulty which in these days of electric and acetylene welding might now be overcome. As a set-off to any extra cost of manufacture, it must be remembered that a tubular section is very rigid and affords the most effective distribution of metal for resisting torsional and compressive stresses, while in tension this section is, of course, as good as any other. It is not the best section to resist a bending stress in any one direction, but it should not be impossible to design a tubular frame in which the side members were satisfactorily stiffened without the use of an undue amount of material, such as by the use of double tubes, or of truss rods.

Considering, then, how impossible it is to determine the direction and intensity of most of the stresses in a frame, the tubular member seems to offer advantages not possessed by the more usual channel and H sections, which while excellent for resisting a bending stress in a given plane, are much less effective under compression than a tube, and are notoriously weak under torsion.

The general shape and construction of the frame depends to a large extent on the type of suspension adopted. There are

in use at the present time a large number of fancy springing arrangements, such as transverse rear springs, and three-quarter elliptical springs; the effect of these, however, is merely to increase the total flexibility, a result which can be as readily and more cheaply attained by using longer and more flexible springs of the usual half-elliptical type.

There is a limit, however, to the range of movement which can be allowed on the axle, controlled by the distance permissible between the axle and the frame, beyond which it is not possible to go. In the Lanchester type of overhung suspension, which is probably the most flexible now in use, this difficulty is overcome by throwing the rear axle out beyond the frame, thus permitting of a very much increased range of movement. It is doubtful, however, whether an overhung suspension could be applied satisfactorily to a cab chassis, owing to the increased length of wheelbase involved.

Some increase of comfort would probably be derived by insulating the body from the frame, by means of further springs, somewhat after the manner adopted on some types of tramcar, and it does not seem that the extra cost of doing this need be very great; but a specially-designed frame would be necessary, and it is questionable whether it would be worth while to make this addition for cab purposes.

Special care should be taken in selecting the material for the springs, as much of the steel used for making springs for many other purposes is quite unsuitable for motor vehicles, where the stresses are apt to be considerable. The top plate should be of the same steel as the remaining plates, and should have the eyes rolled, and bushed with phosphor bronze.

Attention should also be given to the centre bolt, which should not be a common steel black bolt, but a turned bolt of a good quality nickel steel, and this is especially the case where the drive is taken through the springs.

The question of wheelbase is purely a matter of comfort and size of turning circle, which points have already been dealt with. The width of track is, however, a point worthy of careful consideration, and in the writer's opinion 4ft. 8ins. is a preferable dimension to the 4ft. 6ins. and 4ft. 4ins. frequently adopted on these small chassis. It allows, in the case of the front wheels, full advantage to be taken of the maximum lock, while, in the case of the rear wheels, plenty of room is available for the bodywork.

The steering gear is, of course, a matter of importance, and while no striking departure from ordinary practice is suggested, it is very necessary that the whole of it should be of a substantial nature. The wearing surfaces should be as large as is possible, and all the joints of the links should be of casehardened steel. The joints should be preferably of the ball and socket type throughout, as simple means for effecting adjustment for wear are more easily provided with this type than with pin joints, while the system is rendered more flexible. The effective lubrication of these parts is a very difficult matter, and the usual leather stocking, packed with grease, leaves much to be desired, as the messy and somewhat tedious job of cleaning it and replacing the lubricant is very apt to be neglected. Up to the present however, it seems to be the only method available at a reasonable cost.

Of the mechanical portion of the vehicle the item which has the greatest bearing on the running cost is the engine.

Too large an engine involves an undue consumption of fuel, while too little power means more time occupied on journeys, and consequently reduced earning capacity. On the whole an engine of from 12 h.p. to 14 h.p., by R.A.C. rating, would seem to be most suited to present conditions.

The question as to number of cylinders is one that, in the writers opinion, can be answered entirely in favour of the four-cylinder engine, at any rate as far as the two and four-cylinder varieties are concerned. Two cylinders give a shorter engine, but the balance is poor, and the individual impulses greater; the torque is uneven, and a heavy flywheel is necessary to obtain sufficiently smooth running, and this naturally impairs the accelerating power of the engine, a very important point in cab work.

The construction is cheaper than that of a four-cylinder, but a single-cylinder engine would be cheaper still, and there seems no reason to believe that an engine of this type could not be designed to give practically as good results as a two-cylinder. On the whole it may be taken that the higher first cost of a four-cylinder engine is compensated for by the decreased wear and tear of all moving parts, due to the more even torque, and by the improved power of acceleration. Also, with the four-cylinder engine fewer changes of gear are required.

In the matter of cylinder castings there is a considerable divergence of opinion as to the relative merits of the separate and "monobloc" castings.

With regard to first cost, the theoretical advantages lie entirely with the "monobloc" system. With this there are fewer connections to be made, and less piping to be fitted up, and the result is a cleaner-looking engine. Against this, however, is the greater relative cost of the castings, and the higher proportion of scrap, while, owing to the costliness of scrapping such pieces the temptation to make use of castings which have been incorrectly machined in some part is great, thus tending to impair the strict interchangeability of this and adjacent parts.

Single-cylinder castings, on the other hand, mean a longer engine and many joints, so that the advantage probably lies with the double cylinder, which, to a large extent, combines the good points of the two other types, i.e., it is a fairly simple casting and easy to handle; it builds a reasonably short engine, and the joints and pipe connections are few and simple.

Perhaps the worst point about the "monobloc" construction is that to take full advantage of its capacity for shortening the engine involves the use of only two bearings to a four-throw crankshaft, a form of construction difficult to justify on any grounds in a high-speed machine like a petrol motor.

In connection with the statement that 12 h.p. to 14 h.p. by R.A.C. rating is the most usual size for a cab engine, it must not be forgotten that as this formula takes no account of stroke or piston velocity, the figure is a comparative one only, and gives no real indication of the actual horse-power, which may be as much as 22 to 25.

The most rational, and at the same time most accurate, method of estimating the horse-power of an engine is by its fuel consumption, or, in other words, by its capacity, and a useful ratio of brake-horse-power to total capacity (i.e., volume swept by pistons per stroke) is .17. This assumes a speed of about 1,800 r.p.m., so that at other rates of revolution the horse-power will be in proportion.

The $\frac{v}{s}$ ratio to which this figure is suited is .222, and this may be considered a satisfactory proportion for a small engine.

The question of length of stroke is more or less a matter of convenience. Given a required horse-power, a certain capacity is necessary, and to obtain this with a very short stroke involves a correspondingly large cylinder diameter, and possibly a longer engine. On the other hand, a long stroke leads to considerable width of crank chamber, not always convenient to bestow within the limits of a narrow frame.

It is common practice in gas-engine design to make the cylinder proportions such that the ratio of $\frac{\text{surface}}{\text{volume}}$ is at a minimum at about one-third of the stroke, and this proportion will apply very well to petrol engines, the stroke being from about 1.5 to 1.7 diameters, varying, of course, with the valve arrangement and design of combustion chamber.

In dealing with the crankshaft the two main points which have to be aimed at are large bearing surfaces and ample rigidity. The first of these is best met by providing journals and crank pins of sufficient length, but is assisted also by the requirements of the second condition, namely, large diameter. To ensure good running and wearing qualities, the pressure on big end bearings should not exceed 300 lbs. per square inch, and that on the main bearings 200 lbs., based on the mean effective pressure.

As regards rigidity, it is difficult to lay down any hard and fast rule for determining the diameter, but the following formula, while purely empirical, will be found to give satisfactory results:—

Let D = diameter of cylinder in inches,
d = diameter of crankshaft in inches,
and K = a constant, which may be suitably taken as .45. Then:

$$D^2 \times K = d^3$$

$$\text{from which } d = \sqrt[3]{D^2 \times K}$$

Compared with the requirements of the maximum torsional stress in the shaft, the diameters given by this formula may seem large, but they are not too much so to supply a reasonable amount of rigidity. No allowance, of course, is made for the difference in stress due to any considerable variation of the stroke, but the formula will apply quite well over the limits of stroke mentioned above.

If thought necessary for lightness, the centre of the shaft may be bored out to a diameter not exceeding .4 of the diameter of the shaft without any appreciable reduction of strength.

As to the disposition of the valves, these are best placed all on one side of the engine. This gives a more satisfactory combustion chamber than "either-

side" valves, while, as there is only a single camshaft, the construction is more simple and less costly. The valves, caps, springs, tappets, etc., should be all alike and interchangeable. The area of the valves should be as large as possible, and the lift only sufficient to give an area equal to the port area, or even rather less. The gas velocity in the inlet pipes should not exceed about 200 feet per second at maximum engine speed, but some restriction at the valve will not be of serious importance, provided that the pipes and port apertures are of sufficient area. A good proportion for the valve diameter is .5 of the cylinder diameter, and they should not be made very much less than this, or the lift will become excessive. Thus with a 3-inch diameter cylinder, valves of $1\frac{1}{2}$ inches outside diameter with a lift of 3-16 inches will give a gas velocity of about 220 feet per second, which should be found quite satisfactory.

Slide valve engines may, at present, be considered as out of the question for cab work, as, beyond the slightly greater silence claimed for them, they offer no advantages that cannot be obtained equally well with the mushroom valve, at less cost and with greater simplicity.

As this article is not intended for a treatise on engine design, the question of the carburettor is too large to be gone into in detail. It is sufficient to say that simplicity in this portion of the chassis is perhaps of more importance than in any other, and the fewer the adjustments necessary or possible, after the vehicle has left the factory, the better. With an engine of the size which has been under discussion the average consumption should not exceed from .035 to .038 gallons per ton mile.

With regard to ignition also there is little to be said. The high-tension magneto is known and used so universally that any comments on it in this connection would be superfluous. Accumulator ignition is, of course, out of the question for cab work. The point of ignition should be fixed, and while a ready means of adjusting this is very desirable it should not be accessible to the driver.

The accessibility of the magneto and facilities for its rapid replacement if necessary are points which require strict attention. Needless to say, the drive should be effected by enclosed gearing.

The engine control should be of the simplest possible character, and should consist preferably of a foot-operated throttle valve, with a conveniently situated hand adjustment. This adjustment need only operate the throttle from the closed position to about half full opening, and is intended solely for determining the "light" running speed of the engine, and for holding the throttle open when starting.

The question of engine lubrication is a much-discussed one, but, to the writer's mind, there is no question as to the superiority of forced lubrication. This, combined with the use of a good-quality white metal bearing, will ensure thousands of miles of running with hardly any appreciable wear of the journals and crank pins. A lot is heard of the danger of choked pipes with this system, but, in the writer's experience with some hundreds of engines thus fitted, this trouble is almost unknown and not serious, though care should, of course, be taken to see that all pipes and ducts are as large as possible, and the pump should have a considerable surplus capacity to allow for wear of the various bearings. A bye-pass valve should be provided to take care of this surplus delivery, and this should be a proper valve of large area—not a 3-16th inch ball with a piano wire spring behind it. A pressure gauge is not a necessity, but an indicator of some sort, which can be fitted on the dashboard and will show clearly whether the oil is flowing or not, is indispensable.

As an alternative to forced lubrication, a drip feed to the main bearings with banjos to collect the oil from them and deliver to the big ends by centrifugal force is much to be preferred to the splash system, for it gives, at any rate, a positive feed to the crank pin surface which is not attained by any splash methods.

In the matter of cooling the choice lies between two systems, viz., forced and thermo-syphon circulation. By adopting the latter method the cost of the pump and its driving gear is avoided, as is also the cost of repairs to this part. These latter are not a great matter with a well-designed centrifugal pump, which need require but very little attention, but there must of necessity be some appreciable wear in course of time.

As the thermo-syphon system can be made to give excellent results, there seems little reason for including the extra moving parts incidental to the use of a pump, save that it is necessary with a syphon system to keep the radiator full of water, as if the level drops to too great an extent the circulation will cease, and this point is perhaps of more importance than it seems at first sight, as the average cab driver is apt to be rather careless in the matter of water.

Honeycomb radiators being expensive to make, and more so to repair, are not well suited to cab work. Of the tubular type plain tubes are to be preferred to gilled tubes, the tube surface being from two and a half to three times as effective as gill surface. An approximate rule for proportioning radiators is to allow one square foot of gilled tube surface or about .7 square feet of plain tube surface per brake-horse-power. This should prove sufficient for thermo-syphon cooling on cabs.

In selecting the type of clutch most suitable for cab work, consideration must be given to the fact that in this class of vehicle a good deal of work falls on the clutch. Probably the cheapest form is the leather-faced cone, but it is by far the most sensitive to rough usage, and, except on the score of cost, cannot compare with a well-designed metal-to-metal clutch.

A metal-to-metal cone clutch has been found, in the writer's experience, quite satisfactory, and can be made quite small and light. If one of the members be of cast iron and the other of case-hardened steel, the included angle may be as small as 16 degrees, which will permit of a comparatively small diameter, without the use of too heavy a spring. The only disadvantage connected with this form of clutch is that it calls for very accurate workmanship, as the slightest lack of lineability or concentricity will render it practically useless.

On the whole the plate clutch, consisting of one or more plates, is likely to prove the best all-round for cab work, as, properly designed, it will take hold rapidly without being fierce, even with more or less clumsy usage. The multi-plate type also being of small diameter shows less tendency to spin when disengaged, and thus facilitates gear changing.

In multi-plate clutches the plates should not be too thin, say not less than No. 16 S.W.G., otherwise they tend to buckle, and they should be ground on both sides to render them quite flat and to remove all scale.

The pressure between the plates should not exceed about 20 lbs. per square inch, while the coefficient of friction may be taken at the maximum as .06. With the above pressure a light spindle oil will usually be found the best lubricant to use in such a clutch. In designing the clutch every effort should be made to utilise as light a spring as possible, as it must be remembered that the driver will be using it for many hours at a stretch, and a heavy pedal pressure becomes rapidly fatiguing.

In considering the gear box there is little to be said that does not apply as well to pleasure cars as to cabs. The shafts should be of large diameter, and thoroughly well supported, as their stiffness is of the greatest importance in minimising noise and wear of the wheels. The wheels should have faces as broad as possible, and the pitch should not be too coarse. A coarse pitch wheel does not run so quietly as a finer one, other things being equal, and, if under-cutting of the teeth is to be avoided, involves the use of larger diameter wheels, the peripheral velocity of which is considerably higher, thus tending to make changing less easy and running more noisy.

The modern steels enable an ample tooth strength to be obtained with a finer pitch, and this is assisted by the use of broader wheels than are frequently employed.

In the matter of ball bearings the general excellence of the product of the leading firms leaves nothing to be said. The writer is, however, strongly of opinion that better results are obtainable from bearings containing a few large diameter balls than those containing a greater number of smaller ones.

A point which deserves more attention than it obtains is the leakage of oil from the bearings. This applies also to the engine to some extent, but usually the gear box bearings give most trouble from this cause. An ordinary stuffing box with hemp or asbestos packing is sometimes used, but these require very careful packing, and, on the whole, some form of metallic packing will be found more satisfactory. Owing to the fact that sufficient oil may be used at times to bring the level above the bottom of the shaft, the ordinary throw-off ring is not of much use, as the leakage may then take place more when standing than when running. A false head, too, is sometimes caused when running by the oil piling up between the last gear wheel and the end of the box.

There is some difference of opinion as to whether the "gate" or "straight-through" type of change gear is most satisfactory for cab work, but there should not be much difficulty in deciding

that the former has a big advantage over the latter, inasmuch as the gear shafts can be made shorter and be very much better supported.

It is largely due to the "straight-through" change that the very narrow wheels, which used to be so commonly met with, came into being, and it is difficult to see how the gear box can be kept of reasonable length with this system, if broad wheels are to be used.

It is generally considered that the gate system is the more expensive, but this need by no means be the case, while something must also be allowed for the saving effected by the reduction of the size of the gear box.

The whole of this mechanism ought to be of very substantial construction, as it is apt to be handled with more zeal than discretion, and its use is constant.

Gear changing is a very real trouble in cab work, and the result of some trials of epicyclic gears for the purpose would be of considerable interest. As gears of this type are giving every satisfaction on cars up to 30 h.p., there seems no reason to believe that they would not do equally well on a cab chassis. Certainly the passenger's comfort would be increased if the clashing of gears, now so commonly heard, could be abolished, while some good must necessarily result to the mechanism.

In designing the brakes for a cab it must be remembered that the wear on these is considerable, and as large a surface as possible should be provided on the blocks in order to reduce this wear to a minimum. This is best accomplished by making the drums of good width, say $2\frac{1}{2}$ to 3 inches, rather than by making them of larger diameter than is usual. If the latter method is adopted, for any reason, the leverage must be reduced, otherwise the tendency will be to make the brake fierce, and the wheels will lock too readily, with the accompanying loss of braking effect and ill usage of the tyres.

Of course, owing to the widely varying character of the road surface, the question of locking the wheels is one over which the designer has very little control, but it is only on dry surfaces that much harm is done to the tyres. Wet and greasy surfaces must needs be left to the skill of the driver.

In considering the degree of friction between the road surface and tyre a coefficient of .7 is sometimes recommended. This, however, is distinctly high for all-round conditions, and a better average is probably not more than .5, and, of course, under certain conditions it is far less than this.

With regard to the type of brake, band brakes are not very suitable for cab work. They require very nice adjustment, or otherwise they tend to slip or to become unduly fierce; also replacement of the wearing surface cannot be effected easily or quickly. The most satisfactory is what may be called the locomotive type, and Fig. III. shows a design of foot-brake, but this system may be readily adapted to internal application, which is desirable for the rear wheel brakes. In this brake the blocks are made of good cast iron throughout, and, being attached to the levers by means of the fulcrum pin only, can be replaced very easily and quickly, while their cost is small.

The brake drum should be of steel or first-rate malleable iron, but in either case should be of a fairly hard variety to ensure smooth action and to confine the

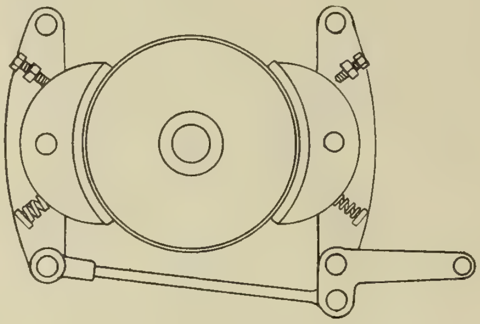


Fig. III.

wear, as much as possible, to the blocks. In fact, this part would be far better case-hardened.

To obtain sweet action of the brakes it is essential that they should be rigidly attached to their supports, and although this is a matter of no difficulty in the case of the rear brakes, which are attached to the axle casing, the foot brake is another matter. Usually this is slung from a comparatively light cross member, which is far from rigid, and some more substantial support should be devised. In light cars there is little or no objection to carrying the brake anchorage on the gear box, if this is suitably strengthened to take the consequent stresses, but the fulcrum pins should be supported on both sides, so as to avoid any bending stresses.

Of front wheel brakes, the writer's experience is small, but there is no doubt of their efficacy. It would seem preferable that they should be connected with the rear brakes, so that all four wheels might be braked at once. In this way the stopping power could be materially increased, while the work thrown on each individual tyre could be reduced by perhaps 30 per cent. or 40 per cent. Probably it would be found best to connect these brakes to the pedal for ordinary use, and retain the shaft brake as a hand brake for emergency purposes.

Like the gear box, the rear axle for a cab chassis does not call for very different treatment to that required for a pleasure car. In attaching the axle to the frame radius rods are perhaps better dispensed with, the tractive effort being taken through the springs, which should be kept comparatively flat. The ordinary triangular torque rod probably offers the most satisfactory method of controlling the torsional effort, but it should not be attached to the axle rigidly in the barbarous manner sometimes encountered, but it should be flexibly attached, so that it can vary its angular relation to the axle in the horizontal plane.

The flexibility possible with this system, combined with its simplicity, are the chief points in its favour, while the axle can generally be rendered more accessible by its use than when the central torque tube, enclosing the propeller shaft, is adopted.

Of the relative merits of the worm and bevel gear drives there is not much to be said. While the worm is far more silent there is not much to choose between them in the way of efficiency. The objection to the worm drive is that, where road clearance is of importance, the worm has to be placed on top, which increases the height of the frame, a point which, though of no real moment, is much objected to by many people. In designing this form of drive it is of the utmost importance to provide ample tooth surface and a heavy thrust bearing with very large balls, also both worm and wheel must be accurately hobbed, as upon the correctness of the tooth form depends the success of the gear.

Some useful data on worm design will be found in Halsey's "Worm and Spiral Gearing," and in a paper on "Worm Contact" read before the Institution of Mechanical Engineers by Mr. Robert Bruce. See Proc. I.M.E., Part 1, 1906, p. 57.

The question of lubrication throughout the chassis is one needing most careful attention. Oil should be used wherever possible in preference to grease, and as large a number of parts as is possible with certainty should be lubricated from one common source. Of course, the gear box and axle are a simple matter, it only being necessary to provide *accessible* apertures for filling and, preferably, some means of preventing an excess of oil from being introduced.

In the case of other moving parts there is room for the exercise of considerable ingenuity in devising means for lubricating them efficiently, and the repair bill will be affected materially by the degree of attention given to this matter.

There remains a point concerning which much discussion has taken place, and that is the advisability of constructing the engine and gear box in one unit. There is no doubt that this system possesses several marked advantages from both the user's and manufacturer's point of view. The chief objections urged against it are usually:—that the component parts of the unit are less accessible, while in place on the car, than when arranged separately, and:—that to deal with any one of them involves the removal of the whole unit instead of only the gear box, clutch, or what not. The first of these is true to only a small extent, while the removal of the unit should be as simple a matter, or sometimes even more simple, than that of the ordinary engine or gear box, as it is, or should be, only attached at three points.

From the manufacturer's point of view the advantages are:—

(1.) The whole of the engine, transmission gear, pedals, gear lever, etc., can be erected on the bench, the final placing in the frame being a very small matter.

(2.) The unit is perfectly rigid, and all parts are located from definitely machined faces, so that shafts, etc., all come perfectly into line, and require very little fitting.

(3.) There is no lining up to be done in the chassis frame; consequently no universal joints are required between clutch and gear box, and a troublesome piece of erecting is avoided.

To the user the following points should commend themselves:—

(1.) Owing to the easy detachability of the unit there is little temptation to keep a cab standing all day while some small repair is being executed. By having a spare power unit in reserve, the defective one can be replaced in a short time, and repaired as convenient, thus a judicious investment in spare units should reduce the amount of night labour required, and

keep the vehicles off the streets to the smallest possible extent.

(2.) Owing to the rigidity of the whole unit the relative positions of the various moving parts are strictly maintained, with consequent reduction of wear and of noise.

(3.) Owing to the conditions mentioned in (2) the interchangeability of the various parts is more readily achieved.

While this method of construction presents the advantages mentioned, it does not necessarily follow, of course, that it is suitable under all conditions. The great objection to it is the weight of the unit, all of which has to be handled for even a small repair, and this, while not perhaps of much moment in a large garage well equipped with lifting appliances, might prove troublesome in a small one.

In this, as in all engineering problems, the designer must be guided by circumstances, and effect the best possible compromise to meet the given conditions.

THE E.N.V. AERO ENGINE.

A short description of some of its most salient points.

By W. G. Aston.

THREE types of E.N.V. engines are made, viz., 60 h.p. eight-cylinder, 40 h.p. eight-cylinder, and 30 h.p. four-cylinder. The last is of the double opposed type, and follows somewhat the lines of the Dutheil Chalmers. With a bore of 90 mm., this engine gives its rated power at

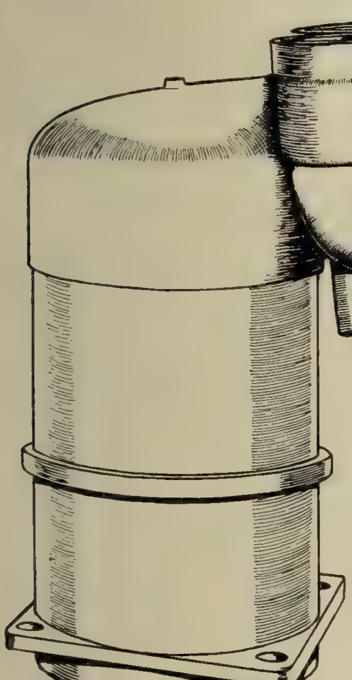


Fig. I.

1,200 revs. per min., but the most remarkable point about it is its extreme lightness, the total weight in full running order and including radiator, water in jacket, etc., being no more than 110 lbs. At present, however, it is proposed to deal only with the two larger types, the 60 h.p. and 40 h.p.

Except for dimensions, the same description applies equally to both productions, as the smaller is simply a reduced facsimile of the larger. The engine is of the V type, the eight pistons being connected to a four-throw crankshaft with cranks at 180°. To avoid the necessity of using bent connecting rods or offset small-ends, the cylinders are staggered so that the big ends lie symmetrically along the centre lines of their respective cylinders. This results in the creation of an unbalanced couple, but its effect is slight. The cylinders are furnished with flanges by which they are bolted to the aluminium crank chamber, the camshaft lying at the apex of the triangular upper part, and being, therefore, set symmetrically between the cylinders. It is driven by timing gear enclosed in a box forming one end of the crankcase, an extension of this gear serving to actuate the centrifugal water circulation pump and the high tension magneto.

The makers of the engine have exercised particular care in the selection of the various metals used in its construction, and this remark applies especially to the cylinder castings, which are of a singularly fine-grained brand of iron. As a copper water jacket is used, the cylinder casting is a fairly simple affair, and is machined both inside and out. The combustion head and valve ports are so designed as to offer the minimum of resistance to the passage of the gases, and care has been taken to secure an equal distribution of heat in order to avoid possible distortion of the valve seatings. Fig. I. represents the cylinder casting after machining, and shows how this part appears when ready for the copper-depositing process. It will be noticed that the cylinder is turned close up to the valve pocket, thus permitting a reduction of weight which would have been impossible with a cylinder cast in one with its jacket.

When in the state represented, the cylinder is tested hydraulically, and is scrapped if it fails to withstand 500 lbs. internal pressure. After passing this test it is placed in a mould for the purpose of having the matrix for the depositing process cast upon it. This matrix is white metal with a low melting point, and this is claimed not only to be more satisfactory than wax as a surface for deposition, but to lend itself to more accurate moulding and the production of a smooth and even surface. The copper

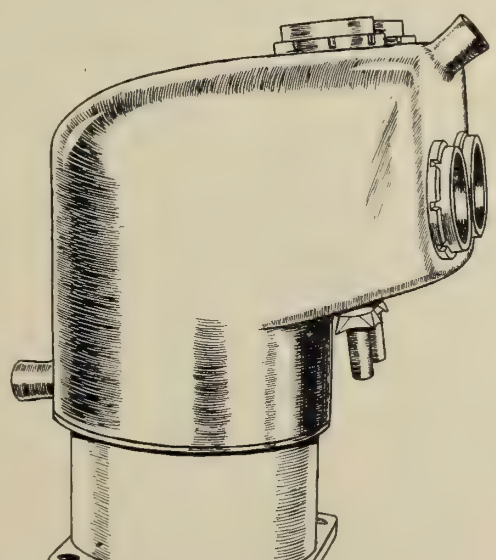


Fig. II.

water jacket is deposited in forty-eight hours, which must be considered a quick piece of work. There is no question of lack of contact, however, for an examination of turnings from the joint shows that there is perfect cohesion. When sufficient copper has been deposited the matrix is melted out by the immersion of the cylinder in a bath of hot oil with a high flash point. The cylinder bore has then a fine finishing cut taken off it to remove any injury done to the surface by contact with the electrolyte. Fig. II. illustrates the shape of the jacket, and shows also the water-inlet and outlet nipples, which are given a radius at their junction with the jacket, to enhance their strength and prevent the possibility of vibration of the pipe causing them to break away. Reference to Figs. I. and II. will show the freedom of the waterways. It must be remembered that when on the engine the cylinders are in such a position that the line joining the inlet and outlet nipples is a vertical one.

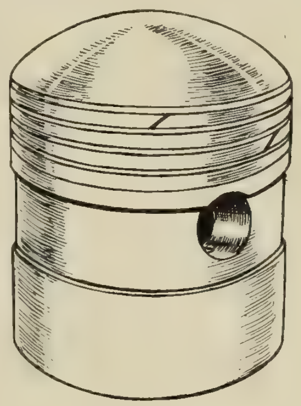


Fig. III.

The bore and stroke of the 60 h.p. engine are respectively 105 mm. and 110 mm., those of the 40 h.p. being 85 mm. and 90 mm.

Fig. III. illustrates the design of the cast steel piston, which is machined inside to a considerable extent, and the piston head has a large—possibly too large—degree of convexity. There are four rings, and the hollow gudgeon pin, Fig. IV., is secured by taper pins; its diameter is 20 mm. and it is drilled with a 15 mm. hole, while the gudgeon pin bush is of steel hardened and ground, this material having been found to give the greatest satisfaction. The connecting rod is a drop forging circular in section and drilled with a stepped hole. It tapers slightly from the small to the big end, the latter being split in the usual way and furnished with white metal linings cast direct upon the steel shell, which is fluted for that purpose; the width of the big end bearings is the same as their diameter, namely 38 mm.

The four-throw crankshaft is supported by five large diameter ball bearings, four of which are of the ordinary radial type, while that at the dead end of the base chamber is of the composite radial and thrust variety, and is so ar-

ranged that the thrust of the direct-coupled propeller can be taken in either direction. The diameter of the shaft at the journals is 60 mm., and at the pins 38 mm. Both shaft and pins are bored with a 25 mm. hole, and the webs are bored for oil passage, the holes being sealed with brass plugs driven in. The ball races have an outside diameter of

Fig. IV.

110 mm. and the internal rings are secured to the shaft by split collars in the manner shown in Fig. V. These collars are slightly taper and when squeezed into position are secured by a set pin driven into the crank web. The outer ring is pressed into an I section aluminium ring, which is furnished with a stop flange as shown, the web of the ring being drilled. The middle pair of crank pins are adjacent to one another and this disposition necessitates the use of a three-piece split collar for the attachment of the ball race, otherwise Fig. V. applies to all the bearings. The open end of the crankshaft is tapered to a degree of 5 per cent., and is furnished with a 90 mm. by 10 mm. by 6 mm. key. The length of the taper is 100 mm., and its extreme diameters are 42 mm. and 37 mm. At the other end the crankshaft extends beyond the timing gear-box, and is drilled and tapped for the reception of a starting handle claw.

The crank chamber itself is a good piece of foundry work, and for lightness combined with strength and the provision of means of ready access to the interior is a creditable piece of design. At the timing gear end the chamber is open, thus allowing the crankshaft complete with its ball races to be introduced into the annular webs which support it, one of these webs being indicated in the background of Fig. VIII. When in position the aluminium outer rings are secured by set pins.

Fig. VI. shows the principal dimensions (the length over all is 888 mm. in the case of the 60 h.p. and 718 mm. in that of the 40 h.p., the height and breadth being 496 mm. and 458.5 mm. respectively) and shows also the arrangement of the timing gearbox. The pinions are extremely light, the breadth across the face of the teeth being no more than 8 mm. Fig. VI. also shows the disposition of the magneto upon a bracket screwed to the camshaft casing. It is driven at twice crankshaft speed, and its driving shaft is extended forward to operate the centrifugal water pump.

The camshaft, which is carried by two outside ball-races and two intermediate plain bearings, is of 6 per cent. nickel steel, and is cut out of the solid bar, complete with its sixteen cams and two journals, on a special Webster and Bennett profiling machine. The diameter of the shaft is 25 mm., and it is drilled down its whole length with a 16 mm. hole. In the case of the 40 h.p. engine, ignition is by means of an eight-cylinder Gibault magneto, but in the 60 h.p. motor the distributor, instead of being incorporated in the magneto, is separate and is mounted upon an extension of the camshaft, this arrangement being illustrated in Fig. VI.

The exhaust and inlet valves are side by side, and are inclined to the axis of the cylinders in order to reduce the combustion space as far as possible, and to obviate the necessity for giving the valve chamber a big overhang to allow the centre line of the valves to intersect in the centre of the camshaft. The valves themselves are 42 mm. diameter across the face and have 8 mm. stems. They have a ring of pure nickel electrically welded upon them to form a face free from the likelihood of pitting, and the cams actuate their tappets direct through the intervention of a large steel ball carried in the lower portion of the phosphor-bronze tappet guides.

The carburettor is a Zenith placed amidst the cylinders and delivering gas through inlet branches, which diverge from a spherical centre.

It is, perhaps, not too much to say that the greatest desideratum of an engine for aeronautical work is a reliable lubrication system, and this fact was realised quickly by the designer of the engine in question. Reference to Fig. VII. will make clear the construction of the direct acting plunger pump, which is driven by a half time wheel keyed to the crankshaft. This wheel has, integral with it, a boss which forms an eccentric having a throw of 15 mm. This eccentric has an external groove turned in it, and holes are drilled down this groove and through the walls of the hollow crankshaft, which is plugged at both ends. The eccentric strap which is applied to the sheaf has a tubular extension which forms the piston of the plunger pump, and the strap is drilled so as to afford a direct and constant communication between the interior of the piston and the crankshaft. The barrel of the pump is a brass tube having a tubular T-piece, which is mounted on a tubular steel pin forming a trunnion to accommodate the angularity of the eccentric strap. Both the piston

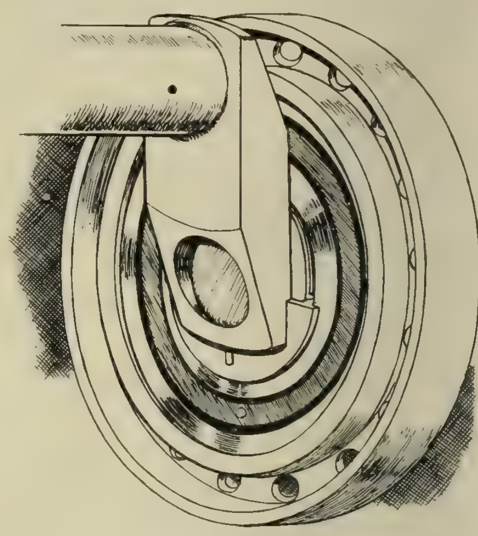
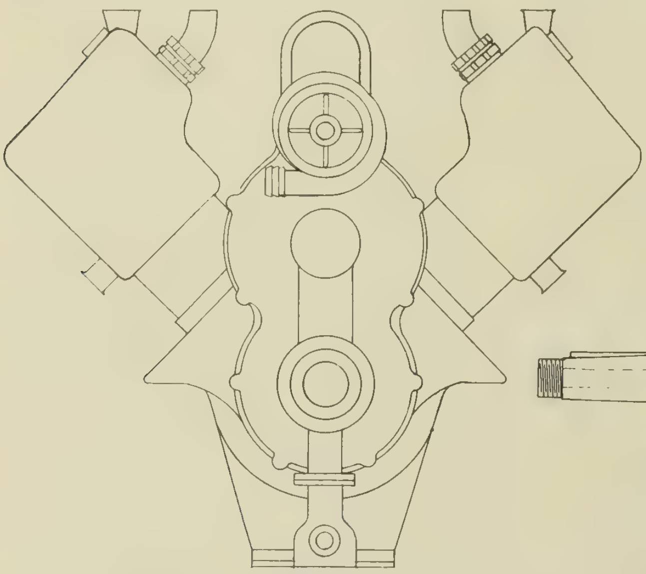


Fig. V.

and the barrel of the pump are furnished with ball valves at their lower extremities, the balls being kept in a cage formed by their seatings and by a couple of crossbars (not shown) a few millimetres above them. The pump barrel and its attachments are mounted in an aluminium case, which is secured by a flange and registers with the lower extremity of the timing gear cover. The steel rocking pin is drilled so as to be in constant communication with the pump barrel through its ball valve, and at one end it is connected up with an external tube which proceeds from the sump cast into the base of the crankchamber.



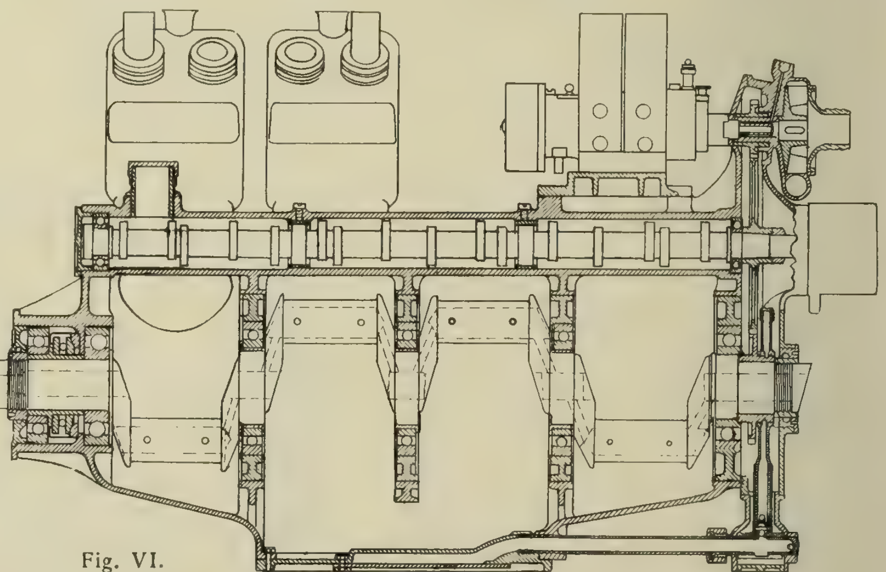


Fig. VI.

The details of this sump are illustrated in Fig. VIII., whence it will be seen that the chamber is sealed by a detachable base plate. In this is cast a shallow circular box, the bottom being formed by a spun copper hemisphere, and the contents can be drawn off through a cast-in duct, connected up, through the aforementioned external tube, to the trunion pin of the plunger-pump. The duct is foreshortened in Fig. VIII., but is shown

clearly in the side elevation in Fig. VI.

The wall of the shallow box is pierced with holes, which permit oil in the sump to be drawn to the plunger pump through the hemispherical basin, but underneath the row of holes there is a gauze dia-

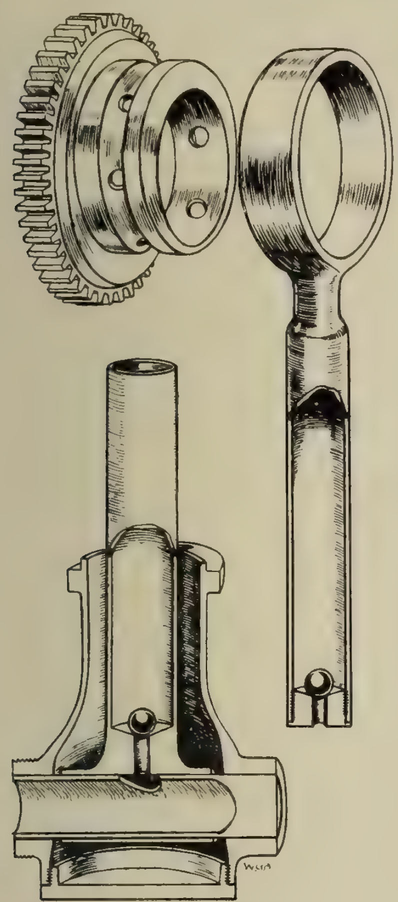


Fig. VII.

phragm which acts as a filter, and effectually prevents any solid matter circulating with the oil.

From the sump the pump forces oil into the hollow crankshaft, whence it passes to the big ends, up the hollow connecting rods to the gudgeon-pin bushes, and out through the hollow gudgeon-pins to the cylinder walls. By dimensioning the delivery holes in the gudgeon-pins carefully, it has been found possible to obtain a positive supply of oil to the pistons, which in no circumstances amounts to sufficient to cause carbonisation and a smoky exhaust.

The plunger has a diameter of 20 mm., whilst the width across the seating of the ball-valve is 8 mm. Each stroke of the pump delivers a volume of oil about equal to a tablespoonful, and at the usual engine speed the pressure of this delivery is about 25lbs. to 30lbs. to the square inch. It will be seen that when the engine is turning at 1,000 r.p.m. the delivery of oil is at the rate of about 10½ pints per minute. A constant level of oil is maintained in the sump by a float which operates a ball-valve through which oil flows from the main supply tank. The ball is secured in a cage formed by bent over segments of brass tube and, the tappets being adjustable, any desired level of oil can be maintained.

As eight cylinders are used a continuous turning couple is exercised upon the crankshaft and the limits of value of this couple are fairly close together; thus there is no need for a flywheel when the engine is attached directly to the propeller, though one would, of course, be fitted when the drive was through chains.

The engine gives its stated power at about 1,200 revolutions per minute, but the 60 h.p. is claimed to give 70 h.p. continuously, and the 40 h.p. about 48 h.p. The weight of the 60 h.p. complete, including piping, carburettor, and magneto, is 287 lbs., whilst the 40 h.p. scales 155 lbs. The petrol consumption of both engines is approximately .6 pints per horse-power-hour, and of oil .08 pints per horse-power-hour. The latter figure compares more than favourably with consumptions of most of the existing aero engines and has the great advantage that, as only a small volume of oil need be carried, the petrol tank may be made large.

At the present time, in monoplane design at all events, the diameter of the propeller is considerably circumscribed, with the result that a high-speed direct-coupled propeller becomes necessary. This, as has already been pointed out in "The Automobile Engineer," is an arrangement which by no means makes for economy, and it is inevitable that, if great distance of uninterrupted flight is to be obtained with high-speed machines, propellers must be considerably enlarged. Indeed, the only obstacle to their being increased in size—and consequently in efficiency—lies at present in the fact that a reduction gear of some sort is necessary, pending the arrival of an engine which is able to give a good power output at a moderate speed, such as 600 r.p.m. It is

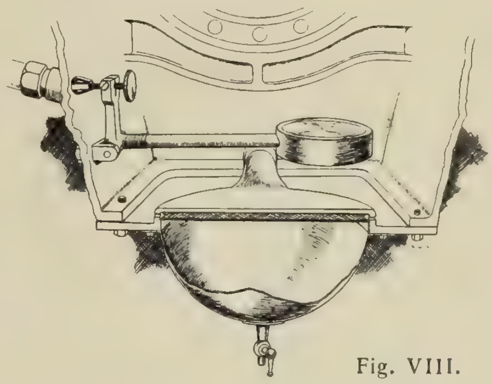


Fig. VIII.

in this respect that criticism can be levelled against these E.N.V. engines, for they are essentially of the high-speed type, and are not able to develop sufficient power at a low rate of revolution to make them suitable for direct coupling to a large-diameter propeller. This might easily be improved by a slight increase in the ratio of stroke to bore for, judged beside a modern high-speed car engine, their piston-speed is decidedly low, and it certainly appears that it would be worth while to increase it.

GEAR CUTTING FOR AUTOMOBILES.

A comparison of the different processes in use, and some hints on the manufacture of accurate cutters.

By Henri Perrot and Maurice Jerome.

It is well recognised, by all who are interested in the matter, that, at the present day, the greatest difficulty encountered in the manufacture of automobiles is the correct cutting of the various gears. Silence is an essential feature of the modern car, and even if this be obtained as regards the engine and rear axle, the problem of rendering the sliding pinions in the change-speed gear box quiet still remains. As quiet operation depends largely on accurate tooth form, it is hoped that the present article, founded on practical experience of gear cutting, will assist to some extent all those who are striving to obtain quiet-running gears.

There are three principal types of machine for cutting spur gears—

(1) Special automatic milling machines.
(2) Hobbing machines.
(3) Planing machines with the cutter in the form of a pinion.

In the special automatic milling machines a circular milling cutter is used, having as profile the shape of the space between the teeth. The operations necessary for making the milling cutter are turning, cutting, backing off, hardening, and grinding, and after being ground it is essential that the milling cutter should fulfil the following conditions:—

(1) The axis of symmetry of the cutter's profile must be perpendicular to the axis of rotation.
(2) It is necessary that all teeth should revolve in exactly the same path.

It is not proposed to go into the question of the cutter's profile here, nor to discuss what means to employ for obtaining cutters strictly true to this profile, as it is hoped that these questions will be gone into later on in a subsequent article.

What concerns us immediately is to enquire into the causes which sometimes render a milling cutter defective, even supposing the profile of the forming tool to be perfect.

If the division for cutting the milling tool be not correct, and yet in dividing for backing off it should be right, the profile of the teeth, although the same for all, will be at a variable distance from the centre (see Fig. I.) As a result only a few of the cutter's teeth will be working, and the only remedy for this is to grind up the gear correctly after hardening.

If the division be not exact when backing off the cutter, or if a few finishing cuts be not taken, to finish the backing off without putting any cut on, the result will be the same as in the preceding case, with this difference however, that the remedy will consist in grinding each tooth progressively until they are all equidistant from the centre (see Fig. II.) Should the centre of the milling cutter not be strictly perpendicular to the axis of rotation, the tool is defective beyond hope of redemption.

Milling cutters are apt to become deformed in the hardening; they may become oval, may warp bodily, or each tooth may warp individually. If the flank of the cutter remains true, and there is no warping, the tool is correct. Of course, the teeth will not all be equidistant from the centre, and they will not all be cutting equally, but the profile obtained will be correct. Still, if the cutter has warped, and it be required for cutting gears at

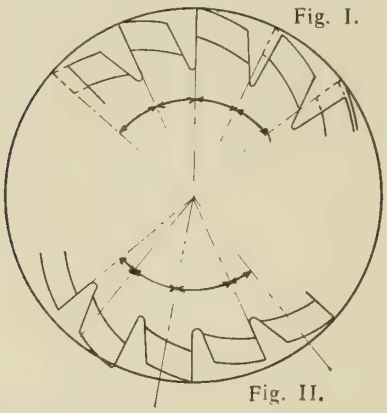


Fig. I.

Fig. II.

high speeds, then if it has more than a thousandth of an inch deformation it must be rejected. Having thus selected all the good cutters after hardening, care must be taken that, in grinding, the centre of rotation be not altered. A special fixture should be used which takes the shape of the cutter flank and grips it on its diameter, the fixture revolving absolutely true, and it is thus possible to grind the bore and faces dead true, the two faces being quite parallel.

It is not necessary that the centre of the cutter's profile should be exactly in the centre of its thickness. Certain manufacturers do this, however, giving the cutter a thickness exactly equal to the pitch. This makes it possible, in cutting racks, to mount several cutters on a mandril, thus cutting several teeth at once, and this arrangement has also its advantages in making special tools for cutting globular worms. Before starting to cut a gear it is necessary to make sure that the cutter revolves absolutely true on its faces, and also that the cutting faces are situated in a plane passing through the centre of the pinion. These two conditions are essential in obtaining a gear with teeth exactly similar to the profile of the milling cutter, and for obtaining a high efficiency it is also necessary that the cutter should revolve absolutely true.

These few preceding remarks will serve to show what difficulties have to be overcome in making high-class milling tools, and besides these there is still the forming tool, for backing off the cutters, which we have not considered.

Milling cutters are ground up mechanically nowadays on machines such as that shown diagramatically in Fig. III.

The cutter is mounted on a mandril, and tightened up against a washer divided to exactly the same number of teeth. Each tooth is brought to bear on an index, and the cutter is ground by giving the slide a backward and forward motion in front of the emery wheel. It is essential that the emery wheel should be quite stiff, and its centre must coincide exactly with the centre of the cutter; also the emery wheel should not bear too hard, otherwise there is a risk of burning the thin cutting edges of the milling tool.

If a section be taken off a milling cutter tooth perpendicularly to the radius a form such as shown in Fig. IV. is obtained. It can be seen that the cutting part is backed off on the side, but that the front face is square with the direction of cutting. The tooth operates in like manner to a tool which performs its work by crushing; it is not like a lathe tool, which separates the cuttings from the mass. It is therefore easy to find out the exact pressure which the milling teeth will be called upon to stand, knowing the cutting area of the tooth and the coefficient of resistance to compression of the material to be cut, but this will only give an idea of the compression as regards the advancing speed of the milling cutter. It is, however, necessary that the edge of the tool should stand up while cutting several blanks without being re-sharpened, and this renders necessary two cutting operations—one a roughing out and the other for finishing.

The work done in compression produces a certain quantity of heat, which spreads itself over the material which is being cut, and over the milling cutter, and this may burn the cutting edge of the tool. The degree of heat cannot be estimated according to the hardness of the steel, as shown by the fact that when cutting a hard steel, provided the steel does not attain the hardness of the milling cutter, the only heat produced is that due to the crushing of the metal, the milling teeth just cutting their way through. If, however, soft steel be operated upon, the metal is thrown up on the sides and comes back in position behind the teeth, pinching the milling cutter and producing extra heat.

For cutting hard materials it is only necessary to have very little backing off, so as to have a small clearance angle, but

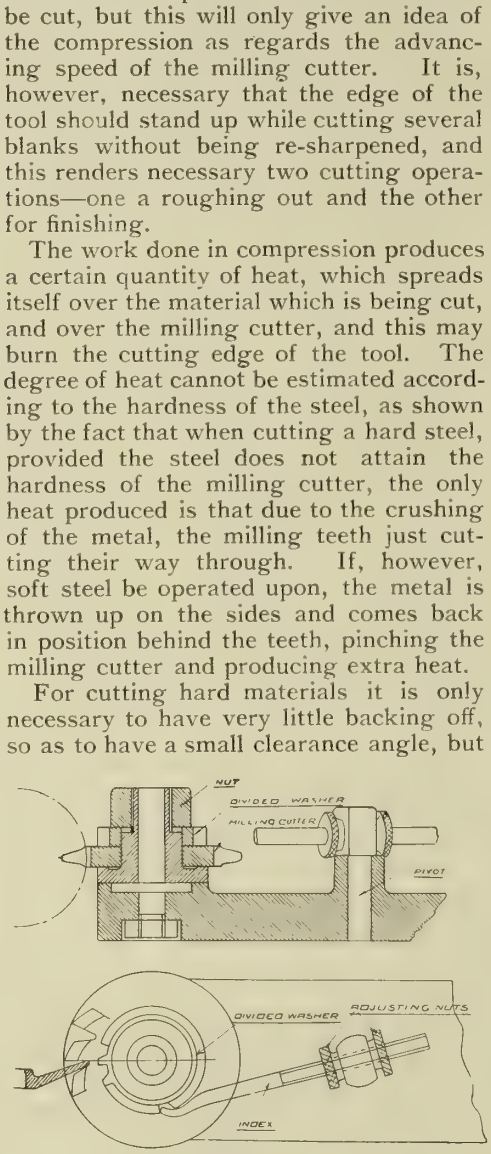


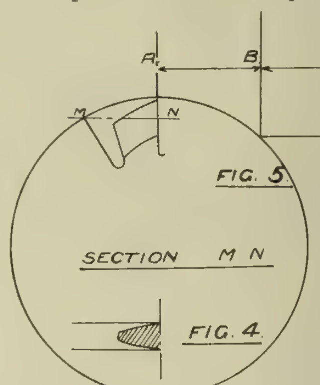
Fig. III.

a strong cutting edge. For soft work, however, a large clearance angle is needed, requiring greater backing off. This question of heating through friction when cutting soft materials leads to the adoption of a quick-feed together with a slow revolving cutter, whilst for hard materials it is better to reduce the rate of feed and increase the speed of the cutting tool.

Milling cutters intended for cutting gears must be maintained with a keen edge, and should be lubricated with a strong oil jet so as to abstract a great proportion of the heat from the pinion.

Minute care should be taken in the hardening of the milling cutter, so as to ensure the faces being as polished and regular as possible, and certain high-speed steels possess the disadvantage of acquiring a very rough surface when subjected to white heat.

Another disadvantage of heating, especially if the gear to be cut happens to be of large diameter, is that its temperature will not be the same all over, and the gear will expand unevenly, so causing errors in the division of the teeth. This is another point in favour of a roughing cut and light finishing cut. To diminish the chances of error due to heating the finishing cut may be so divided as to go several times round the gear before being completed. For example, supposing the



Figs. IV. & V.

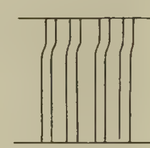


Fig. VI.

gear to have 72 teeth, then the gear can be cut with a division of 5/72 instead of 1/72, thus going five times round the gear before the latter is finished completely.

The automatic gear milling machine most in vogue is undoubtedly the Brown and Sharp automatic gear cutter. This machine consists essentially of a large surface slide carrying the milling cutter, and travelling in a line parallel to the mandril on which the gear blank is mounted. The mandril carrying the gear blank is provided at one end with a dividing mechanism.

The following are the automatic movements accomplished by this machine:—

(1) Forward motion; the milling cutter, revolving and feeding through the gear blank.
(2) Disengagement of the forward motion, and engagement of the reverse.
(3) Stopping of the reverse motion and division, at the moment when the feed mechanism is disengaged.
(4) Engagement of the forward motion.

If AB be the length lost in disengaging the cutter and L the breadth of the gear blank, the total travel will be $L+AB$. If n be the number of teeth in the gear to be cut the time taken will be inversely proportional to the travel a per minute or $\frac{(L+AB)\,n}{a}$ If instead of cutting a single blank by itself N blanks be cut, the time taken to cut one blank will be $\frac{(NL+AB)\,n}{aN}$,

Assuming the return stroke is equal to A and the time taken in dividing is equal to D. When cutting one blank by itself the time taken will be:—

$$\frac{(L+AB)\,n}{a} + \frac{(L+AB)\,n}{A} + nD$$

By cutting N blanks together the average time for each gear will be:—

$$\frac{(NL+AB)\,n}{aN} + \frac{(NL+AB)\,n}{AN} + \frac{nD}{N}$$

Assuming $L=20$ mm $a=25$ mm per min.
$AB=25$ mm $n=25$ Teeth.
$N=10$ mm $A=5400$ mm per min.
$D=\frac{3}{60}$ minute

Then by employing the method corresponding to formula (1) we find the time required to cut a gear is 48 minutes, whilst with the second method we obtain from formula (2) 24 minutes, or a saving in time of 50 per cent. without taking into account the time saved in setting up. This shows clearly the advantages to be gained by mounting a number of gears together on the mandril and cutting them together.

It is essential in a good gear milling machine that

(1) The table and the mandril be absolutely parallel.

(2) The dividing mechanism be very accurate, and not liable to wear or get out of order. For this purpose a worm of large diameter and small pitch should be used, and as few parts as possible likely to develop play should be employed between the worm and the dividing mechanism.

(3) The centring of the milling cutter should be very accurate, i.e., means should be provided for making sure that the plane of symmetry of the milling tool coincides with the vertical plane passing through the axis of the gear blank mandril.

In the Brown and Sharpe type of machine the dividing mechanism is only working while the machine is running idle, and wear is not much to be feared if the machine is well looked after.

Also the parallelism of the table and mandril cannot change if the bearing surface of the table is sufficient and is suitably oiled.

The centering of the milling cutter is performed by an instrument, which is provided with the machine already set. This instrument consists of a needle oscillating round a vertical axis in the vertical plane of the mandril axis; the point of the needle moving over a divided quadrant, zero being situated in the said plane. By the aid of this instrument it is possible to see whether the milling cutter turns true, by setting it to one tooth, and seeing whether all the other teeth give the same setting in this position. If the milling cutter is not bored true with the plane of symmetry this can be corrected by inserting thin paper packing, but if the teeth are out of truth then the only possibility is to strike a suitable average. Of course, the operation of truing up the milling cutter should be done with minute care, even if it occupies a long time.

The reciprocating movement of the slide is obtained by means of stops, which first throw over the operating mechanism, and then start the dividing mechanism when the milling cutter is clear of the pinion.

These stops, which are situated on the side of the table, compress the clutch-actuating spring before the clutch-operating gear is brought into action; so this takes place whilst the milling cutter is still meshing with the gear blank. If, therefore, there be any play in the slide the latter will be pulled slightly over, and the milling cutter will become slightly eccentric, digging into one side of the gear blank, and leaving extra metal on the other side (see Fig. VI.). However accurate the slide may be this disadvantage always remains, and the only remedy is to give sufficient travel to the slide, so that the stop only operates when the milling cutter is clear of the gear blank. To get over this alternative, which lengthens the time taken in cutting, a stop should be used which is subjected to little load, and actuates a mechanism working at high speed. Still, the present system answers the purpose fairly well as long as the table has not too much play.

It is also equally important that the milling cutter spindle should have no end play, and a special adjustment is provided for the purpose. Should end play exist a single warped tooth in the cutter would take it up and spoil the tooth of the pinion.

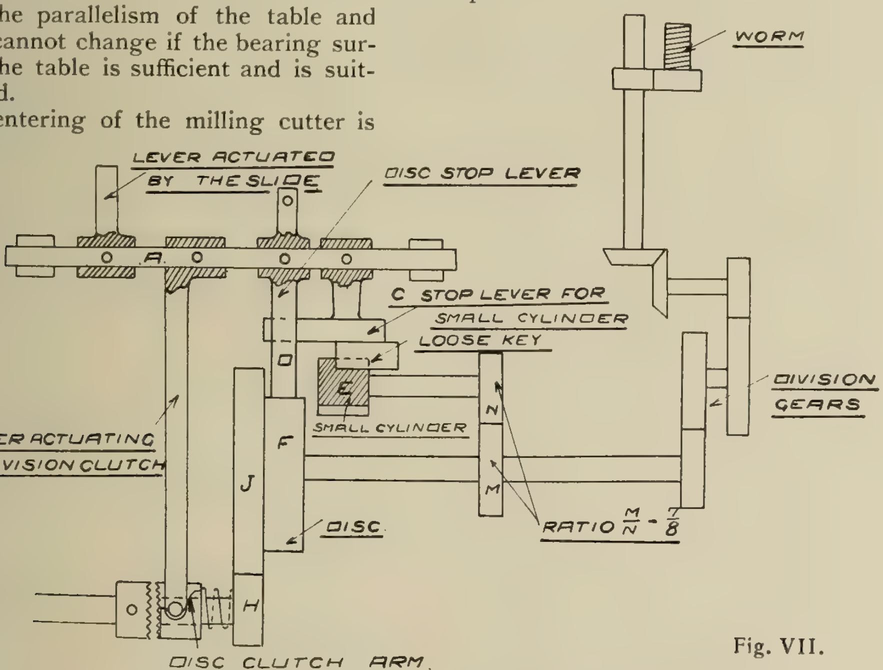


Fig. VII.

We have said previously that a good dividing mechanism should not have any parts liable to wear and become slack; but it is difficult to avoid complication, for the worm gear connected to the dividing mechanism must be provided with an up-and-down motion, in order to suit different sized pinions. It might be possible to fix the dividing mechanism on the work spindle head, but this would, however, increase its weight and complicate its operating mechanism.

It is therefore necessary to fix the main part of the dividing mechanism rigidly to the machine, and to connect up to the worm by means of bevel wheels, serrated shafts, and spur gearing.

The shaft A (Fig. VII.) carries three levers;—The lever B, whose head engages in a recess, and supports the clutch when out of engagement, rendering the dividing mechanism stationary. The lever C, provided with a finger piece, which engages in one of the slots of the little cylinder E; and the lever D, supported by C, and engaging in a recess in the plate F, which is operated by means of the shaft G and the gears H and J. The plate F is fixed to a shaft, which is connected to the dividing gears and the worm.

The stops on the slide push the shaft A slightly, thus freeing all the levers from their slots. The clutch engages under the action of a spring, and puts the mechanism in motion. This motion continues until the levers D and C fall again into their respective slots. At the same moment the lever B returning acts by means of a cam on to the clutch, throwing the latter out of engagement when it catches in its recess, thus stopping all movement of the plate F.

The plate F may vary its number of revolutions, as the gears m and n are in the proportion of 7 to 8, so that the cylinder E, having four slots, set so that the lever C engages one of them at the same time D engages its own slot, every two revolutions of the plate F the lever falls into a slot of the cylinder E, thus permitting D to fall into its slot, and so stopping the dividing mechanism.

The four notches of the cylinder E being of different lengths, and the finger on the lever consisting of a key, whose position may be varied according to the length of its slot, so as to disengage itself at the right moment, the dividing mechanism may be stopped in the following positions:—

(1) After one turn of the plate F without the use of cylinder E.

(2) After two turns of the plate F the cylinder E making one and three-quarter revolutions.

(3) After four turns of the plate F the cylinder E making three and a half revolutions.

(4) After six turns of the plate F the cylinder E making five and a quarter revolutions.

(5) After eight turns of the plate F the cylinder E making seven revolutions.

The dividing plate is locked in position by two stops acting in opposite directions, the one on the plate being the end of the lever D, the other on the clutch forming part of the lever B.

If the gears m and n or the locking key of the clutch take any play the plate F is no longer located properly; and, as a result, it may not turn quite the correct number of revolutions, thus straining the dividing mechanism. The remedy for this is to change the defective part. An alternative method is to take for dividing gears those necessitating the greatest number of revolutions of the plate, for by this means the error occasioned through lost motion is a very small fraction of the division. Care must also be taken that there is no play between the teeth of the worm and wheel, nor end play between the bearings.

Milling cutters as sold to the trade are made for a depth of tooth of twice the module plus 1-10th the thickness of the tooth, and are made so as to mesh without any play. It happens, however, that after cutting the gears have a little play.

caused usually through the milling cutter being slightly warped, and thus cutting a gap wider than it should do in reality. It is therefore better to set the machine according to the proportion of the thickness of the tooth to the pitch circle dia-

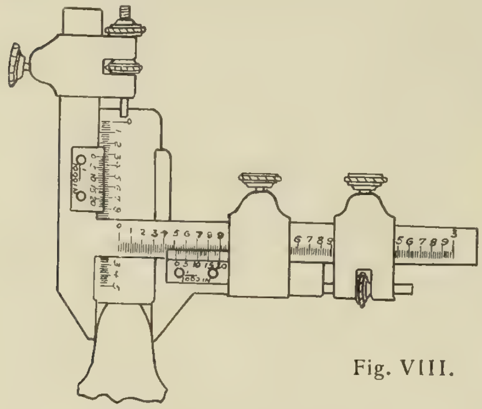


Fig. VIII.

meter, and not to the depth of tooth. All that is necessary for this is to have a tooth gauge as shown in Fig. VIII.

Trade cutters are classed in series, in which the same milling cutter is used for cutting several successive gears, and in which the cutter is correct for the smallest number. The designer should therefore try his utmost to arrange his gears so as to employ only those for which the milling cutters are exact. Should this not be possible we would recommend the following means as being the only way to arrive at an approximately correct result.

Given a pair of gears to be cut, turn up two blanks having the same diameter as the gears to be cut, and mill three or four teeth in each, care being taken that the thickness of tooth at the pitch line is correct. Then mount these gears in a fixture with adjustable centres, so that there is no play when the centre line through a tooth of one of the gears passes through the centre line of the gears. Then turn so that the centre of the tooth of the other gear passes through the centre line of the gears. If there should be play, then it will be necessary to cut fresh teeth and try again, this time cutting rather deeper into the first gear and less into the second, so as to preserve the right centres. This method, although imperfect, will be found sufficiently accurate in the majority of cases.

The best way, if there is a sufficient number of gears to be cut, is to make special cutters which are correct for the number of teeth, and these should only be used for the same train of gears with the same centres. Two standard cutters of the same number may be warped differently, and not give the same profile.

As a rule gears have a hole large enough to allow mounting them on a mandril of sufficient stiffness, and it is always better to try to fix the blanks on the machine in a similar way to that in which they are intended to be used, so as to avoid machining parts carefully which will afterwards have no utility.

There are exceptions, however, such as when the gears are cut in "gangs." In this case it is necessary that the faces of the blanks be exactly parallel, so that the mandril is not bent when tightening. Large diameter gears with a very small hole, such as, for instance, camshaft gears, may advantageously be centred on a mandril provided with a large disc at either end, one of them being fixed rigidly to the mandril, the other being loose. By tightening up these two discs one against the other, a very rigid means of holding the gears is obtained. Great care should be taken in fixing the mandril on the machine, and after being replaced it should be checked with a sensitive gauge to a thousandth of an inch.

The method of cutting gears by means of a hob is based on the principle that all pinions with involute teeth which mesh correctly with the same rack mesh correctly with one another. The tool will therefore take the form of a rack provided with a movement as if it were in gear with the pinion, whilst the latter will turn as if it geared with the rack, and to realise the correct motion between these two profiles a worm is used with a profile of tooth the same as that of a rack.

The gear blank is then rotated independently, as though a worm and wheel were meshing together, whilst the worm-carrying fixture moves bodily forward into gear. The worm can pivot on a table so that the tangent to the pitch circle spiral is projected horizontally if the table is horizontal and parallel to the axis of the gear blank, this being necessary for the hob to cut properly.

If we consider the projection of the pinion and hob on a plane perpendicular to the axis of the pinion, we see that the profile of a rack is obtained by the projection of the true profile, i.e., the section of the tooth will take the form of a rack tooth, and the pitch of the worm will be equal to $\frac{\text{pitch}}{\cos \alpha}$ α being the angle of the worm.

To make a hob, therefore, it is necessary to—

(1) Cut a screw with a pitch $= \frac{m\,\pi}{\cos \alpha}$ (where m = Module) with the normal profile of a straight sided rack.

(2) Cut spiral grooves at right angles to the spiral of the thread, of a number equal to the number of teeth required on the circumference, the cutting faces being radial.

(3) Back off the teeth so as to make a tool with a constant profile, which can afterwards be ground on the face of the teeth. This operation requires to be done on a special machine, giving exactly the same spiral curve as the thread of the hob, and provided with a differential gearing situated on the backing-off operating mechanism, and giving n tool cuts (n being the number of cuts per circumference) to the length of the spiral curve of the grooves.

(4) Hardening, grinding of the bore and faces, and then sharpening.

Everything concerning the making of milling cutters which has been said to be detrimental to their quality is applicable to hobs; but it should, however, be noted that it is all the more important that the appliances for cutting the thread and backing off give exactly the pitch $\frac{m\,\pi}{\cos \alpha}$, and that the appliances for backing off and sharpening have, the former its differential, and the latter its guide, giving the same spiral at right angles to the pitch circle spiral of the worm thread, otherwise the hob will not be concentric.

For the gear blank to turn a distance equal to the pitch it is necessary that the rack should advance a distance equal to the pitch, and therefore the worm (having only a single thread) will make a complete revolution. If we have ten cuts per circumference the profile of the tooth will consist partly of the curve going over these ten successive positions of the straight-sided tooth. The pitch being 12 m/m., these positions will be situated at a distance of 1 m/m. from one another. A tooth outline therefore is obtained composed of small faces. It is therefore advantageous to increase the number of grooves, which are only limited by the question of diameter, deformations in hardening, and the cost of the hob.

If the hob is not well made, only a few of the teeth will be working, and instead of having ten positions for the profile we shall have a far smaller number, and thus not produce such good work. If it is desired to have a symmetrical tooth it is necessary to place the hob so that there are five positions to the right and five to the left of the vertical plane passing through the axis of the pinion, and placed symmetrically in regard to this axis.

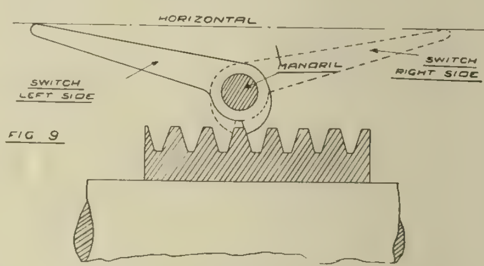


Fig. IX.

It is therefore necessary to centre a tooth of the hob in such a way that the vertical plane passing through the pinion's axis should cut it through the centre. For this purpose an instrument as shown in Fig. IX. may be used.

In the hobbing machine the pinion and worm both revolve, being coupled up by gearing such as spur wheels and shafts giving them the speeds required. The machine must therefore possess all the qualities of the automatic milling machine combined with strength in the gearing between the worm and mandril. Exceptional care must be taken that the spindle carrying the worm turns quite true, and that there is no play in its guides, while the same applies for the spindle carrying the gear blank, and the shafts should be held by long bearings of large diameter, and carefully lubricated.

With this machine excellent results may be obtained, but it requires a good deal of care and a hob irreproachably made. As regards speed, gears are cut much quicker by this process than by milling, owing to there being no loss of time and also to there being several teeth working at once, while it is also possible to cut deeper into gear without any other inconvenience than having a little play in the teeth.

However, before using such gears it is advisable to grind them up slightly, so as to wear off the small ridges formed on the teeth.

The Fellow machine consists of a cutting tool in the form of a pinion with involute teeth having two movements; one being a backward and forward motion giving the cut, the other a rotating movement of the mandril carrying the gear

blank. During the return stroke of the tool the whole fixture carrying the gear blank retires out of mesh under the action of a cam, and then comes quickly into position again ready for the forward cutting stroke.

For rotation of the cutter a worm gears with a wheel, in the interior of which slides the rack turning the shaft which carries the tool, and giving it a backward and forward movement. This shaft has a semi-circular slide at one end, which bears on its flat face with another face forming part of the worm wheel, the worm wheel conveying its rotary motion to the slide. All this mechanism is strong, and is encased so as to protect it from dirt, while the lubrication is carefully provided for.

The motion of the pinion is obtained by means of a worm and wheel operated by a train of gears, which may be varied according to the number of teeth to be cut. All this is carried on a mechanism which is articulated round a centre by a connecting rod. Each end of the connecting rod carries a gear, the two meshing together, and the articulation is provided so as to allow for the pinion and tool retiring from one another on the return stroke. The rotary movement of cutter and pinion starts from a common centre.

The most delicate part of the machine is the cutter. For it to cut properly it must be backed off on each tooth, and after being sharpened it must still possess the same profile; also the teeth must all be absolutely parallel, as each one cuts its tooth in the blank, so that it is also necessary that it be absolutely concentric.

Theoretically it is simple enough to obtain a tool fulfilling these conditions, but the practical difficulties are considerable, for tools of a most precise nature are necessary, combined with highly-trained men, if anything approaching good results are to be obtained, and thus the Fellow type of machine necessitates extremely careful workmen, quick at acquiring the knack of centering the mandrils and of stopping the machine at the right moment.

As with the hob, a slight error in depth of tooth only results in a slight weakening of the tooth, while the meshing remains correct.

When all preceding precautions have been taken, according to the type of machine used, the gears must be verified by trying them two by two. This verification may be carried out either with the aid of a special fixture having its centres exactly correct, or else in a special gear box made to exact centres and used only for this purpose. The gears :—

(1) must not be too stiff to turn,
(2) must have the necessary clearance at the bottom of the teeth,
(3) must mesh on the whole breadth of tooth and on the whole length of the involute curve,
(4) must have a minimum of play between the teeth.

Heat or mechanical treatment after cutting should be avoided as much as possible, so as not to cause any deformations in the gear, for, unfortunately, the rectification of the gears after hardening is a very onerous task. There are certainly some steels which do not change their shape in hardening, or, at any rate, do so to a very small extent, but by employing these steels other troubles are liable to be introduced.

Of course, it does not follow, by any means, that a pair of gears with teeth that are absolutely accurate will necessarily operate in complete silence. Accuracy of tooth form is only one of the factors in the problem, but it is an undoubtedly important factor. One of the greatest advantages of well formed teeth is that the durability of the gears will be greater than it would with inaccurate teeth, and the noise of operation will not be likely to increase as wear takes place.

THE 17 H.P. MAUDSLAY CHASSIS.

A chassis which possesses several interesting peculiarities.

THE 17 h.p. Maudslay is representative of a type of car which has received but scant attention at the hands of manufacturers, during the last few years. That is to say, it is a car of moderate power and carrying capacity, but it commands a comparatively high selling price, and is not designed primarily from the point of view of low cost of production. Throughout the chassis there are refinements to be found which are usally only present on larger cars, and though there are some notable omissions, which will be mentioned in due course, the general attention to detail is above the average.

The general arrangements of the chassis shown in Fig. I. give a good idea of the accessibility of the principal parts, as the cleanliness of outline has not been achieved by the omission of many essential details. The method of attachment of the engine is also shown most clearly in these views, which make plain the large amount of additional free space round the engine, which is obtained by the absence of the usual crank case arms, though this is less needed on an engine with overhead valve gear, and the engine follows accepted Maudslay practice as regards the disposition of the latter. Though rated at 17 h.p., the bore is 90 mm. and the stroke 130 mm., so the power developed is considerably in excess of the nominal. Referring to the sectional illustration (Fig. II.) it will be noticed that the cylinders are cast in pairs, and that each cylinder head is enlarged for the reception of the valves. That the valves are small compared to the bore of the cylinders may be noticed likewise, but their area is not much less than that of the valves of many other engines of similar volumetric dimensions, and the lift is rather above the normal. There are two principal advantages of the overhead-valve system of engine construction, the first, and most important, being that it is easy to obtain exactly equal cylinder volumes, as the whole of the combustion space is machined, and the second that the accessibility of the whole of the valve gear is exceptionally good. The disadvantages are that it is not easy to use really large valves, and great care has to be taken to ensure an ample circulation of water in the neighbourhood of the valve cages, to prevent warping or even cracking due to unequal heating. The relative importance of the advantages and disadvantages must always depend upon the peculiar purpose for which the engine is required, but when, as in the present instance, smooth running is adjudged to be of greater value than high power (as compared with dimensions) then the principal disadvantage is the depth of the engine, and this is perhaps not a disadvantage considering the present trend of body design.

From the workshop point of view the Maudslay engine is probably slightly more expensive than the standard type, as the number of parts are greater, but against this must be set the fact that the crank case is a very simple job, and the cylinder pairs are also free from awkward operations. The open ends of the cylinder castings are of undoubted assistance in moulding, and they provide a very free and efficient central waterway between the cylinder pairs. By bolting together the two castings the rigidity of a monobloc is obtained (and also the attendant disadvantages), while the use of the fan spindle bracket, and the magneto bracket to seal the ends is effective and neat. As the chassis is not being turned out in exceptionally large quantities, the cylinders are bored separately, no special plant having been laid down to deal with both cylinders of a pair simultaneously. The valve chamber pockets are also bored singly, but all drilling operations are conducted with the aid of a single box jig having spigots to centre in the already finished bores. Each casting is marked off before the preliminary operation of facing the underside, this being done to enable the wall thickness round the valve cages to be kept as nearly equal as possible. The valve cages are ground to the cylinders, and are secured by the threaded rings shown in Fig. II. They are all interchangeable, as there is a dowel peg on each in the same position relative to the port, these pegs registering with slots cut in the cylinders.

In order to assist the smooth running qualities of the engine, the pistons are balanced by pairing, and their actual weight varies from about two pounds seven ounces to two pounds nine ounces. The material is ordinary good cast iron, and we are informed that efforts are being made to reduce the average weight to two pounds, without change of material. There is nothing regarding the rings which calls for special comment, except perhaps that the scraper ring is rather higher up than usual, and the oil which it removes from the cylinder walls is passed through a ring of small holes just beneath it to the inside of the piston, instead of being discharged at the bottom. The gudgeon pin might be more securely fixed, as the safety of its attachment depends upon a moderately small split pin in the head of the tangential locking screw, but owing to the fact that any turning of the gudgeon tends to lock the screw, its security is very greatly in excess of the ordin-

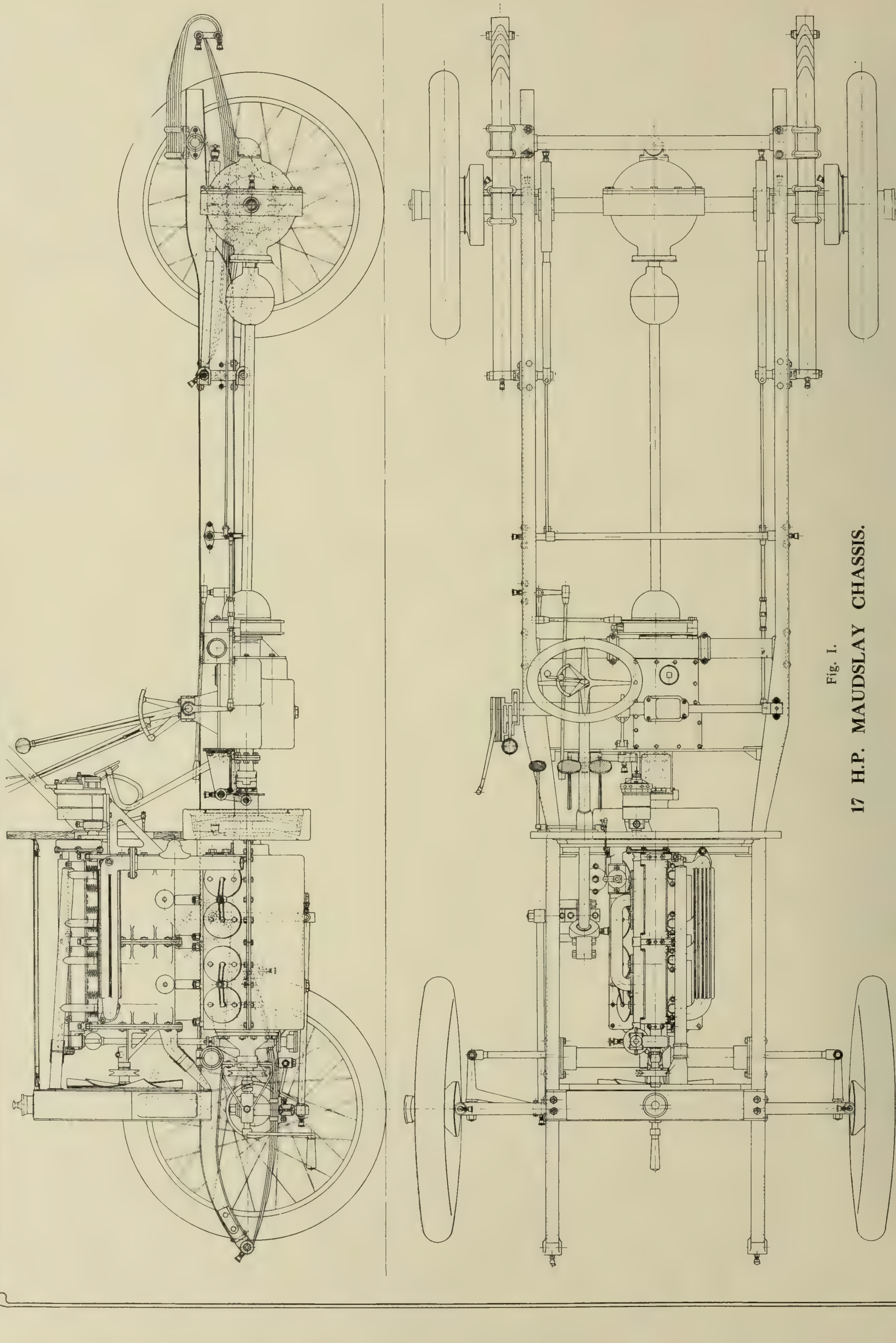


Fig. I.

17 H.P. MAUDSLAY CHASSIS.

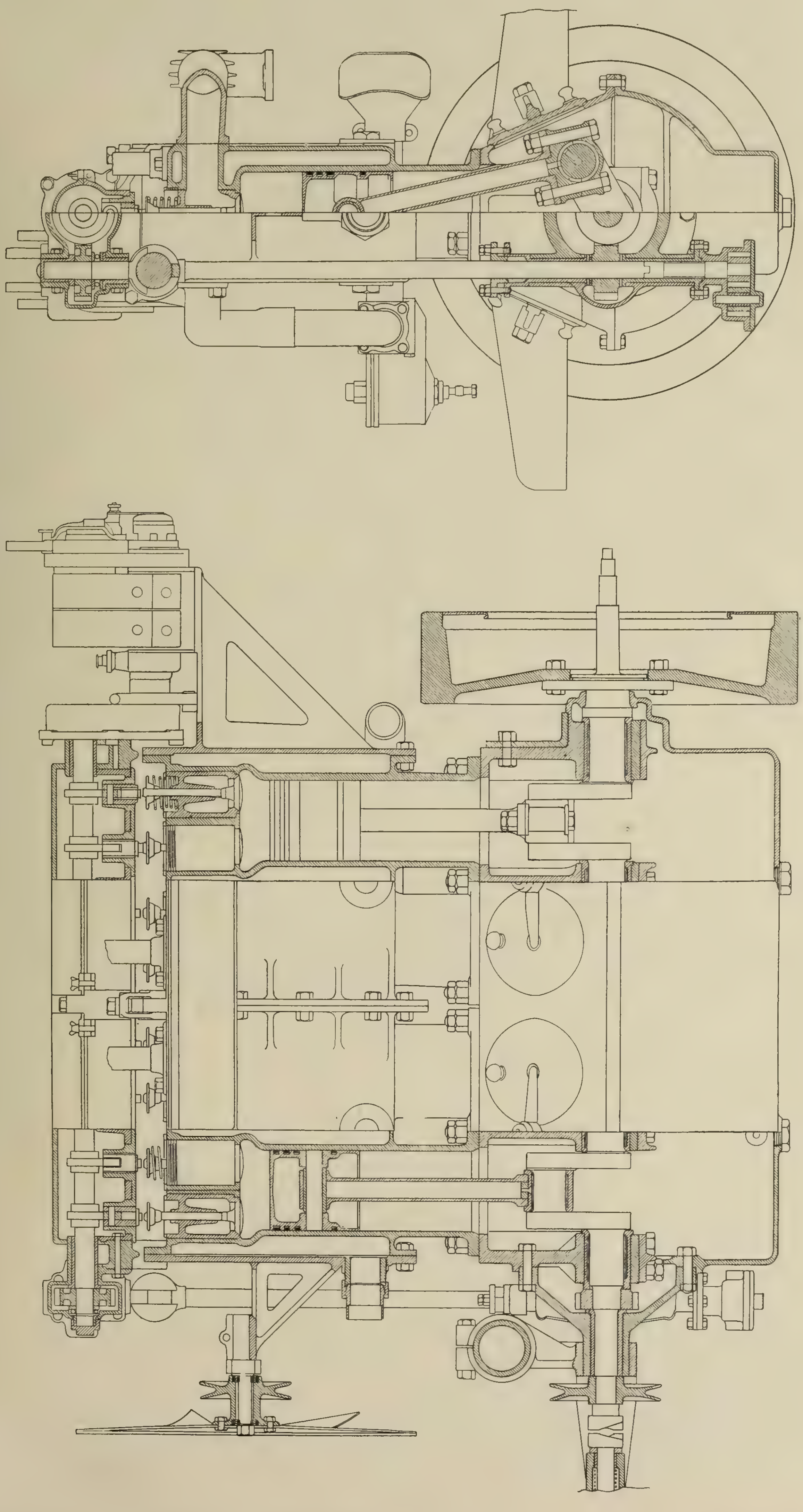


Fig. II.

17 H.P. MAUDSLAY ENGINE.

ary set pin, though it is not the best form of lock. The connecting rod is almost exactly similar to the Lanchester and is very light, but a stamped type of rod, with an external copper oil pipe, is being experimented with, and may possibly be adopted as standard for next year, the chief point in its favour being its lower cost. Phosphor bronze is used for the small end bushes, and white metal, set in bronze, for the big end bearings. The crankshaft is turned up from a forging, and is of a 40 ton steel of the following composition :—

Carbon	.4
Silicon	.15
Manganese ...	.75
Sulphur ...	.05 (not exceeding).
Phosphorous ...	.05 (not exceeding).
Maximum stress	41.02
Elasticity	27.5
Elongation in 2ins.	28.5
Reduction of area	45.4

The diameter of the main journals and pins is $1\frac{1}{2}$ins., while the aggregate length of main bearing is $10\frac{1}{2}$ins., and each big end is $1\frac{7}{8}$ins. long. It should be observed that the main bearing caps are held up by long bolts, which also serve to secure the cylinders. Each of the five main bearings is supplied with oil from a gear pump situated outside the bottom of the crankcase at the front end, as shown in Fig. II., the oil being sucked from the sump and delivered to a longitudinal passage drilled in the upper half of the crank case. From each main bearing oil passes to the big ends through the crankshaft, and is finally led to the small ends by one of the methods already mentioned. Thus the lubrication is entirely forced except to the camshaft, which has a separate supply distributed locally by splash.

The camshaft and oil pump drive is taken from the same skew gear at the front end of the crankshaft, the pump being driven by a tongue coupling and the camshaft by a universal coupling. This universal is necessary on account of the method of obtaining access to the valves, which is by swinging the whole camshaft case on hinges which are not seen in Fig. II., but are concentric with the centre of the universal joint. This joint is of the same pattern as that used for the transmission on the 8 h.p. Rover, but is, in this case, made entirely from phosphor bronze. To swing the camshaft it is necessary to turn the engine until one of the slots on the ball is in a plane at right angles to the axis of the shaft, and damage might ensue were an attempt to turn the engine by hand, made at a time when the camshaft was swung

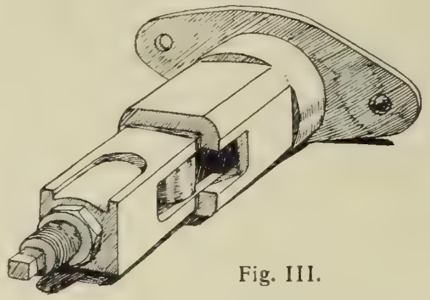


Fig. III.

over, so there would appear to be some advantage in a disconnecting type of coupling. However, the chances of accidental injury of the character we have suggested is probably more or less remote, and the ball type of joint has the particular convenience of being easily separated if it is desired to remove the camshaft altogether. It should be noticed that a small ball thrust is used to back the skew gears at the top of the vertical shaft.

The cam case is gun metal and the phosphor bronze bearings are mounted therein, the aluminium cover caps being no more than their name implies. Separate cams are used, attached by pegging in the usual way, and the design of the tappets, to obtain good bearing surface in small length, is distinctly ingenious. Fig. III. shows the construction of a tappet separately, and from this illustration, together with the section shown in Fig. II., the method of mounting the roller will be made clear without further verbal description. One small advantage of the large sectional area of the tappet piece perhaps deserves mention, and that is that secure locking of the adjustment of the valve strikers, is easily and safely accomplished.

The position of the magneto has a good deal to recommend it from the point of view of convenience and short wiring, but its cover is scarcely an ornament to the dashboard, and any noise which may arise from its driving wheels is in close proximity to an excellent sounding board, and equally close to the passengers. As there are no valves at the sides of the cylinders, there would appear to be points of advantage in placing the magneto in a more usual position, though it might be difficult to provide a convenient driving arrangement. All things considered, across the front of the engine above the frame attachment should be the best possible place, and a skew gear drive could then be taken from the vertical shaft.

From the erecting shop standpoint, the attachment of the engine to the frame is highly commendable. The front end bracket is malleable iron, being placed on the tubular cross member before the frame is built up, and the cylindrical end of the crankcase has simply to be inserted in this bracket, the two parts being free as regards relative movement, which means that the front end of the frame is free to whip if necessary, and such whipping does not throw any stress on the crankcase. The rear end attachment is by means of a channel steel pressing, which is assembled as a part of the engine and is virtually a cross member of the frame when bolted up in place. The gearbox is attached in an exactly similar way as shown in Fig. I., and the one weakness of the system lies in the possibility of aligning troubles. These, however, are always present when the engine and gearbox are secured separately to the side members of the main frame, as any deflection due to road shock will be at a maximum at just about the plane of the clutch coupling. In the illustrations which accompany this description, an Oldham coupling is shown, but this is shortly to be replaced by a more flexible type of joint.

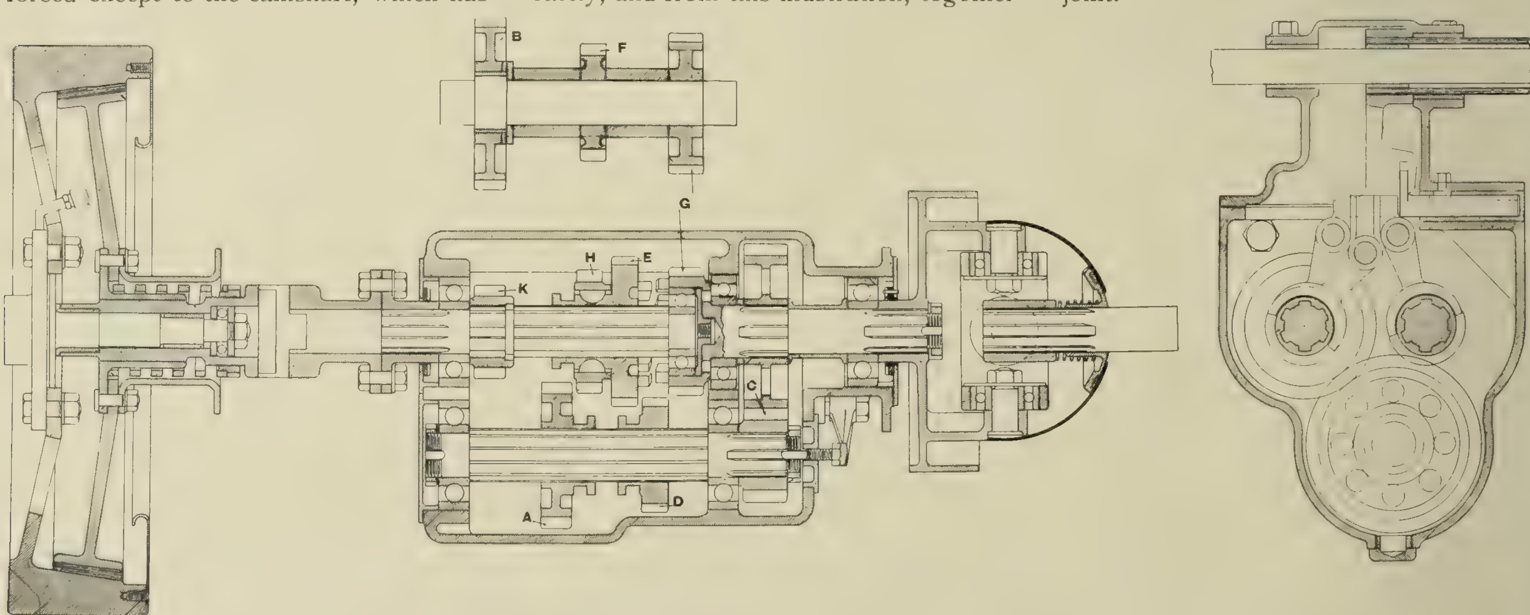


Fig. IV. 17 h.p. MAUDSLAY TRANSMISSION. Horizontal section and vertical section at centre.

Before proceeding to deal with the transmission, it may be mentioned that the standard carburettor is a White and Poppe, and that the radiator is of the honeycomb type, the circulation being maintained by convection alone.

The clutch is a leather-faced cone, and is almost fully explained by Fig. IV. It is perhaps surprising that this form of coupling should be considered in keeping with the rest of the chassis, as it is without question the least advanced portion of the whole; of course, it is quite effective, but it is impossible to neglect the fact that a metal to metal clutch can be made

in several patterns, to give greater ease in use and greater durability than any leather surfaced device. In the Maudslay clutch the durability of the leather and, to a certain extent, the ability to slip the clutch, is increased by running the leather in a bath of collan oil, and it is to retain a small quantity of this fluid that the cover-plate, seen in Fig. IV., is fitted to the flywheel. A small point that is often overlooked is the effective lubrication of the clutch spigot bearing, and the small grease cap fitted inside the clutch for this purpose is perhaps typical of the attention to detail throughout the chassis.

The gearbox is a particularly interesting design, as it provides four forward speeds and still has extremely short shafts. The highest speed is geared up, the third being the straight-through drive, and in order to understand the operation it must first be realised that the striking arm engages more than one of the strikers at the same time, except for the reverse movement when it engages with one only. Thus, movement of the first and second speed layshaft pinions is simultaneous, and on coming through the gate the first speed pinion is released, leaving the second speed striker still engaged. The second speed pinion then moves together with the third speed clutch, this being necessary in order to move it backwards when engaging the third speed. In Fig. IV. the second layshaft which lies below the other two is shown separately in order that the operation may be followed more readily. Starting with the reverse gear it will be seen that the drive passes from K to B, from B to A, and then through the pair C to the propeller shaft. The next motion of the control lever disengages the reverse and picks up the striking lever controlling pinion D. Forward motion then gives the first speed, and backward motion the second speed, using the final drive C as before. Passage across the gate does not release A, which moves idly, and backward motion will move the dog clutch into engagement, pushing pinion D still further backwards where it is out of the way. Next, forward motion, without coming back through the gate, brings wheels E and F into mesh and the drive is then transmitted through the pair G, giving the geared up fourth. The disadvantage of this form of construction is that the two layshafts are always running, and the reverse wheel B is also always running at a different speed to the layshaft which carries it. On the other hand, the shafts are disposed and proportioned excellently as regards elimination of sound, and in actual practice the gears operate with reasonable quietness, creating rather less noise than the average of their kind. The following are the particulars of the principal gears : A, 23 teeth 6 pitch ; pair C, 14 and 28 teeth respectively, 6 pitch ; D, 24 teeth, 8 pitch ; E, 32 teeth, 8 pitch ; F, 24 teeth, 8 pitch ; K 14 teeth, 6 pitch ; B, 28 teeth, 6 pitch ; the pair G, 27 teeth and 29 teeth respectively, 8 pitch.

Thus the ratios given by the gears alone are : 1st speed, .413 to 1 ; 2nd speed, .66 to 1 ; 4th speed, 1.43 to 1 ; and the reverse, .305 to 1.

The large size of the Hoffmann ball bearings is a commendable feature, especially so as regards the spigot bearing of the main shaft, and all the bearings are well supported. The machining of the recesses for the bearings of all three shafts is performed by the use of a series of cutters mounted in turn on a special set of bars having large bearings on the jig, so the accurate spacing of the shafts depends solely upon the maintenance of the profile of the cutters, not an unusually difficult job. In common with all other aluminium parts of the chassis, the box itself is cast by the Birmingham Aluminium Castings Company, and is an alloy almost entirely pure aluminium. The gears are all milled from Ubas, and the whole of the cutting and hardening is done in the Maudslay works, though the bevels for the final drive and differential are manufactured by Wrigleys. It will be noticed that none of the gears slide on, or are attached to, squared shafts, splining being used through the car, even for the sliding portion of the clutch. It may be mentioned that certain of the gear wheels are hardened before their section has assumed its final shape, the additional metal being left on as a preventative of warping. As a whole, the gearbox is an

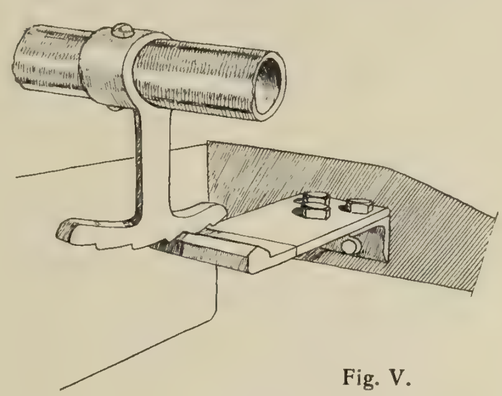


Fig. V.

expensive job, but its performance should be very satisfactory and, as we have said before, cost of production is not the principal factor in determining the design of this chassis.

The control gear is shown in Fig. IV., which also explains the locking mechanism employed to prevent the engagement of the idle strikers ; there is no departure from standard practice as regards the gate, and it is rather surprising that the old-fashioned sliding tube should still be employed, as the fitting of a freer mechanism is both cheap and easy. The locking of the gears when in engagement is performed by the device shown in Fig. V., and this is almost ideally simple and effective, the sound made by the trigger dropping into the slots being sufficient to guard against the danger of a gear being used when only half engaged.

Both the universal joints are of the ball bearing pattern shown in Fig. IV., the telescopic motion being very small, owing to the controlled motion of the back axle. In the June issue of "The Automobile Engineer," the advantages and disadvantages of the ball bearing as applied to universal joints, were considered fully, but it is worth remarking that the reason given for their employment in the Maudslay design is that their lubrication can be relied upon to be sufficient to give good durability, while the lubrication of plain pin joints is extremely difficult.

Fig. VIII. shows the radius and torque rods, which are arranged in duplicate on either side of the back axle, just inside the frame members, and it will be observed that the linkage of the parts gives an almost vertical motion to the axle, while relieving the springs of all stresses due to the drive. Shocks of clutch or brake application are partly absorbed by the spring buffers in the upper, or torque resisting, member, and the lower, or radius, member can be made to act as a road shock absorber by adjusting the pinching bolt. Allowance for unequal altitude of the two back wheels is provided at the forward ends of the rods, and is shown in the small sectional view. There can be no doubt that the flexibility of the springing is improved by the use of these controlling links, and it is probable that they have also a beneficial effect upon the life of the tyres and upon the steadiness of the car when being driven over a greasy surface. The amount of lubrication which the various joints require is quite small, and is ensured by suitable grease caps, not shown in the illustration.

The rear axle is a die-finished steel forging, made by Vickers, Sons and Maxim, and it is additionally interesting because the Maudslay company were amongst the first British makers to make use of a solid forged axle. The outside diameter of the sleeves is 2in., and the internal diameter $1\frac{5}{8}$in., while the driving shafts are $1\frac{1}{8}$in. The central ring is $2\frac{3}{4}$in. wide, and the section is $\frac{3}{4}$in. deep, the outside and inside diameters being 13ins. and $11\frac{1}{2}$ins. respectively. The bevel pinion and the differential are carried in a hemispherical aluminium casting, and are enclosed by an aluminium cover plate, both the aluminium parts being secured to the steel ring by the same studs, as shown in Fig. VI. Four inch Hoffmann single row journals are used to carry the differential case and crown bevel, and a ball thrust of corresponding size is also fitted, while the bevel pinion has a $3\frac{3}{4}$ins. main bearing and a $2\frac{1}{4}$ins. spigot bearing. All these ball bearings are housed in aluminium, but as their size is ample, and as they can all be secured rigidly, this is probably quite satisfactory. Only two pinions are used in the differential, and they are on opposite ends of a continuous steel shaft passing across the centre, there being a clearance of about $1\frac{1}{2}$in. between the inner ends of the driving shafts.

The hub mounting (Fig. VII.) is unusual as regards the amount of bearing given to each wheel. Three small races are used in preference to a smaller number of larger ones, to enable the hub diameter to be kept small, and it seems rather a pity that so many makers should allow appearance to exercise quite so much influence in this respect, for there is no doubt that a number of small bearings are not so durable as is one large bearing. However, the case is complicated by the desire to use Rudge-Whitworth detachable wheels and to make them interexchangeable for front and rear axles, and certainly the four bearings of the Maudslay hubs should give satisfactory service. There is no thrust bearing in either the rear or front hub, this being one of the notable omissions which we referred to in our introductory remarks. It is a point of incompleteness which this chassis shares with many others, but to regard the thrusts which the hub bearings have to resist as negligable is a mistake which will some day come to be realised.

The front axle is a Kirkstall, and the steering gear was described in our last issue, so there is no need to refer to it

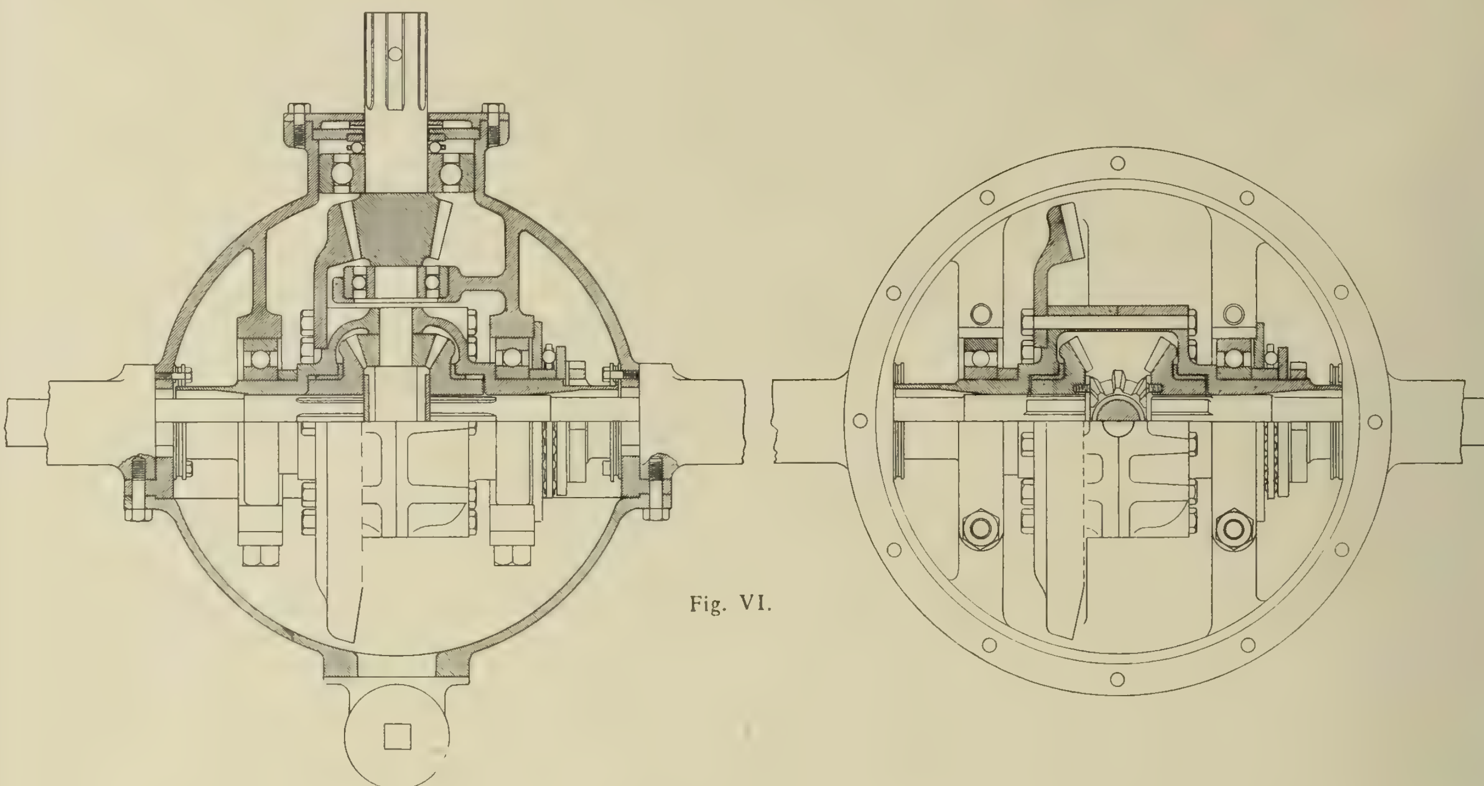


Fig. VI.

17 h.p. Maudslay rear axle, rear wheel hub and brake, and combined radius and torque rods giving parallel motion to the axle.

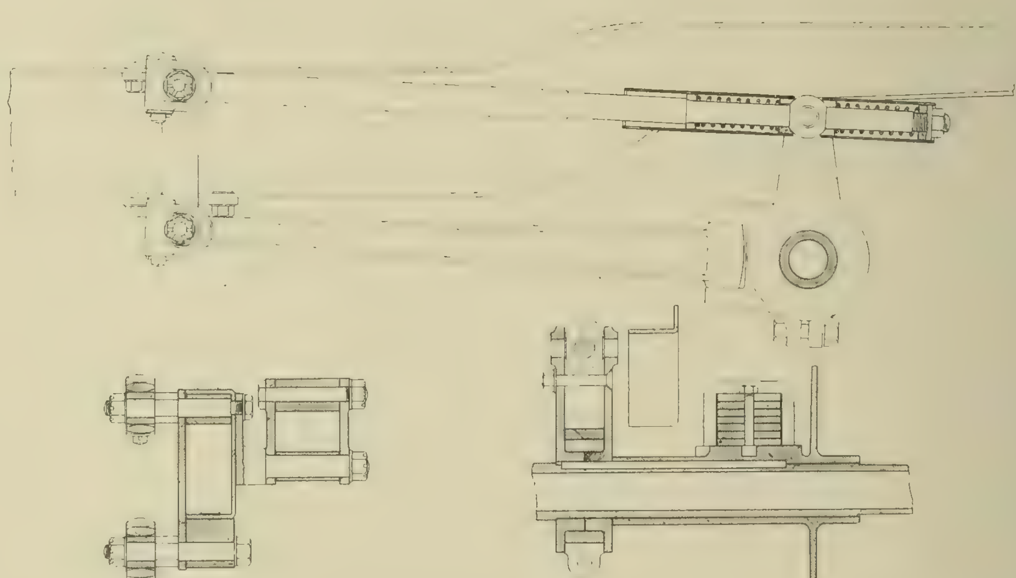


Fig. VIII.

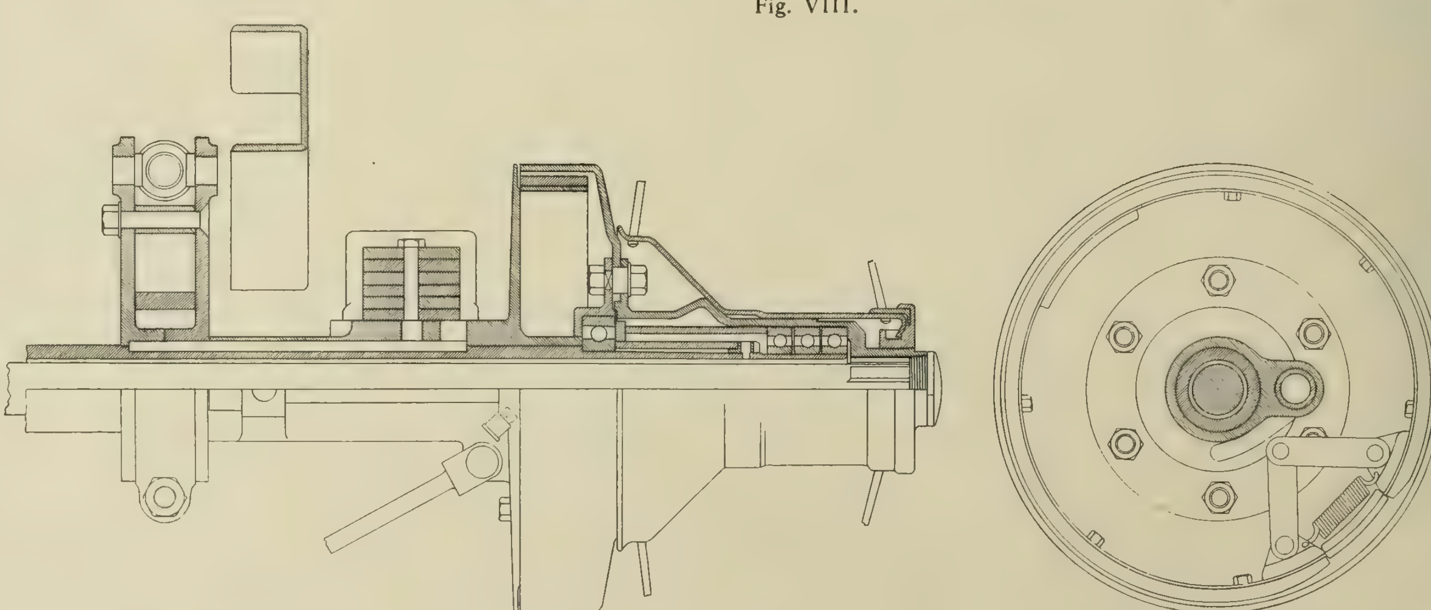


Fig. VII.

here. Still, the exceptional length of the main steering arm may be seen in Fig. I., which also discloses the fact that the connecting rod is quite straight and lies below the tie rod.

The foot brake is of the simple locomotive type with a hand adjustment, its main point of peculiarity being that the joints of the linkwork are all ball joints, it being considered that these can be lubricated more efficiently than pin joints, and can be adjusted to take up wear. The only large bearing on the brakework is that of the bell crank, and it is provided with a screw-down grease cap in a convenient position outside the frame. There is no compensating gear for the hand brakes, but the final connection between the cross-shaft levers and the brake shoes is made by wire cable. The reason for the use of this material is not very clear, but it appears to be principally because it is less liable to rattle. The actual shoe expansion is controlled by a linkwork which compensates between the two shoes, forcing each against the drum with equal pressure, so uneven wear does not cause one shoe to do more work than another, as both must engage equally (see Fig. VII.).

In actual operation, the chassis is distinctly above the average as regards smoothness of running, even at slow speed on the highest gear: the acceleration is not exceptional, but the general comfort is good and the handling is at least as easy as it is on most modern cars. In conclusion, we believe that the description will hold good for the smallest Maudslay chassis for the next year as well as at the present time, as no radical changes are anticipated by the makers.

LUBRICATION INDICATORS.

A review and criticism of standard practice.

By R. Waring Brown.

At the present time there is probably no subject in connection with the internal combustion engine, which claims the attention of designers so deservedly as lubrication, especially forced lubrication. A very necessary adjunct to lubrication systems, viz., the indicator, appears to have suffered some neglect, and this accessory merits considerably more thought than has been expended upon it heretofore, as is made obvious by studying current practice.

An indicator on the principle shown in Fig. I. is in most general use. The oil is pumped up through the inlet pipe, and drips on to the porcelain seat, whence it returns to the sump. The drip of the oil can be seen through the glass, and can be regulated. The disadvantages of this system are that the oil coats the glass, rendering it hard to see the drip, and it is not of much use for night driving unless fitted with a small electric lamp, which is objectionable for several reasons. The particular design illustrated has the advantage over the usual pattern of an easily detachable glass, for by unscrewing the gland nut the dome comes away at once; but as this is another point requiring the attention of the motorist, and

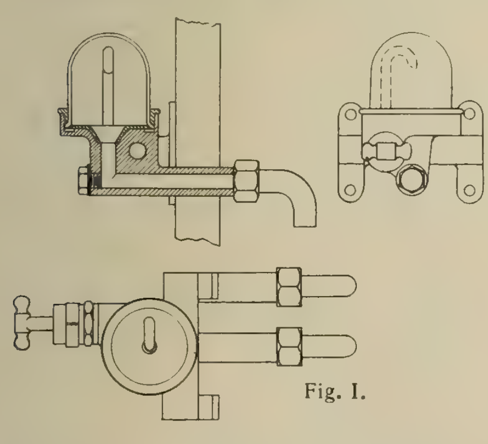


Fig. I.

is frequently overlooked, the advantage is not great.

Fig. II. shows another type of sight glass indicator, the body being screwed on the dashboard. The oil is pumped up to the inlet and the movement of oil through the indicator sets the wheels in motion. This indicates better than the type shown in Fig I., as the revolving wheels are clearly visible through the glass, but the disadvantages for night driving are the same as in the first model.

The ordinary type of pressure gauge is another very common type of indicator, and though of certain value to the engineer as recording the oil pressure, conveys practically nothing to the average driver, while sharing several of the disadvantages of the indicators shown in Figs. I. and II. Another point generally overlooked is the fact that as the viscosity of the oil varies so much when cold and

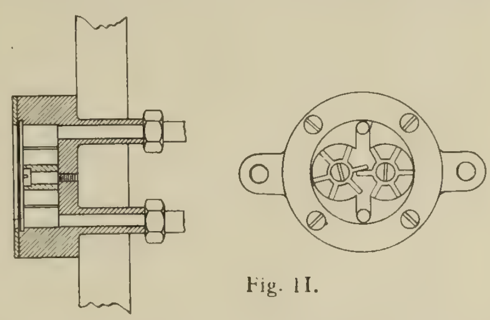


Fig. II.

hot, the indications mean practically nothing, and when the oil becomes warm the gauge more often than not shows no pressure at all if the bearings are slack.

Fig. III. shows another type of indicator which is largely used. The oil is pumped through the inlet, forcing the plunger against the pressure of the spring, so that it projects through the body. The oil returns through the second pipe. The whole thing makes a very neat instrument, and may be screwed on the dashboard. The plunger reciprocates slightly when oil is passing, and this is clearly visible to the driver. It possesses the advantage over previous types that, in night driving, it is easy to feel whether the oil is circulating or not by pushing down the plunger, when if it rises immediately the pump is known to be working; while if it does not rise, the circulation has failed.

Fig. IV. shows a type of indicator fitted to the engine of a well-known car, where it is mounted directly in the crank chamber. The oil is forced up through the inlet, pushes the piston upwards, and returns via the outlet pipe. Attached to the piston is a rod, which is joined, by a lever, to a spindle, on the other end of which is a pointer working over an indicating plate screwed on the dashboard. This is a very simple, and to a certain extent, effective indicator, suitable for a cheap and small car. It indicates the working of the oil circulation clearly, and shares the advantage of the indicator in Fig. III. for night driving.

The rather ingenious indicator shown in Fig. V. has proved in practice very successful. The oil is pumped up into the reservoir, forcing the leather diaphragm against the pressure of the spring. Attached to the diaphragm is a spring cup which is fixed to a rod, having a lever pinned on its upper end. This lever is pivoted to a rocking lever, which magnifies the slightest movement of the rod very considerably. On the end of the lever is screwed a ball working through the slotted indicator plate fixed on the dashboard. When the pressure of the oil forces the diaphragm upwards, it is arranged that it presses on the cover in order to prevent bursting of the leather. The one disadvantage of this system is that the oil in time soaks through the leather and prevents the arrangement working perfectly, wherefore the writer

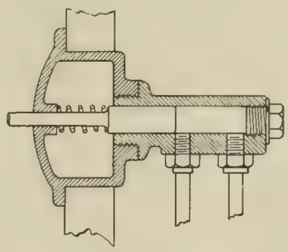


Fig. III.

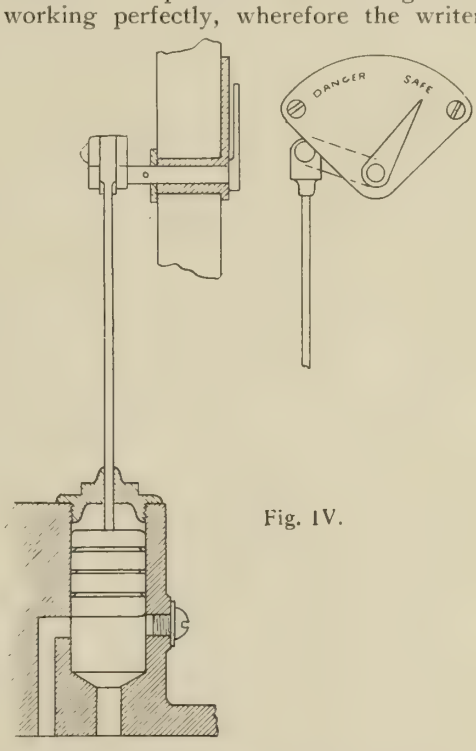


Fig. IV.

would suggest that it might be practicable to use a sheet steel diaphragm about 2/1,000 of an inch thick, and to exaggerate the movement of the rod still further. This indicator has the advantage of being

easily seen in the daytime, and easily felt at night.

The indicator shown in Fig. VI. is in all probability the best in various ways. It not only calls the attention of the driver to the fact if the lubrication is failing, but it does so audibly. In ordinary circumstances the circulation may have been arrested for some time before it is noticed, especially at night, when the attention of the driver is seriously monopolised by the difficulties of the road; but the warning given by the type illustrated in Fig. VI. would not be overlooked.

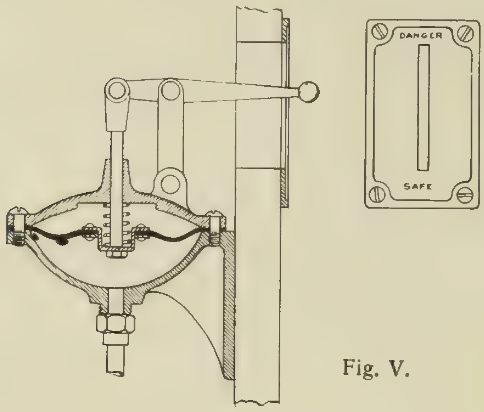


Fig. V.

The pressure of the oil is employed on the combined piston and rod, which is free to move against the pressure of the spring in the cylinder. When the oil is not circulating the piston rod is thrust to the bottom of the cylinder by the spring. Fixed to the piston rod is a metallic arm making contact with an insulated ring of metal. The piston and cylinder are in contact with one pole of an electric circuit, and the insulated piece is connected with the other pole of the same circuit which, when complete, rings a bell on the dashboard. When the oil is circulating it will lift the piston and the electric contact will be broken; consequently, the bell will not ring. If, however, the oil circulation should fail the spring will thrust the piston down until the arm touches the contact piece, so completing the circuit and causing the electric bell to signal the failure of circulation. This method is undoubtedly excellent, excepting that it is necessary to have a bell, battery, and some special wiring, and probably a separate switch. [It might be made to cut out the ignition, so stopping the engine.—Ed.]

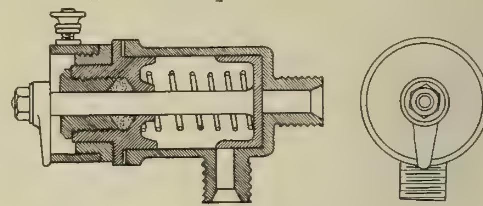


Fig. VI.

After giving these designs much careful consideration, it is evident that there is ample scope for an indicator which would call the attention of the driver at all times, day or night, to the state of the oil circulation. It needs to be simple, cheap to manufacture, free from liability to get out of order, and to require no complicated fitting. It is quite within the bounds of possibility that something could be designed to utilise exhaust pressure, such as a small ball whistle worked in conjunction with the usual piston indicator.

THE ILLUMINATION OF WORKSHOPS.

A consideration of the relative advantages and disadvantages of different systems of lighting, for both general and local purposes.

IN the increasing stringency of modern business competition it behoves us to be constantly examining our manufacturing equipments, and to see that every factor favourable to greater economy in production is taken advantage of. As the subjects of lighting, heating, and ventilating seem to receive, as a rule, rather less attention than their importance deserves, we propose to make a brief comparison of the different systems in common use.

That men must work to better advantage in a good light than in a bad one, and that their output of work will be better if they feel warm and comfortable than it will if they are cold and miserable is too obvious to dwell upon, and yet, going into many modern works at six o'clock of a midwinter morning one finds the temperature about freezing point, and the place in a condition of semi-darkness. There is no necessity to "coddle" men, but the idea that cold is a good incentive to them to work harder than usual in order to keep warm does not work out satisfactorily in practice.

We would not unduly labour the point, but it is well to remember that the very nature of a man's job may be such as to preclude the possibility of active motion; and, moreover, that our best mechanics may be men of inferior physique, who in very cold surroundings are perhaps only half as efficient as they would be if warm and comfortable. This loss of efficiency is a really serious item, and the expenditure necessary to remove the causes of it will be found a first-rate investment.

In dealing with lighting we have to consider it in two aspects, viz. (1) *general lighting* of the various shops and departments, and (2) what may be called *individual lighting* of machines and fitters' benches. The requirements of each are so different that they must be treated quite independently.

For general lighting powerful and well-diffused lights are wanted, preferably placed high up out of the way, and replacing as well as may be the general effect of daylight. The amount of light necessary in various shops varies, of course, with the kind of work carried on in each, but it is much better to err on the excess side than to give too little, and in all departments of a motor factory where fine exact work is a necessity liberal ideas should prevail.

We have made considerable enquiry to ascertain whether from existing installations any sort of empirical rules can be laid down which might be useful for guidance in planning new works, but interpretations of what constitutes "good lighting" vary so much that only an approximate idea can be given. One man says "put in plenty, and let us have really brilliant lighting," and his shops are a pleasure to go into when fully lit up. Another keeps economy in the forefront, and is satisfied with about half, and again a third seems to think that general lighting, as such, is quite needless. However, we can take as a guide one particular installation, of which we have full details, and which we should consider a good example of fairly efficient and satisfactory lighting. For a car assembling department the allowance was 100 candle power per 260 square feet of floor. Judging by the effect, we should call this fair. If the allowance were made 100 candle power for 200 square feet it would be good, but if the area were 300 square feet for the same light we should think it poor for such a class of work.

In the repairs' department, where a medium general light only is required, supplemented by individual lighting at the cars and vice benches, the allowance was 100 candle power to 500 square feet of floor area, which was found to be quite sufficient for the purpose.

It is always desirable to have stores well lighted, and in the case under review the allowance was 100 candle power for 350 square feet, with all lights on, but this gave rather more light than necessary, and two-thirds of the light, or, say 450 square feet per 100 candle power, might be considered fair.

But in taking these calculations into consideration it is necessary to bear in mind that the effect will vary considerably, according to the method of distribution and the size of the units. In the case just referred to the units consisted of 200 candle-power lamps evenly distributed, and their height above the floor was 10ft. to 12ft. If the units had been fewer and more powerful they would not give an equally useful effect with the same amount of candle power, and they would have to be placed higher.

Machine-shop lighting should always be on a liberal scale, but here we rather favour an arrangement which, although giving a general effect, is really a development of individual lighting, that is to say, we would use smaller units and group them over the machines. For lathes and machines of any size two lamps of 100 candle power will be found to give a very good effect. Smaller machines may do with one light, or with smaller units, if the system adopted permits of this.

For components' assembling, both general and individual lighting will be required, and if we adhere to the 200 candle-power units the general lighting should be satisfactory on a basis of 100 candle power per 300 square feet, if this is well supplemented by bench lights.

Tinsmiths' and blacksmiths' departments should be about right on a similar allowance.

Pattern makers should be liberally supplied, and we would say it were wise to allow them 100 candle power per 100 square feet, and for bodymakers, say 100 candle power per 200 square feet. In the paint shop good light is wanted over the

area where work is going on, and if this is in a fixed situation to which all work is brought, it would be well to put the units more closely, but arranged so that only those need be burning that are required to suit the work in hand. It should be particularly noted that the above are suggestions only, and are not to be taken as hard-and-fast rules. They may, however, be of use as giving something definite to start with, something that has proved satisfactory in actual practice, in laying out a new installation. For this purpose, therefore, we tabulate the information to make it easier for reference.

Approximate allowances for general lighting of motor shops with 200 candle-power units evenly distributed :—

Machine shop, about 100 to 200 c.p. each machine.
Tool room, about 100 c.p. per 300 square feet.
Components' assembling, about 100 c.p. per 300 square feet.
Chassis assembling, about 100 c.p. per 250 square feet.
Repairs' department, about 100 c.p. per 700 square feet.
Tinsmiths and blacksmiths, about 100 c.p. per 300 square feet.
Pattern makers, about 100 c.p. per 100 square feet.
Body makers, about 100 c.p. per 200 square feet.
Finished stores, about 100 c.p. per 400 square feet.

These figures assume that shades will be used to throw all the light downwards, and that small units are adopted.

It must also be borne in mind in regard to the foregoing table that it may require modification according to the system of lighting in use. For instance, we would not probably use such small units if arc lamps were adopted. We may now consider which of the available systems of lighting it is best to employ, and they may be divided, broadly, into two, viz., gas and electricity. These again may be sub-divided as follows :—

Gas.—Coal or town gas.
Acetylene gas.
Petrol vapour or "air" gas.
Electricity.—Arc lamps.
Incandescent lamps.
Mercury vapour lamps.

In reviewing the various points in favour of or against these, we have to keep in view the capital cost of installation, cost of working and maintenance (which is far more important), and general convenience.

Town Gas.

Coal gas is too well known to require description. It is undoubtedly the most widely-used illuminant in this country at the present day, and although years ago its immediate demise was confidently predicted as a result of the introduction of electric light, the very reverse has occurred, and it is to-day in a stronger position than ever.

It is hardly needful to say that this fact is due to the invention of the Welsbach incandescent mantle and burner. Previous to that, whilst we had only the flat flame and Argand burners, gas lighting left a good deal to be desired, as it was neither clean, economical, nor over safe. The flat flame burner, in fact, was lamentably inefficient as a converter of gas into light, so the demand for gas, although considerable, was nothing compared with what it has since become. In order to appreciate in its true proportions what the incandescent mantle has done for gas, it is necessary to keep in mind the basis figures on which comparisons may be built. "Standard gas" may be said to be 16 candle power for one hour from five cubic feet of gas. This is not absolutely universal, and in some towns higher candle power is made, but 16 candle-power gas is the average standard.

In 1,000 cubic feet, therefore, we get 16 candle power for 200 hours, or 3,200 candle-power hours. If the cost is 2s. 6d. per 1,000, we have 106.6 candle-power hours for one penny.

The Welsbach incandescent burner gives 60 candle power on a consumption of three and three-quarters cubic feet per hour, or, to be on the safe side, let us say four cubic feet per hour; therefore we get, in round figures, 60 candle power for 250 hours from 1,000 feet, or 15,000 candle-power hours, or 500 candle-power hours for one penny. The incandescent mantle is therefore five times as efficient as the flat flame. Under the circumstances it is not to be wondered at that it created a revolution in the gas world. Instead of being wiped out gas lighting was almost immediately placed in the front, and has well maintained its lead ever since.

Just at first, owing to its fragile nature and somewhat high cost, the incandescent mantle did not make much headway for factory lighting. Vibration was fatal to it, but this difficulty was soon overcome by spring-suspended burners and stronger mantles. Improvements for the special purpose of factory lighting continued, following the line of larger and mechanically stronger mantles and increased pressure of gas at the burner. The outcome of this is seen in the modern "self-intensifying" lamps, such as the "Lucas," and "pressure-gas" systems, of which the "Keith light" and Glover's system with Onslow's patents are good examples. The results attained with pressure gas mark an advance in incandescent lighting that is even more extraordinary than was the original Welsbach burner in comparison with the flat flame, as may be understood when it is said that with self-intensifying burners we get 30 candle power per cubic foot, or 1,000 candle power hours for 1d., whilst in the latest type of Keith light, with inverted burners, an efficiency is claimed of 60 candle power per cubic foot of gas, equivalent to 2,000 candle-power hours for 1d. with gas at 2s. 6d. This is truly a marvellous result, and should provide food for reflection to those who are content to have their men groping about in semi-darkness using a few flat-flame burners, which are probably costing ten or twenty times as much as would a brilliant and cheerful light!

It is stated on the authority of the Chief Engineer of the Edinburgh and Leith Gas Commissioners, and published in the "Journal of Gas Lighting" and "The Gas World," that tests of 1908 inverted type Keith burners showed an efficiency of 73.6 candles per cubic foot of gas, which at the 2s. 6d. rate gives 2,453 candle-power hours for 1d., so it is evident that the claim of 2,000 candle-power hours is not extravagant.

In "self-intensifying" and "pressure" systems the object, as indicated by the titles, is to supply the mixture of gas and air to the mantles at a higher pressure than is permissable with ordinary town mains' pressure. In the first this is attained by the inductive effect of a long chimney and suitably-shaped burner, whilst in the case of the Keith light the gas pressure is "boosted" up by means of a suitable blower.

Both systems have proved most successful for shop lighting, being easy to install, and requiring a minimum of attention. It is within the present writer's knowledge that more than one electric arc lamp installation has been pulled out and replaced by pressure gas with results of the most satisfactory nature to the owners.

Acetylene.

Although we are not aware that the acetylene system of lighting has yet been much adopted for factory work, and especially for motor factories, there can be no doubt that great improvements have been made in regard to the appliances used in its manufacture, and as now it may be said that the experimental stage has been passed, there seems no reason to ignore its claims. Carbide of calcium is now available at a fairly reasonable price and without any difficulty, and it is practically free from danger in storage.

The claims made for acetylene may be just touched upon. In regard to ease of installation it compares quite on equal terms with any other system. It is its own inspector as regards defects, for its pungent odour calls attention to leakages.

The quality of the light is such that in it colours have almost their natural value, and it is remarkably free from destructive products of combustion which tend to tarnish and spoil articles kept near it. It is claimed that, when properly made and purified, it burns without smell, and contaminates the air less than oil lamps or coal gas. It is claimed also to be many times more brilliant than coal gas, or in other words the light is more intense, but this is a doubtful advantage for the purpose under review. We do not want our light too much concentrated. Finally, as regards cost, it is calculated that 200 lbs. of carbide will produce 1,000 cubic ft. of gas with an illuminating power of 60,000 candle power hours at a cost of 19s. 3d. for the carbide, which is equivalent to 260 candle power hours for 1d.

These figures are taken from a publication which may be expected to make the best case it can for acetylene, and it is evident that in economy it cannot hold its own with the best systems of gas lighting.

In the book referred to the statement is made that to produce 60,000 candle power from coal gas, 9,500 cubic feet of the latter would be required, and that the cost would be 28s. 6d. But the figures we have given show that with only the ordinary domestic mantles we can get 15,000 c.p. per 1,000 feet of coal gas, so only 4,000 feet, costing, say 10s. to 13s., are required.

Evidently we shall have to wait till carbide becomes very considerably cheaper before—other things being equal—we can seriously consider acetylene as a candidate for shop lighting, except in isolated instances which, like its use as a private house illuminant, do not come within the scope of this article.

Air Gas.

Petrol Vapour or "air" gas, is simply air carburetted in a special type of petrol vaporizer and converted into light by means of special burners and incandescent mantles. Although at a first glance the system seems to be the essence of simplicity, it has really proved a most complex and difficult problem, in the solution of which many thousands of pounds have been expended.

The gas produced by this system consists of air to the extent of about 98½ per cent. and hydro-carbon vapour 1½ per cent. It can only be burned in a special type of burner, and has of itself no illuminating power at all; but it burns with an extremely hot flame, and when used with an incandescent mantle produces an intense light of particularly good quality. The apparatus required for its production is not unduly complicated, and can be looked after by any ordinary workman. Distribution is by ordinary gas pipes, but it is desirable that these should be somewhat larger than usual. There is no condensation, so that long leads of pipe are not objectionable. Any size of unit light can be adopted, from 25 candle power up to 2,500 candle power, with single burners, which is fully sufficient for our purpose. The quality of the light is of very pure and excellent colour, free from green and ultra-violet rays.

One of the disadvantages of coal gas is the fact that it is not unvarying in quality and light-giving power. In this, air gas is claimed to be greatly superior, as there is no loss of the raw material used in its manufacture; the whole of the heat units go to the burner, and are consumed, and there are no non-useful (so far as light is concerned) constituents to get rid of such as naptha, tar, etc., as is the case with coal gas. It results that in regard to air gas it is possible to give it a candle power value which remains true throughout the country, for any slight variation in the quality of petrol used has no appreciable effect.

As regards the cost of air gas, it will be remembered that petrol used for this purpose is not subject to any duty. The National Air Gas Co. state that their plants are now constructed to use .760 petrol, at as little as 6½d. per gallon. From each gallon of this spirit there is produced 1,600 cub. ft. of gas, with an efficiency of 16,000 candle power hours or 2,461 candle power hours for 1d. These figures place this system in the very front rank, and indeed, assuming their correctness, it is questionable whether any other system can come up to it.

Of course, the initial capital outlay for the plant has to be faced, including burners and fittings, but the saving effected would soon wipe out this expenditure. Already a number of plants are at work in engineering shops, so that it is possible to see for oneself the effect, and get actual figures as to cost.

Electricity.

For general lighting by electricity we have a choice of three systems, viz., the arc lamp, the metallic filament incandescent lamp, and the mercury vapour lamp.

One of the chief attractions of electric lighting is the ease with which it can be installed. The cables being so flexible lend themselves to easy distribution in places where piping might present considerable difficulty, whilst their small size is an added advantage. It is hardly necessary to refer to the technicalities of wiring, beyond pointing out the necessity for the most careful work and for the use of only good quality cable. All electric light wires are covered with an insulating sheath, consisting of a layer of pure rubber next the wire and on top of that vulcanised rubber to a greater thickness, varying according to the size of cable; over this, again, is wound adhesive tape, and finally on top of all is a braided cover of cotton, the whole being soaked after completion in a bituminous compound. Owing to the high price of rubber many makers—principally foreign—reduced the thickness of the pure rubber and adulterated the vulcanised portion to such an extent that very imperfect insulation resulted, and after a very short life such inferior covering was found to perish and afford very unreliable protection against short circuits. Many disastrous fires have without doubt been due to this cause. It is, however, easy to make sure of the right quality by specifying that all cable must be made by one or other of the firms who form the English Cable Makers' Association, or, as it is usually described, "ring" cable.

Until comparatively recently the arc lamp provided by far the most efficient method of converting electric current into light, and its high candle power made it very suitable for schemes of general lighting. The candle power of arc lamps presents rather a difficulty, and it will be found that in the majority of catalogues the term is conspicuous by its absence. In early days it was used, but led to trouble because usually "nominal" candle power was specified, viz., 1,000 or 2,000, but in actual working nothing like this was given off. The fact is that the power of an arc lamp varies so much at different angles and is so largely affected by the different kinds of globes used, that for long it has been the custom to designate lamps by their current consumption, e.g., "5 ampere" or "10 ampere" lamps, and to ignore candle power entirely To those in the trade this did not cause any inconvenience, for they soon became accustomed to the conditions and knew just what would answer in any given circumstances, but for the user it is rather unsatisfactory, as it deprives him of the means of knowing whether he is getting value for his money, i.e., an adequate return in candle power for the current he pays for. He wants to know that too large a proportion of the current is not being wasted in the resistances (which form an essential adjunct of all arc lamps) instead of being converted into light.

The chief difficulty in regard to the candle power of arc lamps is in the varying intensity of the light at different angles, for measured at an angle of 15 to 20 degrees from the horizontal it may be as much as 4,000, whilst if taken vertically underneath it may not be a quarter of this. However, some of the leading makers are now publishing figures for comparison which deal with *mean* hemispherical candle power, and it would seem likely that in time all makers must do so if they are to keep in the race. We should therefore have a reliable basis of comparison as between one arc lamp and another. We had hoped to be able to institute a comparison of cost as between arc lamps and gas, on a candle power basis, but owing to the impossibility of getting figures which are comparable have come to the conclusion that any such comparison would not serve a useful purpose. In such tests as we have seen the candle powers of the two lights are taken in a different way and do not seem to compare on even terms. Arc lamp makers claim and bring forward a formidable array of figures to prove that nothing can approach their system for efficiency and economy, but the gas people claim to prove that nothing can touch their results.

We do not believe that, with the best types of arc lamps, there is a great difference in the cost of electric current as compared with cost of gas (taking current at 3d. per unit and gas at 2s. 6d. per thousand) for a given effect, and it will generally be found that other conditions have to be taken into consideration in deciding which system to adopt.

The arc lamp is really in a class by itself, and there can be no doubt that in its latest types, using carbons which are impregnated with certain chemical earths or salts and embodying all the most valuable results of long years of improvements, it stands out for brilliancy and efficiency as one of the best forms of modern illumination. Its only real competitor is the pressure gas lamp. It always has been the most efficient method of converting electric current into light, even in earlier days when its efficiency was not comparable with what we have nowadays. The chief disadvantages are that arc lamps require a good deal of more or less skilled attention, owing to the comparatively short life of the carbons and the fact that they contain a considerable number of working parts that are liable to get out of order if neglected; also the cost of the carbons is a not inconsiderable item when reckoned on a year's consumption.

The burning life of a pair of carbons varies considerably with different makes of lamps. 17 or 18 hours is a fair example, but the Jandus Company claim a life of 70 hours with their 5 ampere lamp, and 60 hours in the 8 ampere size.

Obviously the longer a lamp will run without attention the more favourably is it likely to be considered by prospective users, but there are many other points to take into consideration, of which may be mentioned simplicity and reliability of mechanism, ease of access for re-carboning and cleaning, efficiency of the means provided for dealing with the fumes produced by the chemically treated carbons, and—not least—economy in current consumption and carbons.

The flame arc lamp, which is the modern type using chemically impregnated carbons, is made in two principal types, viz., one in which the carbons are arranged vertically and the other having them arranged side by side with the points converging at the lower ends where the arc is formed. Of the former type the Jandus lamp is an excellent example, and of the latter the Excello is one of the best known and most successful. In the Jandus the lower, positive carbon is short and thick, 7¾in. by 7-8in., and is of star section, the grooves between the eight rays being filled in with chemicals applied in the form of a paste and

baked. The upper carbon is 18in. long by ⅝in. diam., and slides in a central tube, feeding downwards as it is consumed.

The carbons are enclosed in an inner glass cylinder and an outer globe, the inner cylinder communicating freely at the top with two metal tubes leading to its base. The heated gases and chemical vapours from the arc rise through the cylinder, pass down the tubes and re-ascend past the arc, intensifying it, and this cycle of operations is continuous, the chemicals being used over and over again. About 15 grains of composition is volatilized hourly. No fumes can escape from the lamp, a point of considerable importance if it is to be used in any enclosed room where ventilation is not good, as such fumes smell unpleasantly. The working parts of this lamp are very simple and solid in construction, and are protected from the fumes. The efficiency claimed is very high, a mean hemispherical candle power of 2,800 being obtained with a current of 6.6 amps., with a voltage at the arc of 70, giving an efficiency of .166 watts per mean hemispherical candle power, which with current at 3d. per unit represents 2,008 mean hemispherical candle power hours for 1d.

The Excello lamp of the Union Electric Co., is made both with superposed and side by side inclined carbons, but the latter is the type most in demand. The carbons for direct current lamps are in pairs, 15⅜in. or 23½in. long (for different sized lamps) which give approximately 10 hours and 17 hours burning respectively. The points project downwards through an inverted saucer-shaped reflector of rather deep section and of highly refractory material, called the economiser reflector. When the lamp is burning properly the arc is formed inside this cavity, which becomes, of course, highly incandescent. There is an outer enclosing globe with a ventilating inlet plug which also forms a tray to catch the ash, and an inner open-bottomed globe reaching well down into the outer one and leaving an enclosed air space between the two. Air enters through the ash tray, is slightly heated in the space between the globes, and rising through the inner globe to the arc, is charged with fumes and then deflected sideways towards the inner surface of the inner globe, which is at a considerable temperature due to its proximity to the arc. Being further heated the current of air passes upwards and escapes through an annular space around the lamp casing. The effect of this action is to carry away the fumes so that they have no chance of depositing on the globes, and this freedom from deposit is of no little moment, because experiments have shown that from 30 per cent. to 40 per cent. of the light of a lamp could be lost owing to deposits on the globes. In order to give a more uniform distribution of the light, the Excello inner globe is made dioptric, which has the effect of modifying the polar curve of the light and greatly improves the lighting effect. Regarding the efficiency of this lamp, burning two in series on a 110 volt direct current circuit and using yellow flame carbons, the consumption was found to be from .24 to .17 watts per hemispherical candle power, including all line resistance losses. This gives practically the same result as the Jandus lamp quoted above.

Thus we can compare one arc lamp with another as regards candle power efficiency, but obviously that is not by any means the only, or, indeed, the principal factor to be considered in choosing a lamp. Owing to the number of highly technical points of detail involved, it is impossible in an article like this to attempt to deal with them; it will generally be found necessary to take expert advice. As already stated, it has not been found possible from the data available to make a trustworthy comparison between arc lighting and gas lighting, but from the figures just quoted it can be seen that arc lamp efficiency is very high, and for general lighting choice is between that and pressure gas.

The invention of what is called the metallic filament lamp, of which the "Osram" is perhaps the best known, has done almost as much for the general advancement of domestic electric lighting as did the Welsbach mantle for gas lighting. The carbon filament bulb required 3½ to 4 watts per candle power, but the metallic filament lamp only takes 1¼ watts. That gives—with current at 3d. per unit—80 candle power hours for 1d. with the carbon lamp, and for the metallic lamp 266 candle power hours for 1d. This represents a very large saving, no doubt, but for the particular purpose of general factory lighting it is evident that in point of economy even the metal filament lamp is a long way behind. When, therefore, the statement is made, as it now constantly is, that electric light at 4d. per unit with metal filament lamps is as cheap as gas at 2s. 6d. per thousand, it is clear that the comparison is only relative, and whilst it would be approximately correct for domestic lighting, if the indirect as well as the direct savings were taken into account, it would be very far from the truth as applied to works lighting. Even compared with arc lighting the cost is four times too high at least, so although it requires a minimum of attention, and the light is of a very pleasing quality, incandescent general lighting can only be considered an expensive means of illumination. The filaments, though much improved, are still very fragile and easily broken, which is rather a serious drawback seeing that the lamps cost from 3s. 9d. to 6s. and 7s. each.

The Mercury Vapour Lamp consists of a long glass tube about an inch in diam., with a bulb at one end containing a small quantity of mercury, and provided at both ends with electric connections sealed in The tube is suspended at a slight angle from the horizontal, and when in operation gives off a peculiar light of considerable power. For factory use the tubes are about 45in. long, used singly for voltages of 100-120, or two in series for 200-220. One such tube requires 3½ amperes at 110 volts and gives 700 candle power (mean), so the current consumption is .55 watts per candle power. Taking cost per unit, as before, at 3d., we thus get 600 candle power hours for 1d., which is not extravagant. The lamp is mechanically strong, easy to install and requires very little attention, but in starting it it is necessary to pull down the high end until the mercury runs to that end, and then let it up again slowly. Sometimes the men are careless in doing this, so that breakages result, which with renewals at about £2 each, is rather a serious matter.

The chief advantage of this lamp is that it gives a very uniformly distributed light, but it has a counterbalancing disadvantage in the peculiar quality of the light which is of a greenish tinge owing to absence of red rays. Faces seen in this light have a particularly ghastly appearance, and it is not at all cheerful, but the light is very steady and is said to be easy on the eyes, so that for machine shops, when supplemented by a certain amount of individual lighting, it is rather highly spoken of.

Even when general lighting is on a liberal scale, it is practically impossible to eliminate individual lighting, by which is meant the provision of comparatively small units for quite local purposes at the various machines and fitting benches. Such lights should be easily portable, and arranged with flexible connections so that they can be fixed in any desired position to suit the work, and they should be mechanically strong, as they are likely to get a good deal of knocking about. Incandescent gas mantles and metallic filament electric lamps are unsuitable by reason of their fragility, so that we really have only a choice of two systems, the flat flame gas burner and the carbon filament electric lamp. The latter is the most convenient and hardly inferior to the gas in point of economy. The carbon lamp is strong and cheap, the light is very steady and can be used in any position, while by means of reflectors it can be concentrated where required. By means of what is called a magnetic holder the electric lamp can be "stuck on" to any convenient piece of metal, and will support itself there as long as it is in operation. This is a great convenience in chassis erecting or repair work, and it is surprising that the magnetic holder has not been more largely taken advantage of. If used with a counterweight pulley it will be found useful in lathe work, too, but it is not desirable for use in such a situation at the end of a long loose flexible wire, owing to the danger of the latter being caught in the revolving parts.

Fixed at the end of a double or triple jointed bracket, a flat flame gas-light can be used for local purposes, but in the matter of convenience it is not to be compared with the electric bulb. It is unsteady, smoky when used in anything but an upright position, and cannot conveniently be concentrated on to a particular spot, but rather has to be kept away owing to the risk of burning the operator's face or hair.

In regard to cost, taking current at 3d. per unit and gas at 2s. 6d. per thousand, we get respectively 80 candle power hours electric, and 106 hours gas light for 1d., but the probability is that there will be so much waste of gas that the actual results will hardly differ, and in any case the extra convenience of the electric light would warrant the increase.

In conclusion, although the figures we have taken in ascertaining the relative costs of various systems are by no means universal throughout the country, they are a sufficiently good average for purposes of comparison. In large towns gas is probably cheaper than 2s. 6d. per thousand feet, but may be as high as 3s. 6d. On the other hand electric current at 3d. is probably a little under the average, though large consumers should not pay more for it.

For general lighting pressure gas or arc lamp systems both give highly economical results, which may be said to be approximately 2,000 candle power hours for 1d. No other system of lighting except the air gas seems to come anywhere near this, and most of the others are so very far behind that their use must mean a serious loss of money.

Pressure gas has the strong recommendation that it requires very little attention, and mantle renewals are found to be neither expensive nor very frequent.

For individual or local lighting the carbon filament incandescent bulb is probably best all round, and where portability is desired the use of a magnetic holder will much increase its usefulness.

CONTEMPORARY KNOWLEDGE.

Various ways and means for gaining useful experience.

THE question as to whether designers and senior draughtsmen should be allowed the frequent use of cars is one which has caused much wrangling between employers and employees, and seems still to be far from settled. Some manufacturers consider that the ability to drive a car is quite unnecessary, and the argument is often put forward that as locomotive designers are not in the habit of spending odd half days in an engine cab, there is similarly no need for their inventive staff to waste time on the road. On the other hand, there is no doubt that much time has been squandered by many staffs in driving for their own enjoyment when they ought to have been sticking closely to their indoor work, and seeing the differences of opinion that exist, it is worth while to endeavour to sum up the advantages and disadvantages of both methods.

First of all the designer is employed principally to improve designs already in existence and to evolve new designs from time to time, bearing some sort of relation to the old ones. It is but rarely that a man is needed to begin *de novo* with a free hand to do as he likes. Thus improvement, from either the users or the works point of view, is the chief business of the drawing office staff, and the improvements taken singly are usually quite small affairs.

The next step is to discover how the ideas for improvements arise, and here it becomes necessary to distinguish between "users' improvements" and manufacturing improvements. As regards the latter it is (or ought to be) the business of the works manager to report all extra costly jobs with a view to eliminating them if possible, and it is the duty of the drawing office to respect these criticisms and to endeavour to make alterations which will reduce the cost of production without affecting the usefulness of the altered part. If this plan is followed it is not necessary for a draughtsman to concern himself much with the exact workshop processes, outside the general knowledge which is possessed by all engineers who are worthy of the name, but there is seldom anyone amongst the testing staff who is in a position to give advice as good, in its way, as the machine shop comments of the works manager.

As a rule the head of a testing department is a man without sufficient general engineering knowledge to enable him to offer deep criticisms, and in an incredible number of instances he is filled with the idea that it is derogatory to his own reputation to complain of a fierce clutch, an awkward change-speed mechanism, or a control arranged inconveniently. In the rare instances where the chief tester is a really intelligent and helpful man it is usually found that he soon tires of the work and finds something better to do.

The case may perhaps be put more convincingly in another way. If improvements as regards production were dependent upon the suggestions and complaints of the various section foremen, made direct to the chief draughtsman, progress would not be so rapid as it is when those complaints are taken to the works manager, who can look into them himself and arrive at conclusions from his own observation. Being a much more highly trained engineer than the average foreman he is likely to be able to see more deeply, and can so tell the drawing office exactly what is required.

In just the same way an engineer is needed to experiment with not just an occasional chassis, nor every chassis that is turned out, but with a sufficient number to show clearly what the average is like. In a very large establishment it is of course possible to employ a fairly highly trained man who may have sole charge of testing and be accountable to no one less than the general manager, but in smaller works it is impossible to pay a sufficient salary to retain a satisfactory man, and the duty must then fall upon some other member of the staff. Who this other person should be must always depend to some extent upon circumstances, but there are several good reasons in favour of the choice falling upon a designer, the main one being that the designer having a creative mind is likely to think of useful additions as well as of alterations, and can see how to carry his ideas into effect without the necessity for explaining them at length to some other individual.

Also another thing which ought not to be forgotten is that there are advantages in allowing more than one man to make observation on the road, because the greater the number of opinions the less is personal prejudice liable to colour the result. For this reason it is scarcely advisable to depend upon one man alone, even if he is a specialist at the work with nothing else to do, and statement of this fact leads to the consideration of a quite different aspect of the case.

If it be agreed that the chief work of a designer is to make improvements, and that the better the knowledge of his own car possessed by a designer the more rapid is improvement likely to be, there still remains the difficulty of making comparison between the chassis made by one firm and those turned out by others. If it is found that a car is lacking in some particular respect a knowledge of other cars will often assist the observer to suggest means for removing the defect in the most efficient manner, and it may often happen that a designer's acquaintance with some more or less obscure details of another chassis will save much expenditure of time and money upon experiments.

In the June issue of "The Automobile Engineer," the Editor of "The Autocar" remarked upon the lamentable lack of all-round knowledge on the part of manufacturers and their staffs generally, and doubtless this will decrease somewhat now that automobile engineers have an organ devoted entirely to their peculiar interests. Still, something more than this even is really needed, for it is not enough to merely study even the most minute description, as it is not possible to convey the sensations of the performance and handling of a car in black and white, or indeed otherwise than by actual demonstration. A very few manufacturers have gone to the extent of buying a car of some foreign make for the purpose of making exhaustive tests with it, and in each case, within the writer's knowledge, the effect upon the next season's designs has been most marked; but the purchase of a single car is not enough, for it only gives the ideas of the one firm who produce it. There ought to be some employee of every moderately large motor manufacturing concern who has practical experience of at least a dozen of the best cars, and that familiarity ought to be maintained and not be left to become out of date, yet to buy twelve chassis every year from competitors would be very expensive as well as distasteful, and it might often happen that the knowledge gained would not be worth by any means as much as it cost.

As far as British cars are concerned, the knowledge possessed by one maker concerning the others is usually much greater than his corresponding knowledge of foreign practice, so the cars which ought to be bought (if any) would be of foreign origin. This being so, it seems reasonable to suggest that a number of manufacturers should agree to each purchase one good foreign car per year, and that the car should be circulated amongst the parties to the agreement. This would give the same advantages as the purchase of a number of chassis at a much smaller cost, and it would not mean that a maker was spending money to assist his competitors, because if there were twelve firms in the agreement each one of them would receive eleven experiences in exchange for giving one. Could not the British Manufacturers' section of the Society of Motor Manufacturers and Traders do something towards the furtherance of some such scheme?

The writer does not claim originality for this suggestion as it must have occurred to many dfferent people, but it never seems to have been considered seriously by the men who have the power to decide whether it shall be tried or not. A fact which ought always to be remembered is that it is impossible for an employee of one firm to keep in touch with the practice of others, for however extensive his experience may have been, it is all in the past tense, and the longer the time he remains in the employment of any one concern the

less valuable does his experience become, because it is more and more out of date every day.

Much is to be learnt by intelligent study of published accounts of R.A.C. tests, when they include such matters as fuel consumption, oil consumption, acceleration, maximum speed, etc., because they enable comparison to be made with any other car with known capabilities; but the results of such tests are misleading, because they do not represent a true average, but are rather examples of what can be done by a particular manufacturer when he is doing his utmost, and they are therefore much less instructive than tests which are conducted privately.

As a matter of fact, information ought to be gathered in every way possible, and the writer was once very much interested to find that the designer of one of the most successful cars of to-day had, for years, made a practice of keeping scrap-books of cuttings, from many different journals, arranged in classes and forming books of reference where records of tests, to compare with private observations, could be found, or where search might be made for a suggestive idea when some alteration became necessary or advisable.

MACHINING ENGINE CRANKSHAFTS.

A comparison of two modern methods.

The machining up of multiple throw crankshafts for automobile engines is a subject on which both English and foreign machine toolmakers have spent a fair amount of time during the last few years, with the object of designing tools and appliances for the production of accurate crankshafts in the least possible time, and the success attained in this direction may be gauged from the fact that a crankshaft which, by the ordinary methods of a few years ago, would have taken from forty to fifty hours to turn, can now be thus far completed in about one-tenth of that time; and that without any greater effort on the part of the workman concerned, and with perhaps considerably less risk in spoiling the work than by former methods. Crank turning on multiple throw crankshafts called for a very high degree of skill on the part of the workman, due to having all the measurements and dimensions to gauge and check by hand, whereas modern machine tools designed specially for this class of work have adjustable stops, with micrometer adjustments in some cases, and automatic trip motions which will control the cutting tools automatically, independent of the attention of the operator. This, of course, makes the machines semi-automatic in their action, and the trip motions are a safeguard against spoiled work, to a certain extent, by not being entirely dependent upon the constant attention of the workman.

As the output of engines by the majority of leading English and foreign automobile manufacturers has now reached very large proportions, several firms have found it a good investment to instal special machine tools and appliances for dealing with their crankshafts, whilst the smaller firms, whose comparatively small output does not warrant them going to the expense of laying down special plant, very often buy their crankshafts finished complete by jobbing crankmakers who forge and machine up for the trade, and are generally in a position to supply a cheaper and more accurately-finished product than the small manufacturer. It will, of course, be understood that a plant for the complete machining up of four or six-throw crankshafts, made from a first-grade quality of high tensile steel, is necessarily somewhat expensive, as several operations are involved; the most important being turning up the end journal bearings and the flange for the flywheel, milling or planing the cheeks of the webs, removing the stock from the centre journal bearings and also the crankpins, by turning or milling (and, in some cases, by drilling and sawing slabs out), machining over the tops of the webs, finishing all the circular bearing surfaces by grinding, and, finally, drilling or boring out the centres of the shaft and pins to reduce the weight, as well as a few minor operations, which will not be discussed in this article, such as drilling oil holes, chipping oil ways, polishing and filing up rough edges, etc.

One of the most common types of

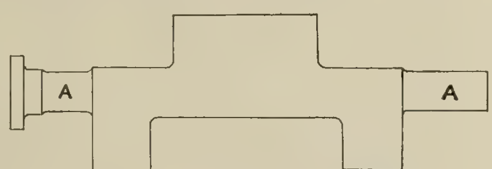


Fig. I.

crank is the four-throw one with the crank pins 180 degrees apart. This type of crankshaft is very often forged similarly to that shown in Fig. I., whilst Fig. II. shows the finished product made from this forging. The first operation is to centre the forging, and rough the two ends A down to about $\frac{1}{8}$in. larger than the finished size, taking care to see that both ends are exactly the same size, this being necessary for subsequent operations. This can be done in any ordinary engine lathe of sufficient capacity, and calls for no special tools or appliances whatever. The forgings are then taken and placed in two vee blocks, as shown at B in Figs. III. and IV., which support the shafts by the previously-turned necks and locate the work for machining the flat sides of the webs, the work being secured to the table by the straps and bolts shown at each end.

Figs. III. and IV. show two crankshafts being planed up at once on a high-speed planer, with two tool boxes on the cross slide, but an alternative and often

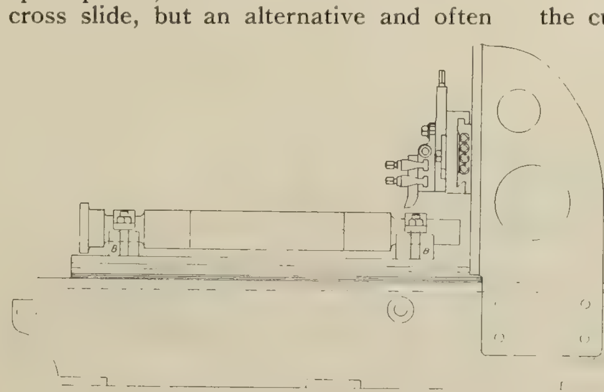


Fig. III.

preferable way is to mill them with a face milling cutter of the inserted cutter type, either on a powerful vertical spindle machine, or a horizontal spindle machine with a long bed and traversing table for the work, very much after the style of a lathe bed. Some firms construct their own special machines for this operation, and use two cutting heads at once, each operating on a separate side, the work being fed automatically between the two cutters, and in this case the forgings are not laid flat, as shown in Figs. III. and IV., but are turned one-quarter round, or stand on their edges as it were, to present two flat faces to the cutters. However, where a modern high-speed planing machine is already installed, planing is quite a satisfactory method, although perhaps not quite so rapid as milling.

We now come to the machining up of the journal bearings and pins, which is one of the chief operations here considered. This operation was formerly done exclusively on the engine lathe, but now two modern methods are in everyday use, either turning with specially-designed high-speed lathes, using multiple cutting tools, or milling with a specially-

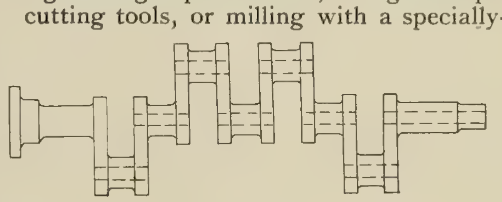


Fig. II.

constructed machine for this purpose, such as the one made by Wm. Muir and Co., Ltd., of Manchester (Fig. VII.), and this latter system has found much favour during the last few years.

Fig. V. shows a crankshaft turning lathe, by Ryder and Sons, specially designed for turning up the journal bearings. It consists of a massive double bed, as shown, the headstocks and stays being carried on the back one, and the two carriages for the cutting tools on the front one, thus

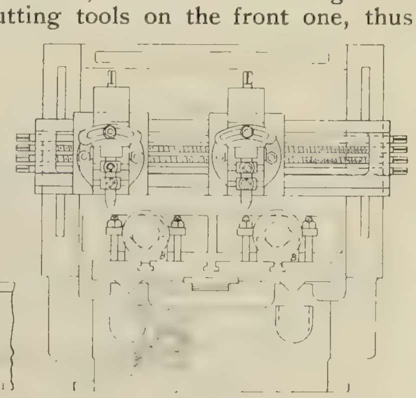


Fig. IV.

allowing the latter free movement along the full length of the bed. The fast headstock is geared and provided with the necessary speeds, whilst the loose headstock differs from the usual type by having a spur wheel driving gear, under the guard shown, which is identical with the one

shown on the fast headstock. The cone pulley, contrary to usual practice, instead of driving a live centre, transmits power to a back shaft through a train of two

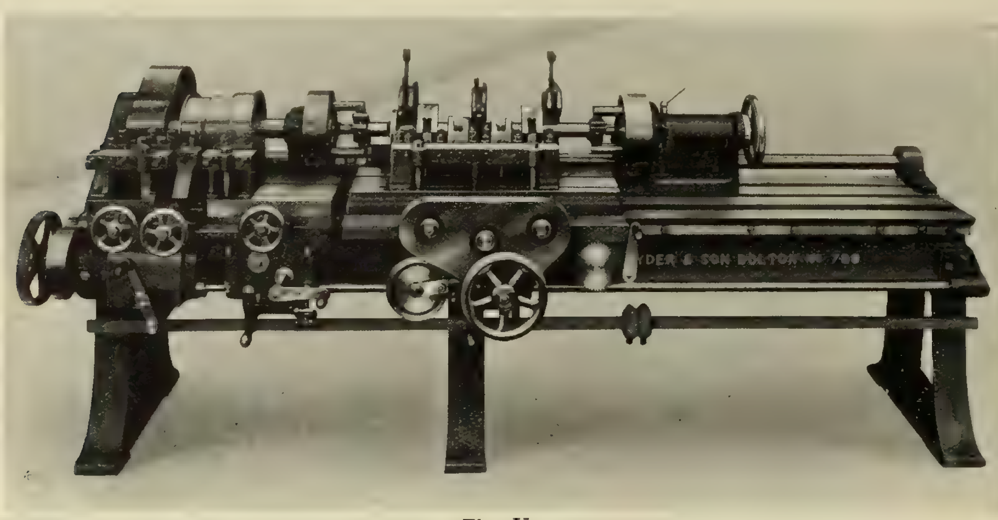


Fig. V.

spur gears, and this shaft drives back the two driving plates, as shown, through similar spur gearing, so that the work is supported on two dead centres and the back shaft, being fitted with a differential gear behind the saddle, drives both ends with an equal load, thus preventing jar and chatter.

The crankshafts ought to have the three journals for the steady rests turned down to grinding size previously, and it is also advisable to drill and saw the centres of the webs out as shown, before mounting in this lathe. Both the rests are fitted with adjustable longitudinal and cross trips, and one carries the necessary tools for turning the tops of the crank webs, whilst the other receives the multiple tool holder for facing the outsides of the webs and turning the journal bearing necks.

The capacity of this lathe may perhaps be appreciated more fully when it is borne in mind that it is possible to have nine cutting tools in operation at the same time. Both the carriages have hand traverse along the bed, as well as the automatic feed and trip motion described, and the particular lathe illustrated will swing 10 inches, and take in 5 feet between the centres. It weighs about four tons when in complete working trim, as shown in the illustration.

For turning the crank pins, and the inside of the crank webs, another suitable lathe by the same makers is shown in Fig. VI. It is driven by a back shaft in the same manner as the lathe described above, the work being carried on two dead centres, but in this case no differential is necessary as the double drive is obtained through two spur rings in the centre of the two steadies, which are of large proportions and support the work close up to the cutting tools. No driving mechanism is attached to either the fast or loose headstock in this lathe, and front and back cutting tools of the crankpin type are mounted on each rest. These have a self-acting surfacing motion and, by means of forming tools, turn between the webs until the correct diameter of the crankpin is obtained, when the feed is automatically tripped by the adjustable stops shown on the front of the apron in the same illustration.

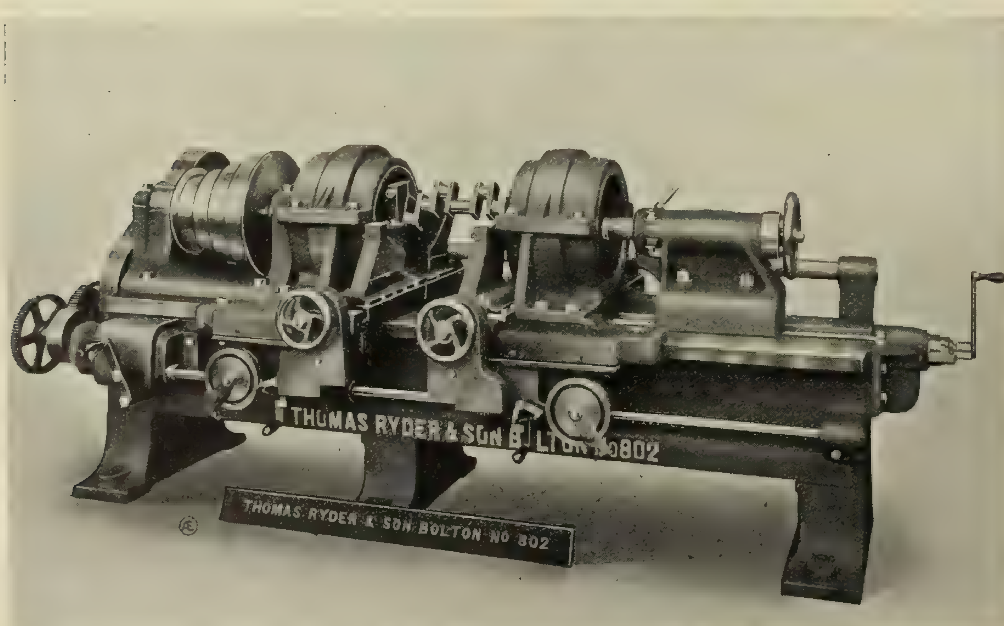


Fig. VI.

For the other system of crankshaft machining Fig. VII. shows Melloy's patent milling machine, which possesses several features of interest. It will first of all be noticed that the bed carries the cutter headstock on a short slide at the back, which is not shown, the work being carried on a table on the long front slide at right angles to the cutter slide. The cutter spindle is driven by spur gearing, shown on the extreme left in Fig. VII., which has superseded the worm drive and bevels that were used

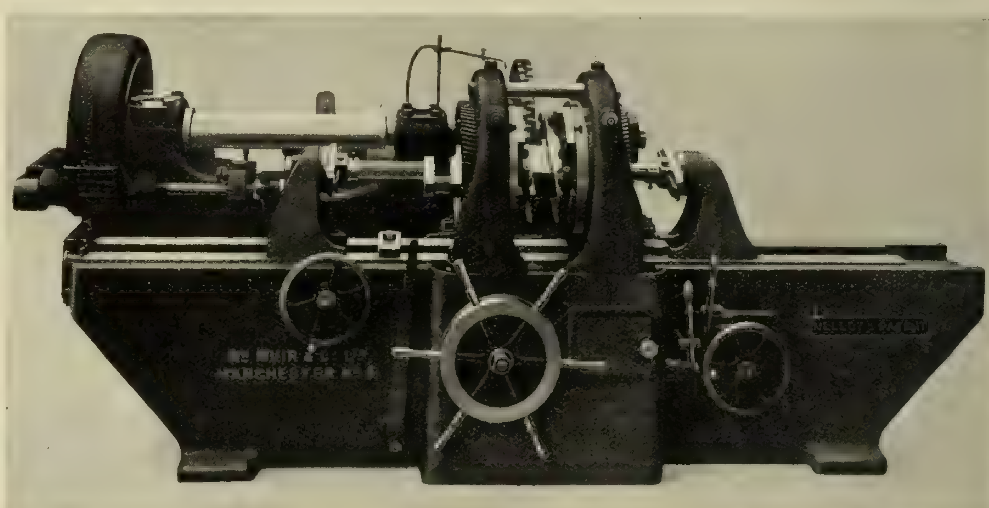


Fig. VII.

previously. The cone pulleys are 25ins. and 30ins. diameter, and take a 5in. belt, so that ample driving power is provided: the cutter spindle is 4in. diameter, and the height of centres 10in. There are two massive bearings and stays on the spindle close up to the cutter, which keep it quite rigid.

The previously turned crankshaft ends are placed in the cup centres of the adjustable headstocks, which are firmly clamped to the table, and the cheeks of the crank webs are gripped by the jaws in the revolving vices, as shown in Fig. VII. The cutter, which is of special design, and of which more will be said later, cuts or saws its way through the blank automatically, as shown in Fig. VIII., until the correct diameter of the pin is reached, plus the grinding allowance of about .030in., when a dog clutch trips the feed automatically; then the binding bolts on top of the revolving heads are slackened, and the circular feed thrown in by a hand lever.

The pair of revolving heads are driven by the two spur-geared rings shown in Fig. VII., and the milling continues round the pin. Then the vice jaws are slacked off, and the table which carries the end stays and work is travelled along until

the next pin is reached, the correct spacing being obtained by dead-length gauges, which are inserted between the end of the table and a micrometer stop,

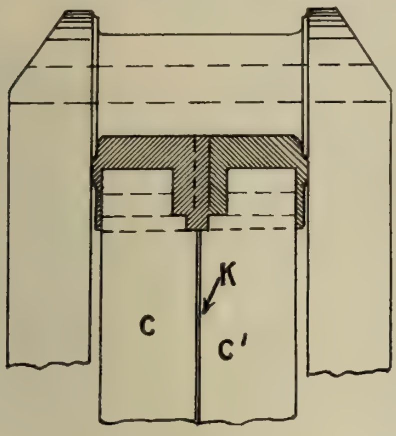


Fig. VIII.

and so no marking out is required. A micrometer adjustment is also fitted to the cross feed, as shown on the pilot hand-wheel.

Referring to the cutter, a section of which is shown in Figs. IX. and X., C and C^1 are two mild steel members. D are the high-speed inserted cutters, held secure by the Bessemer steel wedges E, which have a rounded corner, as shown, to prevent C from bursting. The teeth are staggered as shown to split up the chips, and have two notches ground in them. An enlarged view of the cutters is shown in Fig. X., which show the milled wedge path. It is obvious that the cutter can be adjusted in width to compensate for sharpening, by slackening the bolts F, and inserting thin strips of paper or

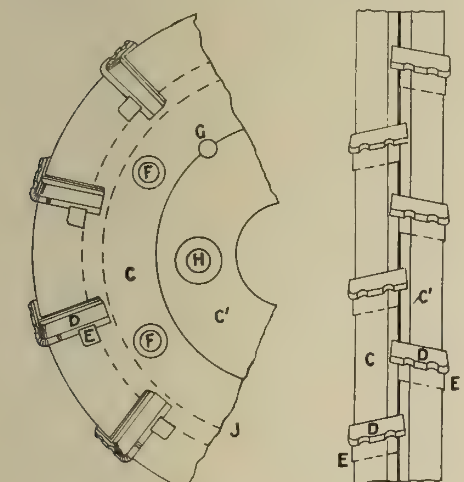


Fig. IX.

other suitable material between the two sides C and C^1.

C is a ring similar to the junk ring of a piston, secured to C^1 by the bolts F, whilst two locating pegs G, across the joint, also help to take the driving strains. Four stout bolts passing through the holes H, and through similar ones in a flange on the spindle, drive the cutter, the two dotted lines I and J, also shown in Fig. VIII., being the two steps turned inside the discs C and C^1, the packing strips being shown at K. The cutter, when dull, may be ground up most economically by a portable electric grinding device, clamped on the bed of the machine for that purpose. Some users of this machine have made their own cutters, which vary somewhat in their design from the one here shown, but after a large amount of experimenting Messrs. Muir claim to have found this construction meets all requirements in a satisfactory manner, although its first cost is somewhat high.

After the shafts are turned or milled, all the circular parts should be ground dead true to size, about 1-64in. being left on for that purpose. Crank-grinding machines at present on the market are very limited in number, the one having the largest sale in this country being the Landis (made in America, and imported into this country by C. W. Burton, Griffiths and Co.), which was described fully in our June issue.

After the shafts have been ground there remains the job of drilling out the centres, as shown by the dotted lines in Fig. 11. An old-fashioned way of doing this is in a lathe, but that method is painfully slow, and obsolete where a large quantity require to be dealt with. Here again several of the large makers construct their own machines to suit their special requirements, but they are all practically very much alike and take the form of a multiple spindle horizontal drilling machine, as shown in Fig. XI. In this machine three holes are drilled at once by the three fixed spindles, one for each pair of pins, and one for the shaft, on a four-throw crankshaft. The case M covers the spur driving gears, which are driven by the constant speed pulley N, the changes of speed being effected by the levers O. The carriage P, carrying the work table Q, is automatically traversed along the bed R by a positive feed gear, obtained either by sprocket wheels S and a driving chain, or by a train of spur gears driving a splined shaft and worm gear, giving the necessary reduction, inside the apron in front of the saddle P. T is the feed box, the changes being effected by the lever U, and a nest of gears, as in modern English lathe practice, and the wheel W is for feeding the carriage along the bed by rack and pinion. X is the lever for reversing the

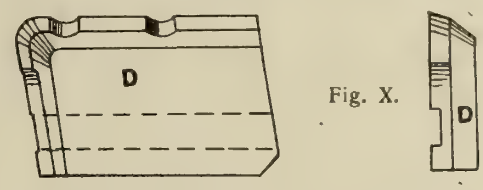


Fig. X.

feed motion of the carriage. The work is located in the two vee blocks Y, and secured by the usual clamps and bolts, adjustable packing pieces being inserted under the crank webs in the centre to give additional rigidity. The three long twist drills should be fitted with oil tubes, and supplied with oil fed under pressure to the points to force out the chips, and thus prevent breakage of drills, and to facilitate rapid boring. All the three drills must be rigidly supported close up to their work, by the hardened and ground steel bushes shown dotted in the brackets Z, which are cast solid on Q. These bushes keep the drills central, and act as jigs, so that once the operator gets the drills started he need have no fear of their running far out of truth, which they would do without these supports. It will be realised that with three drills all working at the same time a very great economy can be effected as compared with boring the holes singly in a lathe.

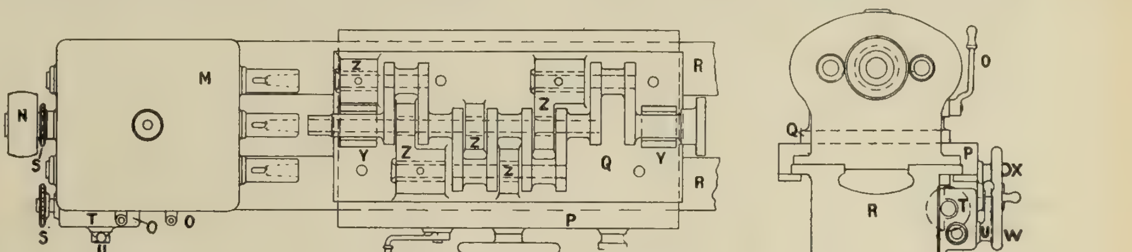


Fig. XI.

The minor operations to complete the crankshafts, such as oil-hole drilling, chipping oil groves, filing and polishing up, etc., call for no special mention here, as no special tools or appliances are needed beyond those found in ordinary everyday use in the majority of engineering establishments where motor vehicle building is carried on.

D. WALTERS.

CORRESPONDENCE.

No responsibility is accepted by the Editor for the opinions of correspondents, and the right is reserved to omit any portion of a letter if it is considered desirable to do so. Letters to be inserted with a nom de plume must be accompanied by the name and address of the writer.

HUBS AND ROLLER BEARINGS.

Sir,—In your July number there was a letter signed "Doublerace," calling attention to the lack of durability of multiple row ball bearings. Having myself had some disappointing experiences with pairs of single row ball journal bearings, placed side by side, although they were fitted with the extremest accuracy, I am wondering whether the double row bearings that are now being put on the market will be any better. Of course, they must be more durable than half themselves, so to speak, but I am very doubtful whether a single large bearing of equivalent price to a double row one would not outwear the latter very considerably. Of course, special designs suit special circumstances always, and there are many places about a car where such a bearing could be housed, while a larger one could not, and it might be possible to alter the ball paths, or the shape of the surfaces on which the balls run, so that any tendency to overload one side of the bearing could be balanced automatically. But the place where most of the trouble seems to arise is in the hub bearings, where there is temptation to keep the bearings small for the sake of the good appearance of a hub that is externally small. Surely if this small size is to be regarded as essential there ought to be some chance for the much-abused and little understood roller bearing.

I think few people realise that roller bearings have been used in Lanchester cars for years (though your pointing this out last month may do some good) and is it not true that the roller bearings on the crankshafts of the 15 h.p. Napiers have given quite satisfactory service? The conditions under which both the last-named bearings and the Lanchester worm-shaft bearing work are much more severe than those which prevail in an ordinary hub. Also, it would be possible to com-

bine a roller bearing with a plain thrust in both directions and the total cost ought to be considerably less than a pair of ordinary ball bearings, and much less than a pair of bearings plus a ball thrust washer. The accuracy of workmanship that is common to-day is so much finer than it was when the first experiments were made with roller bearings that there seems no reason why they should not be all right, and I should be interested to know whether there are any others who agree with me. Why do not the ball bearing makers supply something of the kind?

"ROLLERACE."

ARCHED AXLES.

Sir,—Your article on the arched axle does not seem to be as widely appreciated in some quarters as it ought to be. Personally, I think you have shown very clearly the utter uselessness of the design, and it seems strange to me that first class firms like the Sheffield Simplex Co. consider that the arched axle is worthy of the extra expense of production. Extra useless work in design or production is never of any commercial benefit to any firm, and in these days of keen competition the cost side of every detail must be subjected to a searching scrutiny.

The Sheffield Simplex Co. claim that the following "many useful points are brought out in this type of axle."

First, "the arched axle avoids the extremely

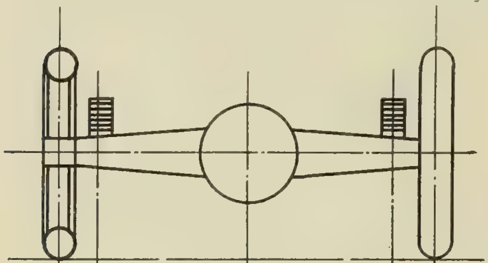


Fig. I.

ugly unmechanical appearance which is presented by a great many fine cars, which are to be seen running about the roads with the wheels doing their best to form a very bad letter A."

Second, the arched axle allows the wheels to be inclined so that the point of contact with the road comes nearer to the spring tables, with the result that "the arched axle, as manufactured by this firm, does substantially reduce the load on the axle case."

Now comes the question, can "the above substantial advantages fairly be claimed for this type?" As regards the first claim, it is common knowledge that there are cars on the road which

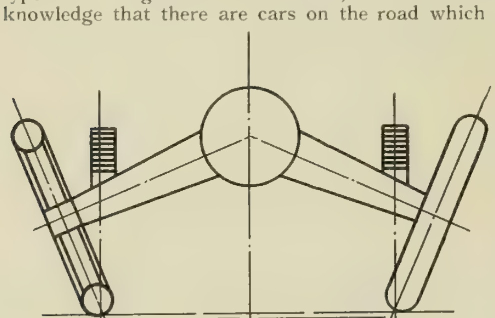


Fig. II.

present a very knock-kneed appearance, but is it due to the inherent defects of the design or the material from which the design is executed? If the design is mechanically wrong, we might expect to find a good majority of cars to be "weak at the knees." As it is, most cars are built with straight rear axles and behave as well as is necessary. Axles may be undersized for the load they have to carry, or may be overloaded. In either case it is detrimental to add load beyond that which the axle is designed to carry. If the arched axle is correct, we may yet hope to see the Sheffield Simplex Co. and others bringing out a car having the front axle arched. What is sauce for the goose is sauce for the gander. In the meantime the vast majority of firms will continue to use either straight or negatively arched front axles. Fashion may decree that the rear axle shall be arched; in that case we must bow the knee to fashion.

The second claim for the arched design certainly seems at first glance to be reasonable, but closer scrutiny reveals its weakness. This claim is shown in an exaggerated form in Fig. II. The axle is so inclined that the points of contact of the tyre with the road are vertically co-axial with the load on the spring tables. The total weight on the rear axle remaining the same, and the claim of the Sheffield Simplex Co. being allowed, namely, that "the arched axle does substantially reduce the load on the axle case," what has become of the excess weight? Consideration will show that a very objectionable bending stress is now brought to bear upon the spokes of the wheel. Out of the frying pan into the fire. Take the load off the axle case, which admittedly will adequately support it, and put it as side thrust on to

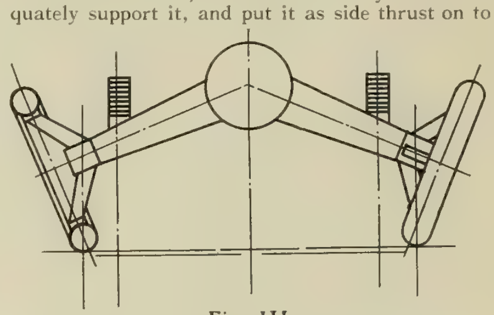


Fig. III.

the spoke, which admittedly is one of the weakest places of the car. As the Simplex Co. favour the dished wheel, it can be here used as a friend indeed. Using a dished wheel, it is possible to get back to a vertical spoke. Now a careful comparison will show that the distance between the point of contact with the road and the centre of the spring table is equal for Fig. I. and Fig. III., so that, in conclusion, the advantages of the arched axle do not seem very pronounced, and as it is a more expensive design and seems to have no material advantages, it might be relegated to oblivion without anybody being any the worse off.

W. SELLS.

SPECIALISATION AND DEPARTMENT FACTORIES.

Sir,—Mr. J. Cooper suggests, and you ask opinions thereon, that articles on Repairs are included in your journal. May I respectfully record my vote against such a policy, which I might liken to a supposed practice of an editor of a fashion journal, who includes columns on the art of stocking mending in his periodical. Records of failure and wear will, however, be more to the point, such, for instance, as touched upon in your very practical article on "Ball Journals and End Thrust."

May I trespass further, in the hope of raising a discussion on manufacture, as at present carried on, in so far as it affects quality of production and, incidentally, the remuneration of capable and experienced engineers.

In my opinion a modern motor car factory is too large, and has too many departments under one management. The motor trade is a good trade, and those responsible for organising a factory do so simply as a business undertaking. The trade being a good one, a firm of manufacturers ought to pay good dividends, and ought not to fail, yet that large manufacturing firms have actually failed is only too well known by those who have been connected with the motor trade in the past. Also, quite a large number of firms have paid small dividends, or no dividends at all, and some have even paid dividends out of capital. Taken altogether, motor manufacturing concerns have not been proportionately so successful as the generality of many other manufacturing firms.

Failures of "mushroom" firms are of no particular interest to engineers, but the fact that large firms have failed in the past, and that firms of standing have made, and are making, small average profits, are worthy of serious consideration. Various reasons have been given for past failures, and poor dividends, such as bad trade, too little capital, or too much capital. Those engineers who worked in the now defunct or resurrected factories will probably call to mind a hundred other reasons for failure, of which the greater number could be catalogued under "overhead charges."

As a rule, it is noticeable that sales drop off for some time before a failure. Reasonable advertising will not sell faulty cars for any length of time, but on the other hand a reputation for excellence will do so, just as with any other goods in demand. If this statement is correct, and it is obviously so to me, continued falling off of sales must be chiefly due to lack of reputation, caused by ignorance or disregard for excellence of output. If, however, sales fall temporarily, on account of bad trade generally, a motor manufacturing firm, as a business concern, ought to have sufficient funds to carry itself over the bad season. If the firm has insufficient funds, as a business firm, its failure must be due to bad management, or a faulty system; assuming the fluctuations of trade average a reasonable return for capital invested. It is a fact that there are large firms doing a healthy trade, being, presumably, well managed, yet I am not satisfied that their system of working is the best, but acknowledge the trade is a vigorous one, and not subject to protracted bad spells. It is a goodly sight to see a large factory at a busy time, when every department is, apparently, working at high efficiency, but I think most practical engineers will agree that the high efficiency of every department is often only apparent, and not real. The truth of this statement can be tested by visiting a busy works with, say, a provincial engineering society, the members of which are practical men. Let one of the party ask the separate opinions of some of his co-visitors, and I am sure he will be answered—by one that such a department is inefficient, and by another that such another department is not as it might be, and he will conclude, after comparing his own opinions, that very few of the departments are working at their best possible efficiency, but that certain departments are paying for the shortcomings of others, and that each is limited in its actions by the whole factory which, in short, must have finished cars, and not a stores full of bits.

At the same works, during a slack season, probably the same staff is kept on as in busy times. The establishment expenses will be about as much as ever, while in the shops there will be rows of idle machines and empty benches. The few departments still at work will probably be behind in deliveries of parts for the tag-end of the last batch of cars. Sales will have fallen off, stock will have to be cleared before further batches are put in hand, and in the meantime the still huge weekly amount must be paid for unproductive labour, which includes the salaries of each departmental expert, as well as those of the men necessary to keep the works organisation from chaos.

It appears to me a question of chance whether the company will or will not tire of paying for nothing before sales have returned to their happier state. If the cars they have sold embody faults in design and workmanship, it is unlikely that they will automatically regain their former volume after a period of bad trade. The cars may be excellent except for one or two minor details, which are only second or third class—little oversights there was no time to remedy during a busy time. In such a case one can believe the few second or third class details will qualify the whole machines in the eyes of the public, who already know cars by their faults and weaknesses quite as well as by their good qualities.

Cars are made in a modern factory pretty much as follows:—

The works manager decides to buy drop forgings, pressings, iron and steel castings, springs, ball bearings, road wheels, tyres, magnetos, accumulators, and a host of accessories. He makes everything he knows how to, and can get plant for in the way of pattern-making, machining, fitting, assembling, etc. Why he does not make stampings and pressings, yet makes lots of little things on automatics and capstans, is beyond my comprehension—it has been suggested the reason is because he is afraid to attempt work outside his experience. The suggestion, however, is not satisfactory, because he is continually attempting to do other work which is quite outside his experience.

The hurried chief draughtsman designs and modifies (sometimes spoils) designs, so that the work may be executed as much as possible with existing plant (including the automatics and capstans). The order clerk feeds the hungry factory with material, a little at a time. The examiner has a difficult time between the frying pan of quality and the fire of output; if not accused of passing scrap he is proved to be hanging up the works. By very hard work the thousands of bits are got through to the assemblers, and cars result. All the heads are intensely interested and enthusiastic, and are always learning something new.—But they are not specialists.

Engineers who talk glibly of turning cars out like sausages have had a remarkably fortunate experience. Many of us remember moderately large batches of cars put in hand before all the designs were completed, and drawings issued; before jigs and special tools were made, before a tithe of the material had arrived, when capstan lathes were set up to do only a dozen pieces before changing for something else urgently wanted, when gears were cut one at a time, and bolts were made on engine lathes, while the army of fitters were filing, filing, filing! The people up-

stairs were smiling and saying, "very busy, turning out —— cars per week."

As a matter of manufacturing fact, the firm, we may remember, could not have done justice to their plant except by practically closing their drawing office, making all necessary jigs and tools, getting in all material, and setting deliberately to work to turn out 1,000 exactly similar cars in a batch, and then only by arranging matters so that no one machine switched off making an article until the whole of the batch were completed—which arrangement would naturally have meant that no cars could have been turned out for a long period of time. However, the management had to have cars or sales would have been lost for ever. So the unfortunate works manager endeavoured to obtain a profitable return for an expenditure of, say, £30 in wages per hour, while the chief draughtsman (with the consciousness, bred of wide engineering experience, that every detail of design would go to make perfect cars) carried in his head as many as he could of the 50,000 dimensions which were, or ought to have been, on the drawings in the shops.—Many a good man attempts too much.

Allow three types of cars being turned out of an ordinary large factory, allow 2,000 *different* parts per car, all to be the perfection of design and workmanship, and all wanted, more or less, at once, because output must be kept up. It is almost an impossible state of affairs; no factory has the requisite number of machine tools to do such work economically in the manner in which the machine tools were designed to be used. This and that machine must sacrifice efficiency in order to help output. That little fault in design, discovered on test, must be overlooked now and corrected on some future cars. Interchangeability must be, for the time, forgotten. Every department must give and take; during a busy time each expert must use his brains upon the cars as a whole, and wink at certain details, in order to keep the assemblers busy.

We have seen, historically speaking, minor faults or oversights in design and workmanship, unavoidable inefficiency of departments, a slack season, low funds. Who is to blame for this state of things?

I consider that no one man is responsible for small average profits, and a questionable reputation, because the system under which he works is wrong. It is wrong in that it supposes that the works manager is a genius, that the chief draughtsman is a genius, that the order clerk is a genius, that the examiner is a genius, and further, that their several ideas in reference to quality and output are exactly of one standard.

Even supposing that each has extraordinary ability, and colossal experience, their ideas, surely, cannot be the same; one may put "output" first, another quality, for instance. Genius is not very common, and I believe the man has yet to be born who can master every detail of a modern motor car factory and turn out perfect cars at a real profit with existing plant. When he is born he will find few engineers who will acknowledge his genius.

We can have specialists, though, who will make a study of the design and manufacture of portions of modern cars, a study which should make for perfection, within human limitations, and profit equal to that of any other business or profession. Specialisation might be arranged in a factory. There might be department foremen specialists, department draughtsmen specialists, etc., carefully trained, and finally allowed to keep their several departments separate from one another, and not permitted to make sacrifices on account of the misfortunes of others, while the managing director, general manager, works manager, etc., would not be called upon to decide minor details. If they do not interfere with the specialists, their salaries would simply come out of the profits. If they interfere, the consequences of their interferences would also come out of the profits, and possibly, out of the reputation of the firm. It appears to me they would be mere figureheads, and their salaries and consequences of their well-meant interferences, pure waste. In a period of bad trade the shareholders would still have to provide their salaries, as well as those of the specialists. If they were loth to do so, the ordinary remedy would be to discharge the specialists. The specialists have, we suppose, been trained at the shareholders' expense, and their places must be refilled when busy times come again, and the expense and delay of training again considered.

Why not dispense with the managing director, general manager, works manager, etc., who have no genius, and are not specialists (the business demands genius or specialisation). Because, obviously, the specialists would be useless in a modern large factory without them.

Then give up the huge factory with its unwieldy system, or rather, split it up into a number of detail factories, each under separate capital, and managed by an expert. It would soon be found which pay and which do not. The vampire departments would have to re-organise. The clamour for output would not destroy the efficiency of every department. The genius whom boards expect, but do not find, would be unnecessary, and the trade, or number of small trades, would be as good as any other.

I believe that, by some such arrangement, better cars would result, because each specialist head of a small department factory would, if he supplied more than one assembling factory, be compelled to work for a reputation; he would live on his merits. Less failures would be chronicled of motor manufacturing firms, because of greater departmental efficiency and therefore proportionately more funds to carry a firm over slack times. A saving of unproductive labour costs would follow; there need be no non-workers in each small department factory. Greater opportunities would obtain for ambitious motor engineers, who would endeavour to specialise and become expert owners or heads of department factories, and pocket a fairer share of the money which now goes to those mysteriously-styled "business men," who, in my opinion, are as much out of place in a factory as colliery owners and coal merchants in a pit. Finally, imagine the joy of all the workers! Joy to be allowed to work all the time, and not be compelled to spend a third of their useful lives in writing on cards and tickets in an almost hopeless effort to keep the complex organisation of the jack-of-all-trades factory in some sort of working order!

I refrain from outlining a definite scheme for department factories. Certain American factories appear to be run on lines somewhat similar to what I have in mind, but it is hard to compare our trade with theirs, on account of the boom in the States. However, though their cars cannot be called better than ours, the Americans have improved designs, and worked at a much faster rate than we have, in the past few years. Whether they will pass us, or even catch us up, is another matter. If we continue to work as we do in large factories, making so many widely different parts, yet turning out comparatively small batches of cars, it is possible that the American specialists will eventually beat us in quality, as they now can in price.

I would like to ask one manager, "why do you not make your ball bearings?" Another, "Why do you not make your piston-rings?" And another, "Why do you try to make gear wheels?"

Perhaps if I could receive sufficient answers to such questions I might alter my belief that the present system of manufacturing motor cars is all wrong, and believe, instead, that there is no system at all.

Occasionally a young man finds his abilities cramped in a large works, and expends his savings in making a repair shop or garage, and England looses an engineer.

Could not this qualified young man employ, say 20 workers, making, for instance, axles, and earn more money than in a factory, and, moreover, turn out better axles than a large motor firm, who may be too busy attending to engines to bother about them?

In my ideal department-factory there are no non-workers. The owner, or manager, has a grasp of the minutest details, he works for a reputation and increased business, and his "axles" will increase in quality, but decrease in price, as he gradually eliminates waste. But he must not make gear-boxes in his axle factory, or he will begin to overlook tiny faults in design and workmanship, and cease to be troubled over the loss of a half-penny here and there.

A salesman, may be, now impresses upon a prospective customer that this car is fitted with a Z magneto. In my millenium he will add it is fitted with A.'s steering column, B.'s frame, C.'s gear-box, etc., and the prospective customer will remember that A., B., and C. are conscientious specialists, who are jealous of their several reputations.

REX.

SKEFKO BALL BEARINGS.

Sir,—Our attention has been drawn to a description of "Skefko" (S.K.F.) Ball Bearings in the August number of "The Automobile Engineer," which we fear will convey the impression to most readers that these bearings are still in the experimental stage. To counteract any such impression, we trust you will be good enough to insert this letter in the next issue of your valuable journal.

Although we have only recently introduced "Skefko" bearings to the English market, they have, *during the last four years*, earned so excellent a reputation on the Continent that the daily output of 20 bearings in 1907 has increased to over 3,000 at the present time. Even with this enormously increased output, the demand is so great that buildings are in progress which, when completed, will more than double the capacity of our factory.

Every "Skefko" bearing is produced from absolutely the highest grade of Swedish charcoal crucible cast steel, hardened and tempered by special processes, involving five distinct operations, which remove all strains and give results unequalled for reliability and durability under the most severe service conditions.

THE SKEFKO BALL BEARING CO., LTD.

CENTRAL PIVOT STEERING.

Sir,—Has Mr. J. L. Napier ever seen the Renouf system of steering applied to a wheel which is interchangeable on front and rear axles.

We can from personal experience say that it is far less complicated than any others we have handled, and bear out Mr. Napier's statement that the raking of the centres in the Renouf hub, which causes the wheel to incline inwards when rounding a curve, has the advantages indicated in your article.

MOORE BROS.

INCHES AND MILLIMETRES.

Sir,—We are pleased to see in your August issue that Mr. Henry Lea champions the cause of the British inch against the millimetre for automobile work. May we also add that our experience in manufacturing the fine mechanism required in our Time Recorders convinces us that the thousandth of an inch is a much better unit for standardising to than the hundredth of a millimetre. We are as much in favour of an international standard as is Mr. Toby, but we make one condition, and that is, let the standard adopted *be the best* we can find for manufacturing purposes. Fortunately for British manufacturers there is nothing in the metric system so useful as the British inch, and we have strong reasons for believing that if it were not for the scientific faddists on the Continent who have the ear of their respective governments, the manufacturers of France, Germany, and other Continental nations would either use the British inch, or slightly increase the size of the metre so as to make it 40 inches exactly, and thus give them the easy relation of 25 millimetres exactly equalling one inch. Hoping to see the "Automobile Engineer" drop metric measurements altogether unless they are made commensurable with the imperial inch—the unit which is probably used in 80 per cent. of the world's manufactures,

GEORGE MOORES AND CO.

CATALOGUES RECEIVED.

STEEL.—J. Armer, 10 Eastcheap, E.C., has sent us a descriptive booklet dealing with the special automobile steels made by Felix Bischoff, of Duisburg. Various kinds of nickel and chrome-nickel steels are mentioned, together with a brief description of the treatment they require. The steels are all supplied either in bars, in forged or drop forged parts, or in finished parts.

SPRINGS.—A very complete catalogue, with the fullest possible particulars and dimensioned illustrations, of their leaf springs for automobiles, has recently been issued by the Westfalische Stahlwerk, of Bochum.

LATHES.—The Colchester Lathe Company have issued a leaflet descriptive of their miniature high-speed lathe, which is intended for general purposes and appears to be a clean and convenient tool.

GEAR CUTTING.—Barlow and Chidlaw, Ltd., have sent us a list detailing the prices for many different varieties of gear cutting for the automobile and other trades. It is well illustrated and should be a handy book of reference for users.

TYRE FILLING COMPOUND.—A price list of Pfleumatic tyre filling containing a full description of the compound, and particulars of some of the tests that have been made of it, has been sent to us by the makers, Pfleumatic (1910), Ltd.

TYRES AND INDIARUBBER SUNDRIES.—A convenient desk card, giving the prices of their inner tubes for car tyres has been issued by the St. Helens Cable and Rubber Co., Ltd. A similar card without prices gives a list of some of the more important sundries manufactured for the motor trade by the same firm.

RECENT AUTOMOBILE PATENTS.

By Eric W. Walford, F.C.I.P.A.

A Compact Four-cylinder Engine.

The details of this engine are described in two recently published patent specifications, and the engine comprises a compact rectangular body forming the water jacket and enclosing four cylinders arranged at the corners of a square and acting through rockers on to the two cranks. The vertical section shows two

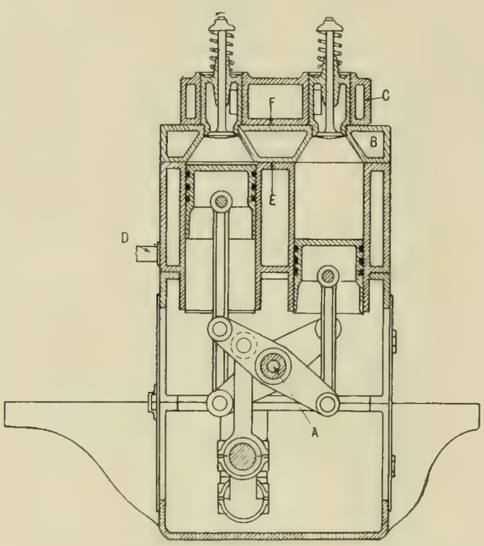


Fig. I.

cylinders with their pistons and connecting rods acting upon opposite ends of rockers mounted upon a central rock shaft A. This shaft is hollow and is fed with oil which is supplied by means of

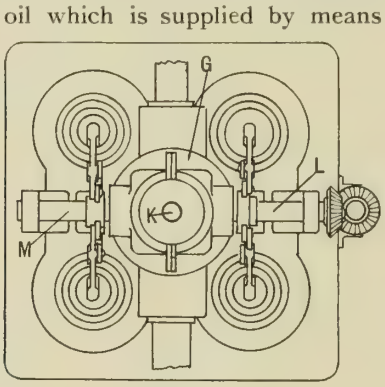


Fig. II.

ducts to the various bearing pins. The rockers are connected by connecting rods to cranks on the crank shaft, which is mounted in the base chamber. Above the cylinders is arranged a separate combustion head B and above this the valve head C. Water is admitted to the water jacket by the inlet pipe D and the cylinders are webbed, so that the water has to travel all round them, finally passing out through an opening at E into

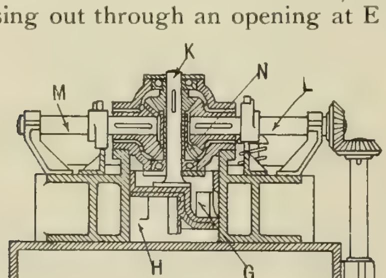


Fig. III.

a water chamber immediately above. The combustion head B is similarly webbed, so that the water has to pass right round the heads before it issues by the opening F into the valve head. Here a similar construction is adopted, so that a circulation system is provided, which results in every part being amply cooled. It will be noticed that each cylinder is provided with a single central valve. Each of these valve chambers communicates with a two-part distributing valve G (see Fig. II.). This is cylindrical, and contains a cylindrical rotating valve, which is divided by means of a web H into two parts. The periphery of the cylindrical valve is formed with ports, each of which communicates with a port controlling one of the valve chambers. By the rotation of the distributing valve the different cylinders are put into communication with the chamber G in turn. The web H, as stated, divides the chamber into two parts, the upper part communicating with the induction pipe, and the lower with the exhaust pipe. The rotating valve is driven by a shaft K, which carries a bevel wheel meshing with a similar wheel on the shafts L and M. These shafts L and M carry cams, which actuate rockers for operating the valves in the four cylinders. In this manner a single broken shaft L and M is used to actuate all the four-cylinder valves, and also the common distributing valve, which is used for inlet and exhaust. It will be noticed that the shaft K is supported on ball bearings arranged in a gear box. The lower bearing is enclosed by a shroud N. This allows the gear box to be filled with oil without fear of it passing down through the ball bearing.

For the purpose of obtaining access to the valve the two shaft portions L and M are moved right and left out of engagement with their respective bevel wheels. The gear box and the wheels are then removed, after which the distributing valve can be raised.

W. J. Robb, W. H. Welch, and Banner Motors, Ltd. No. 10,764/09 and No. 10,874/09.

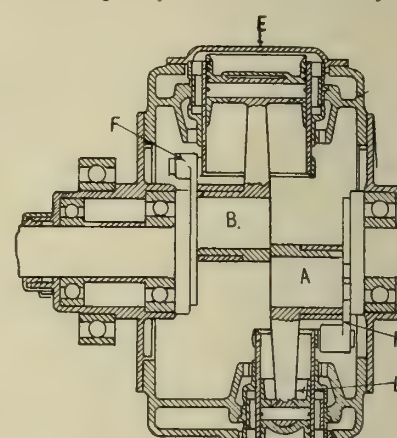
Motor Vehicle Construction.

This construction constitutes an amplification of the unit system, inasmuch as the engine gear box and clutch casing are all bolted together, but in addition thereto the propeller-shaft casing is attached to the gear box, and this in turn is similarly secured to the back axle. These members therefore constitute a rigid backbone supported directly on the back wheels at the rear, and by a trunnion bearing at M at the centre. A somewhat similar construction was adopted in the early 8 h.p. Rover cars. The invention in the present case consists in the method of attachment of the various elements to one another. For this purpose each part is turned with a flanged and tongued joint, and bolted together, so that accurate assembling and alignment is obtained. The invention is shown in the patent specification as applied to a motor car chassis, for which it is claimed to be particularly suitable.

Société Lacoste and Battmann. No. 14,722/09.

An Hydraulic Transmission Gear.

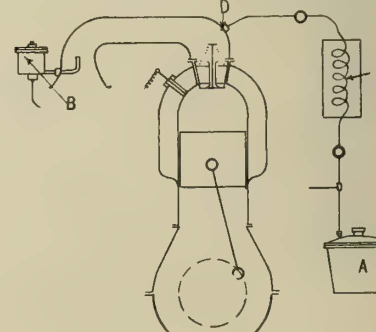
The gear comprises a driving shaft carrying a crank A and a driven shaft carrying a crank B, the two being free and independent of one another. Outside the whole is a casing, supported on ball bearings on the two shafts, and this casing is formed with pump cylinders D of suitable dimensions acting on the crank A, and motor cylinders E of larger dimensions acting on the crank B. Each of the pump and motor cylinders is provided with a slide valve controlled by a rod F driven from an eccentric. By this construction the gear is rendered automatically adjustable, suiting itself to variations in load. When the gear is started up the pumps D are driven by the driving crankshaft, and as they are of smaller capacity than the motor cylinders



E, the latter are rotated at a decreased speed, according to the difference in capacity. As the resistance afforded by the shaft carrying the cranks B decreases the flow of liquid gradually decreases, due to the casing rotating at an increased speed, till a point is arrived at when the casing, the two sets of cylinders, and the two crankshafts rotate at the same speed as a solid mass. At this point the pumps cease to operate. Over each pump cylinder may be arranged a spring-loaded cap or head to afford a resistance to the flow of liquid. This resistance is rendered adjustable. F. Lamplough. No. 15,175/09.

A Paraffin Starting System.

The reservoir A communicates with the ordinary carburettor B, and it is also provided with a by-pass, which traverses a heating chamber C, and enters the induction pipe at D. At starting the heater C is warmed up and the fuel admitted by means of the nozzle D into the induction

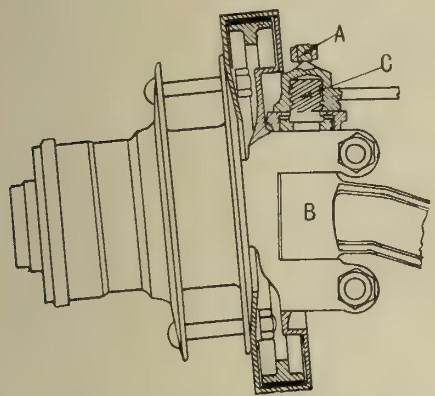


pipe. When the engine is warm enough the by-pass is shut down, and the fuel supplied from the ordinary carburettor B in the ordinary manner. In a multicylinder engine branches are led to each of the cylinders from the heater C.

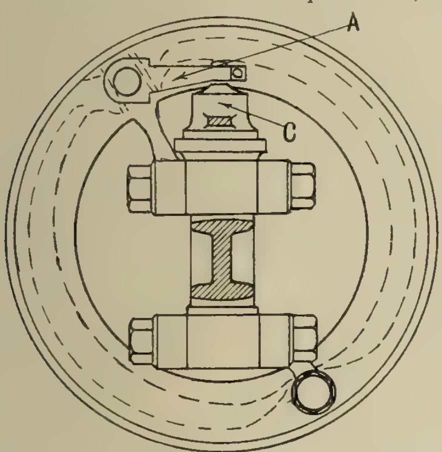
P. Daimler and Daimler Motoren Gesellschaft. No. 15,297/09.

A Front Wheel Brake.

To permit the brake elements to swivel with the wheel and provide a simple means of operating them, the brake-operating lever A is extended so as to pass over the axis of the steering pivot B.



At a point immediately above the axis it is provided with an adjustable bearing portion resting upon a rotatable cap C. This cap is provided with an operating lever connected to the brake lever or pedal, and is adapted to rotate upon a screwed head of the steering pivot B. The screw connection is a quick one, so

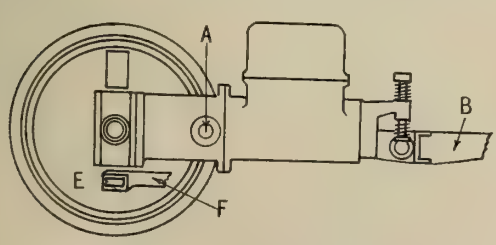


that for small movement of the operating lever the cap rises a considerable amount, turning the actuating lever A about its pivot, and spreading the brake blocks. It will be seen that the construction does not in any way interfere with the swivelling of the wheel for steering purposes.

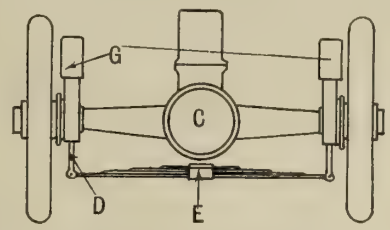
T. G. Allen and The Allen-Liversidge Front Wheel Brakes, Ltd. No. 17,352/09.

Engine Suspension.

The engine and gear box are bolted together, and mounted by means of trunnions A on the frame side members B.



The trunnions A are arranged close to the longitudinal centre of gravity of the gear box, and the gear box terminates in a ring portion, upon which is mounted a ring C attached to the back axle. In this manner the back axle is free to swivel about the gear box as required, and the axle ends are supported by means of rods D upon a transverse spring, which is carried at E by an arm F attached to the



frame. The rods D are provided with pneumatic shock-absorbing devices G, and the engine is mounted upon spring buffers at the front end, as will be seen. A somewhat similar construction is adopted with regard to the front axle.

J. M. Hewitt. No. 15,907/09.

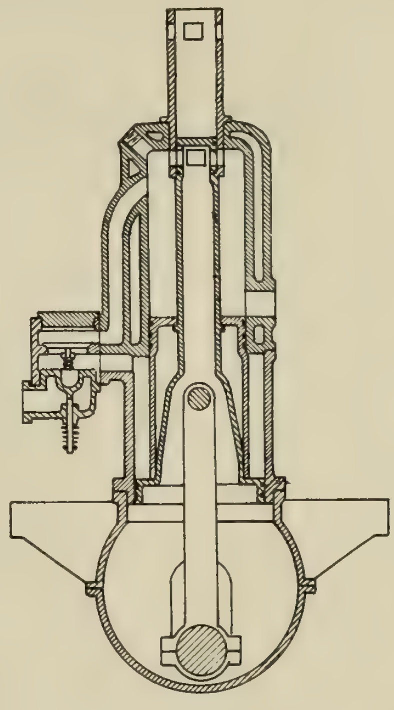
A Scavenging System.

This engine appears at first sight to operate on the two-stroke cycle, and its suction, compression, working, and exhaust are effected in two strokes, but intermediate strokes are used for scavenging purposes. Each of the crank chamber compartments communicates with a valve chest containing a two-part rotary valve A B. This is formed with ports adapted to communicate on one hand with the crank chamber passages C, and on the other with the cylinder passages D, and this valve is rotated by spiral gearing arranged at the front of the engine. The outer valve part A communicates by way of an expansion chamber E with the carburettor, part of the air going through the carburetting chamber and part through an extra air valve. The mixture is admitted to the inner portion B of the rotary valve. During operation it will be understood that on each down stroke gas is compressed in the crank chamber compartments, so that a compressed volume is maintained in the chamber E and the outer valve part A. With the parts as illustrated in the transverse section some of the compressed air in the valve chamber A is admitted to the cylinder at the bottom of the piston stroke, so as to blow out the burnt gases through the exhaust chamber F. In this manner scavenging is effected, and on the upstroke of the piston some of this scavenging air is compressed and finally exhausted at the end of the down stroke. The valve A B is now rotated, so that the inner valve part B is in communication with the cylinder passage D. Thus the compressed air in E is set in motion, passing through the carburettor, and is admitted to the cylinder through the passage D to be carburetted. On the upstroke this air is compressed, at the top fired, and at the bottom exhausted, and the parts again come into the position illustrated in the transverse section, when scavenging is repeated.

Wolseley Tool and Motor Car Co., Ltd., and A. J. Rowledge. No. 15,805/09.

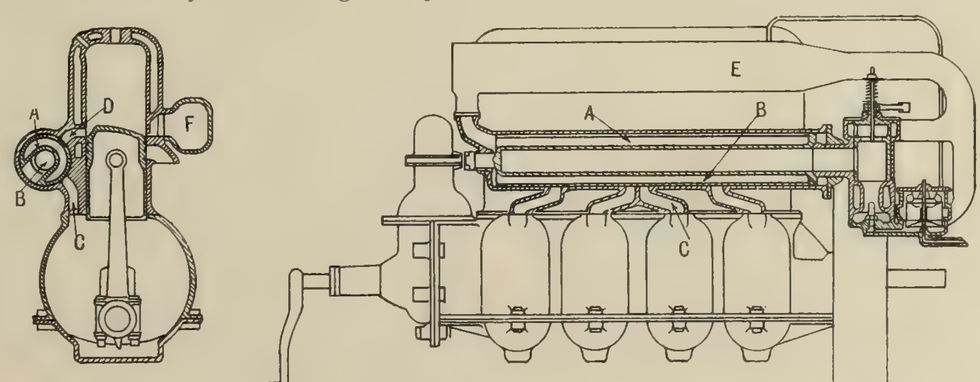
A Two-stroke Engine.

This engine employs a tandem piston and cylinder, the lower cylinder portion drawing in carburetted mixture through the inlet valve, and pumping it through another valve into the top of the working cylinder, exhaust taking place through a port at the bottom of the piston stroke as usual. The feature of the invention consists in the employment of the crank chamber for the compression of air used for scavenging only. It will be noticed that passing through the pistons is a tube which is formed with ports communicating with similar ports in each end



of the sleeve. At the top of the stroke of the piston cool air is admitted through the ports in the tube, allowing the crank chamber to be filled. On the down stroke this is compressed until the position illustrated is reached, when the ports register with each other, and permit the air compressed in the crank case to pass through into the cylinder, and blow the products of combustion out through the exhaust port. In this manner one of the chief difficulties with two-stroke engines is said to be overcome.

C. W. Pradeau. No 1,023/10.



A Scavenging System.

STANDARDS OF MEASUREMENT.

We have lately received a good deal of correspondence, some of which we have published, concerning the relative advantages of the British inch and Continental millimetre for automobile work, and our correspondents have in almost every case urged us to adopt one system to the complete exclusion of the other. It would be easy to do this under certain circumstances, *i.e.*, if all our readers were agreed in giving preference to the same standard of measurement, and if inches and millimetres could be expressed in terms of each other without the use of many decimal figures (as dimensions we may wish to publish will probably be exact in simple figures of either one system or the other).

While we agree entirely that the inch is familiar to a greater number of engineers than is the millimetre, yet but few of those who have once become thoroughly accustomed to the use of the latter would care to have to revert to the inch, principally because of the great convenience of the decimal reckoning. This applies not only to quantities larger than an inch, but to those much smaller, although for some obscure reason it appears to take longer for most men to be able to substitute decimals of a millimetre for thousandths of an inch, than to think of inches as multiples of the same unit.

Therefore, we think we shall be acting in the best interests of our readers if we continue to give dimensions in the standard used by the maker of the mechanism we are describing. It is likely to be many years before we have a single universal system of measurement, and the influence of England and America makes the task of establishing a metric decimal system much harder, and also renders it possible (though not probable) that a decimal system with the inch as standard unit will come into use rather than the metric system as it is to-day.

The mention of standards of measurement with reference to the inch and millimetre suggests that there are very many other standards in other varieties of measurement in an equally chaotic state as regards the comparison of one scale with another. Standards of length are, of course, of the greatest importance to the automobile engineer, as weights, temperatures, and other units in constant use by other travellers of the profession are not matters of everyday reference. Probably the millimetre is used more for automobile work, in Great Britain, than for any other industrial purpose, though it must not be forgotten that the metric system is used almost universally by scientists. This means that a student of engineering who spends a space of two or three years at a college, learning the theory of his business, acquires a considerable familiarity with all forms of metric measurement, and can make use of them with perfect ease, if it happens to be convenient for him to do so, in later years.

This being so, we cannot refrain from the reflection that the next generation of engineers will not be exclusively familiar with the British inch, and, as more and more of them spend their most impressionable years in academic surroundings, the tendency to adopt the existing metric system will most probably increase, on account of the great convenience of the interchangeability of the units of length and volume and their decimal relation to each other.

Legislation with regard to standards of length is not anticipated, at least for very many years to come, and, as we have said already, America complicates the problem; but familiarity with the metric system will grow in the United States just as it will do here, if affairs progress as at present.

We believe that most of those who complain of the inconvenience of the millimetres would soon find that it was much more imaginary than real if they had occasion to work, for a few months, in one of the few British automobile factories where inches are unheard of. It is extraordinary how very quickly workmen become accustomed to the change after being employed for the first time in such a works, in fact the writer has known cases where men of middle age quite lost their "inch sense" in a year or two. This goes to show that universal adoption of the French metric system as it stands would not result in quite the chaos which is usually supposed.

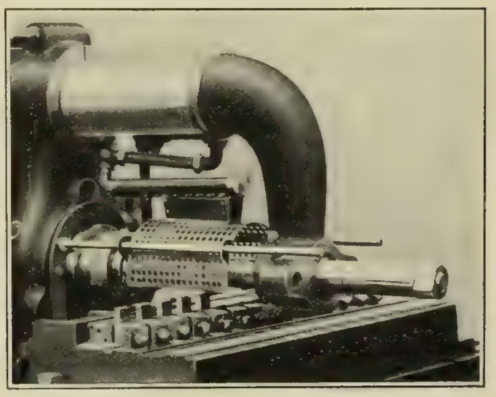
A NEW ALUMINIUM SOLDER.

At the request of Standard Alloys, Ltd., the writer recently witnessed some demonstrations of aluminium soldering with a patent solder made by that company. The operations consisted of lap jointing two pieces of aluminium sheet, about thirty-two gauge, end to end joining two similar pieces, and, finally, making an end to end join of an aluminium tube about ten millimeters in diameter and with walls a little over a millimeter thick. The parts were taken separately and heated in an ordinary Bunsen flame till the solder would run on them, and this appeared to take place at a temperature but little above that at which ordinary soft solder will melt. The hot part was then brushed with an iron wire scratch brush to cause thorough amalgamation of the solder with the piece, no flux being employed. When thus "tinned" the parts were joined by simple simultaneous heating. When the soldering was completed and the parts were quite cold the writer broke all but the lap joint, which resisted all efforts to cause it to part at the join. Both the plate and the tube broke at points close to the joints, but the joints themselves did not give way, the metal parting on either side, much as it would have done had the joints not been there.

The solder is stated to contain no aluminium, but to be similar to ordinary solders, except that it is remelted, after being cast, and is poured, in drops, into a solution of various metallic salts. In solidifying it combines to a very small extent with the metals present in the solution, and will afterwards flow on the aluminium. It is remelted after the precipitation and cast in convenient form. The solder should lend itself to the repair of broken castings as the low temperature needed would remove many of the previously existing difficulties of repair work of this kind.

MILLING CUTTER GUARDS.

We have received the following letter from Messsrs. Alfred Herbert, Ltd. :—" We are in possession of a patent, No. 22,392, dated October 10th, 1906, for a type of milling cutter guard, which has been used extensively in our own works, and by a number of other engineers under license. This guard, one construction of which is shown in the illustration herewith, has been found quite satisfactory in use and has had the approval of H.M. inspectors of factories, having, in fact, been the subject of a special Home Office illustrated circular of recommendation. Believing that it is in the public interest that such pro-

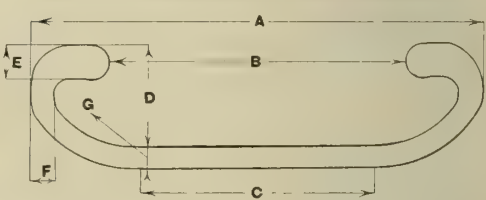


tective devices should be freely available by all users of machinery, we have decided to give up our rights under the above patent for the free use of the public. In view of the agitation which has been in progress for some time on the subject of the guarding of milling cutters, we shall be obliged if you will favour this letter with the publicity of your columns."

RIM STANDARDIZATION.

It may possibly be remembered that the Society of Motor Manufacturers and Traders, some years ago, made investigation of the sections of rim employed by different tyre makers, with a view to the establishment of a series of standard sections. From the information gathered it became obvious that great variations existed, and considerable trouble was experienced in obtaining sufficient details, so, as a result, the matter was dropped. While the subject of tyre interchangeability is not one which concerns the car manufacturer so much as the tyre maker, still it is to the former's advantage to supply rims on his cars which will be the most convenient to his customers, and therefore we hope that the technical committee of the society may see fit to reopen the matter at some not far distant date. The difficulty of establishing a standard lies partly in the fact that there are obvious reasons why the rims used by any particular tyre maker should be objected to by his competitors. We were interested recently to hear that the Rudge-Whitworth Company have adopted their own standards for the rims which they fit to their wire wheels, and we publish the dimensioned sections hereunder, in the hope that they may be useful as a basis of comparison with which to compare other rims, so displaying the differences which exist.

LIMITS OF DIMENSIONS, IN MILLIMETRES.



Size:—	90	105	120	135	175
A	76·2	89·6	98·4	108	122
	77·7	91·1	99·9	110	124
B	50	62·7	67·4	75·4	92·2
	51·5	64·2	68 9	76·9	93·7
C	40.6	40·6	48·2	48·2	82·2
D	16·6	19·8	21·4	22·2	22·2
E	4·57	4·57	4·82	5·08	5·08
	5·08	5·08	5 33	5·59	5·59
F	5·08	5·08	5·33	5·59	4·75
	5·59	5·59	5·84	6·09	5·25
G	2 54	2·54	2·54	2·54	3·54
	3·05	3·05	3·05	3·05	3·3

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Correspondence on interesting subjects, which come within the scope of this journal is cordially invited. All letters must contain the name and address of the writer, but not necessarily for publication. If a reply is desired by post enquiries must be accompanied by a stamped addressed envelope.

CONTENTS.

VALVE MECHANISMS.

ELSEWHERE in this issue we publish a full account of an interesting American view of the relative value of different systems of valve gear for internal combustion engines, but as the author has confined his attention almost entirely to but a single aspect of the case, his conclusions may possibly lack force. The protagonists of the poppet valve are seldom unwise enough to claim for it superiority on all points, unless they are forced to do so from commercial motives alone, as is often the case; and, while the paper read before the American Society of Automobile Engineers concerns little more than valve opening diagrams, there are many other practical matters which may, or may not, outweigh the theoretical advantages of any particular mechanism. Also, so far as we in England are concerned, we have only had experience of a quite limited number of examples of a particular design of valve (if we except the still smaller number of piston valves), and the rest of the world is little, if any, better off, so the calculated superiority of the rotary valve over either the poppet type or the sliding sleeve engine may be said to remain to be demonstrated. It is admitted that a quite perceptible amount of power may be absorbed in the revolution of a rotary valve, just as it has been argued that the sliding sleeves of the Knight engine may require more energy to keep them in motion than do poppet valves, but that this even may account for very little is already common knowledge.

It is not too much to say that, so far as the small internal combustion engine is concerned—developing not more than some ten to twelve horse-power per cylinder—experiment alone can decide whether one design is superior to another if they differ very widely, but in arguing the pros and cons of the valve question nearly everyone must have the Knight engine in mind, consciously or unconsciously. For the sake of the advancement of knowledge it is a good thing that an increasing number of manufacturers are using engines on the Knight principle, because in the course of a year or so small differences in design will begin to appear, and there is little doubt that these evidences of experience will show the position and nature of the weak spots in the Knight system—for every system, however good, must have its weak spots.

At the present moment we have the makers of poppet valve engines experimenting with all kinds of variations of cam and tappet mechanism to eliminate the smallest sounds and to obtain the maximum of efficiency. That the comparative success of the Knight engine is the cause of this activity admits of no denial, and it is also certain that experiment has not been made with poppet valves alone, as witness the several piston valve engines, but so far it is not possible to point to any one engine and to say that it has been proved superior to all others on all points. It is, of course, almost certain that no one engine will ever be in a supreme position again; certainly it will not be so for many years, especially if it is allowed that the four-stroke cycle is just at the beginning of a struggle to the death with the multitudinous examples of two-stroke prime movers utilising the same fuel. It is even practically certain that steam has a permanent place for heavy road vehicles and light boats, where the importance of running cost is very great, although there are indications that the internal combustion engine may replace steam on an increasing number of types of sea-going craft. Really, the matter is one of relative costs, and it seems that steam may be the cheaper form of power under some conditions, and oil or gas in other circumstances.

However, for the present we may confine ourselves to the engines of light road chassis alone, thus simplifying the problem a little. It is perhaps not out of place to urge manufacturers very strongly against the adoption of any new engine system for saleroom purposes only. Unless the new idea is really better than the old idea it cannot live for long, and even if it serves as a stimulus to trade for a short time, once let it prove to be no improvement, once let it show that it has troublesome new faults which require to be learnt, and its days are numbered.

One thing which has made it less easy to weigh the advantages of the Knight engine is the fact that it has a combustion chamber of peculiar shape, and so little is known (though much may be surmised) concerning the effect of combustion chamber form on performance. (It is rather curious that more experiment has not been made with valve pockets in order to obtain some definite data.) Still, it is generally accepted that it is an advantage to obtain equal compression, and so equal mean effective pressure of expansion, in each cylinder of a multi-cylinder engine, as the equal effort results in even torque at slow speeds, perceptible to the senses as smooth running, and it is a fact that this smoothness is to be observed on almost any engine where especial care is taken to secure equal-sized combustion chambers. It is more easy to obtain this desired end with the Knight than with most other engines, so this may be reckoned as one advantage. Again, it must be remembered that very small differences in combustion chamber volume are sufficient to affect the running, likewise in any engine the smoothness

of running depends upon the maintenance of equal valve tightness and equal opening area on each cylinder, and these could scarcely differ on a Knight engine, though they can but seldom be made exactly similar with poppet valves. Briefly, it is possible to get equally smooth slow speed running from either type, but it is less difficult to get it with the Knight.

To continue the good points, there is the fact that the valve diagram compares favourably with that for the best poppet valves of reasonable size, and, still more important, the valves operate with equal accuracy at the highest and lowest speeds of crankshaft revolution, whereas there is some trouble to be met if a large poppet valve is to follow its cam exactly at high speeds.

The effect of age upon these advantages cannot be determined as yet, owing to the small number of Knight engines that have been in use for more than two years at most. Doubtless the piston and the inner surface of the inner sleeve will wear just as an ordinary piston and cylinder, but it is conceivable that the life of the main cylinders and outer sleeves might be very long indeed, except that if lubrication fails the sleeves are liable to break away from their operating mechanism. Poor lubrication will certainly not injure a poppet valve engine so much as it will harm the Knight, but though this is a disadvantage it is not a very important one, especially as lubrication becomes more and more automatic, and so less and less liable to neglect. However, putting aside complete failures, and considering only as short a period of use as twelve months, it would appear that the Knight valve gear would require less attention than poppet valves, and would be in better order at the finish; for unless the grinding in and tappet adjustment of the latter was performed frequently and carefully, the efficiency would drop, and the Knight valves have no deterioration which corresponds to the pitting of the poppet valve, while there are no parts like the cams, tappets, springs, and valve guides to cause inaccuracy of timing by wear.

On the other hand, as an outstanding advantage of the poppet valve, there is its lower cost of production, and also the ease of replacement of worn parts, for if wear takes place more quickly, its effects are more easily removed. It might be maintained that the sleeves and their operating mechanism could be produced as cheaply as all the more numerous components of a poppet valve system, but if cost was made a basis of competition the poppet valve maker could gain much by a small sacrifice to noise, whereas the Knight engine must be either well made or not made at all. Still, it is hardly fair to consider this as an advantage when efficiency is the chief matter.

Mention of noise brings us to the sound comparison of the two systems, and here again it is more easy to make the Knight valves free from noise-producing-features than it is to overcome the natural tendencies of the poppet valve. Theoretically the poppet valve is lifted gently from its seating by the action of the cam, and is as gently let down again, but, of course, this can never be so in practice above quite moderate speeds, and it is necessary to introduce cushioning springs between tappet and valve stem end, and perhaps to provide other springs to maintain the contact between the cams and the tappets; but it is also a fact that the poppet valve can be made so nearly as silent as the Knight valve that the advantge of the latter is negligible. It would be indeed hard to say which was the less complicated, a Knight engine or a poppet valve engine quite equally silent; probably there would be but little to choose between them, but the silencing of the standard type would destroy much of its advantage as regards cost.

It must not be forgotten that it has been found impossible to obtain the maximum of efficiency and silence from any one poppet valve engine; that is to say, if the utmost power is to be secured from a given size of cylinder, very large valves have to be used, with very powerful springs, and besides, the cam form requires to be such that its tendency is to produce much noise. Also, engines which have a very free intake and outlet are less susceptible to small throttle variations as a rule, which means that they are not smooth in action or convenient to control. In fact, the successful contest-car engine is invariably unsuitable for ordinary use, on account of its uncontrollability. The majority of poppet valve engines are more or less throttled on their valves, while the Knight engine with its large ports requires to be throttled on the carburettor, at the air intake, or elsewhere. This means that full advantage cannot be taken of rapid and large valve opening on any type of engine, if the maximum smoothness and controllability is to be obtained.

Of course, all the sleeve-valve engines that have been made in this country so far have been of the touring car type, and all the successful racing engines, for either cars or boats, have had poppet valves, but it is safe to assume that if a sleeve valve-engined vehicle was to be entered for the honours of the racing track, it would prove to be a very different mechanism to handle from its less efficient prototype. We do not suggest that unsuitability for racing is a drawback; it may, indeed, be quite the opposite, but we would point out that because all the Knight engines now to be found in use are smooth running and quiet, it does not follow that they would be any better than the poppet valve type, if made for racing, except that the Knight engine would always be more quiet by reason of the absence of concussions in the valve gear.

We have already made mention of wear, and have expressed the opinion that the poppet valve is likely to give more trouble from minor wear and maladjustment, but complete failure ought to be considered also, and the Knight is at a disadvantage here. Not only is it more susceptible to any vagaries of the lubrication system, but almost any malady to which it is subject is paralysing in its effects, if not absolutely fatal. A poppet valve engine will continue to run and do work even when it is in thoroughly bad order, though it will, of course, not do its best work; but if there is anything wrong with the Knight engine it would appear that it must either stop or break some part. Practically, the only harmless form of derangement to which it is liable is wear of the eccentrics, or the connecting rod ends, and this can do no more than to cause a knock, until it reaches a stage where it begins to affect the timing; but leakage past a burnt junk ring, or such wear of the sleeves that hot gases began to get between them could only result in rapid complete breakdown. This is, of course, surmise only, but it is an undoubtedly reasonable assumption.

So far but small progress has been made towards any definite conclusion as regards the merits and demerits of the respective designs, and we think that it would be wrong to form conclusions until such time as there are not one, not one hundred, but many hundred old Knight engines to observe. It is from the repair shop that the ultimate decision must come. Much criticism of a perfectly reasonable nature was heard at the time of the introduction of the Knight engine, and much of that criticism has since proved to be unfounded, as has some of the poppet valve criticism made on behalf of the Knight. Still, many manufacturers feel that so daring an innovation, that has so far succeeded so well, is a power in the land, and is not to be regarded any longer as a momentary phase, to be forgotten in a year or so. The question is: Does this success mean that the Knight engine is an all-round better engine than the old pattern, or does it mean that it is merely equally reliable and efficient?

Always with the reservation that time may alter conclusions, it appears that it is in some respects better, in some worse, and, as a whole, much the same. It is a means to an end, or a means to several ends; it is not the only means; it is not necessarily the best means, but it is at least better than its predecessors in absence of liability to petty derangements.

As regards acceleration, it will do no more than any other equally well-made engine with equal dimensions and equal valve diagram; in fact, there are many poppet valve engines which develop more horse-power in proportion to their size than any Knight engine yet built, so far as we are aware, but if acceleration and absence of noise are considered together, it is better than the great majority of other engines, though not than all. It is considerably less likely to become noisy as it wears than is the poppet type, because the small connecting rods are the only part of the Knight valve gear which could possibly become noisy through wear, and, if large enough, this wear would be very slow indeed. Finally, the Knight probably requires rather less "tuning up" than most engines, which is, of course, a manufacturing advantage.

The user is so little troubled with engines to-day that any new engine would have to work for years, without any attention, save lubrication and cooling, if it were to be conspicuously more reliable than the average. Also nearly every inadvertent stop is due to either ignition or carburation, neither of which come within the limits of the present discussion, and it is hard to see in what way the Knight engine is better than any other first-class engines for the passenger in, or the owner of, the car to which it is fitted; except that the valves never require any attention.

It appears to be a manufacturer's question pure and simple. It is possible to obtain desired smoothness, accelerative powers, reliability, and quietness in different ways. The use of the Knight patents is one way, and is, to an extent, simple, but the best way must depend upon the peculiar circumstances of the particular maker.

THE ARRANGEMENT OF HUB BEARINGS.

By John V. Pugh.

"DESIGN" does not convey my meaning, nor does "arrangement," but "arrangement" comes nearer. Any obscurity remaining will be cleared up in the context.

In the following article I do not intend to deal with plain bearings. A good case for their use can certainly be made out on paper, and possibly in practice, but

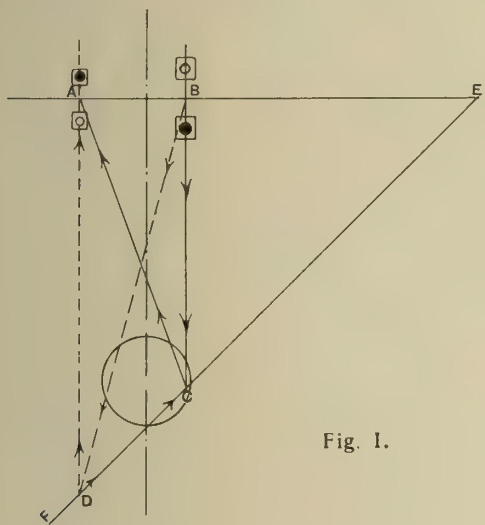


Fig. I.

an equally good case can be made out for the use of plain bearings in bicycles. I am confident no attempt will be made to re-introduce plain bearings in bicycles, and it is nearly as unlikely that they will again be used on the road wheels of automobiles.

I believe the merits of ball bearings are more than a mere talking point for a salesman, but putting the matter on a no higher level—regarding it merely as a selling point—I do not think plain bearings will ever be used again. Roller bearings, either cylindrical or conical, have not yet gained general acceptance and may never do so. In any case the reasoning that follows applies to them too, with but a few modifications, and I shall therefore confine my remarks to ball bearings alone. Though so universally accepted and used, ball bearings for road wheels are a far more recent development than is generally recognised, and in common with most new uses of an old thing, many of the points which ought to have been considered have been ignored. It is not overstating the case to say that their employment in road wheels has not proceeded to any large extent on the lines of deductive reasoning, the more costly procedure of trial and error having been employed in most cases. This is the more remarkable in view of the fact that so many of the men prominent in the design of motor cars have been drawn from the cycle trade, where so much has been learned and so much more has been bitterly unlearned with regard to ball bearings.

On the other hand, it must be admitted that bicycle traditions account directly for some failures. For example, in the earliest applications ball bearings were taken from the bicycle, lock, stock, and barrel, or rather, cup, balls, and cone; were enlarged slightly and placed on the motor car without much thought.

The conditions differ very widely, however. The bicycle, unlike an automobile, is not cleaned with a hose, and bicycle bearings that will keep out rain will get full of water if a hose is used. Again, a bicycle is propelled by a sensitive owner, and if a bearing is screwed too tight the owner soon finds it out and adjusts it properly; also, a bicycle can be lifted up to see if the bearings are free, but with a car, unless a bearing shrieks (an unusual occurrence), a stiffness that is breaking up a bearing is usually not noticed, or if it slows down the car appreciably, which is very unlikely, the result will be attributed to some one or other of the many little matters that may cause lack of power on so complex a machine. The hose trouble is a very real one, and is not discovered during testing, for when a car of a new type is put on test the hose is rarely used, and in spite of the bad appearance all goes well. Give that car to the private owner and the hose is used; the car may look splendid, but the bearings become a whited sepulchre full of water, which brings trouble, and so the cup and cone bearing failed because no proper provision for the exclusion of water was made.

It is possible that the cup and cone bearing, properly protected, may some day become popular, but the bad results it gave owing to defective design enabled the journal bearing that had been developed for other engineering purposes to take its place.

It was well made, and even in its first applications was rather better protected from the ingress of water and dirt than the cup and cone. The automobile designer looked at the makers' catalogue and found that a bearing 25 mm. inside and 52 mm. outside was listed as suitable for a load of 350lbs. He pro-

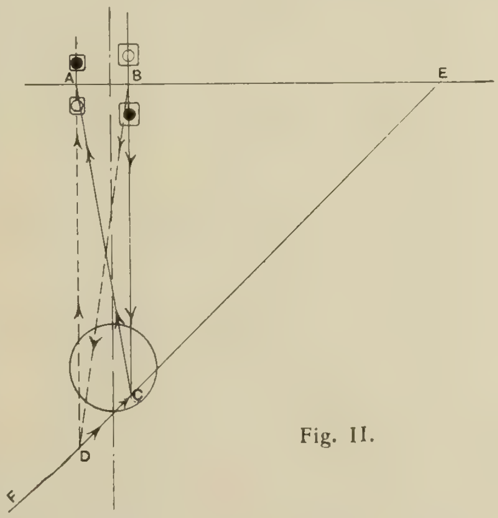


Fig. II.

posed employing eight bearings to carry his car, and the exigencies of design, or some unconscious instinct, caused him to use larger bearings than this in six out of the eight applications, and he considered he had a very ample margin of safety; for eight times 350 lbs. equals 2,800 lbs., which is rather more than the weight of the car of those days.

In view of the bearing pressures that are later shown to exist, it is really remarkable how few of these small bearings gave trouble, even though there is no doubt that many makers understate the safe load. The percentage of breakages will never be ascertained, but broken bearings are costly and they are dangerous, and only a few failures compel a re-design. In cases of failure, various remedies are at once suggested. The maker of the bearings that have gone wrong demands the use of larger bearings, and the provision of special and rather costly thrust bearings. (The thrust bearings in themselves are not all

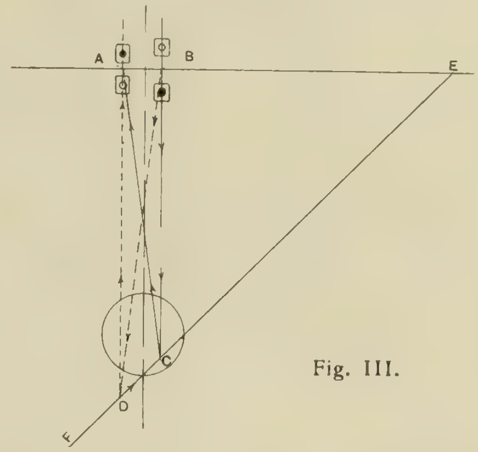


Fig. III.

the expense, the mounting is complicated and needs great precision in manufacture.) The repair department notices the rusty condition of the broken or worn out bearings and, associating rust with water, demands a means of keeping the water out and keeping the grease in. Rival bearing makers endorse all the other suggestions, but not unnaturally advocate a change in the source of supply.

As a rule all these remedies are adopted and there is no more trouble, though there are sometimes more failures and further changes, but the result is deplorably unscientific in any case. The car manufacturer has changed size and make, has provided thrust bearings, and has kept oil in and water out, but he has no chance of obtaining the slightest idea as to which of the remedies has relieved him from trouble.

It seems desirable to set out at this stage what happens to bearings in the way of forces acting on them.

It is quite clear that the only external force is applied to the tyre at its contact with the road, and there is no simple axial thrust applied to the centre of the wheel. The problem yields exceedingly well to the ordinary methods of graphic analysis, as shown in Figs. I., II., III., IV., V., VI. and VII. We will call the force of the road on the tyre F.; and its direction depends on the coefficient of friction, which varies with different road surfaces. It is unlikely ever to exceed unity, which means that F. can be no nearer the horizontal than 45°. The maximum magnitude of the force F. depends entirely on the design of the car, the position of the centre of gravity, and the angle of lock of the wheels. The limit is reached with a front wheel when there is a sufficient lock for the axis of the wheel on the outside of the curve to pass through the centre of

gravity, and for the relation between the curve and the speed of the car to be such that the one wheel will take the whole of the load. Under these conditions the force F. will reach the remarkable figure of 1.413 times the weight of the car. Of

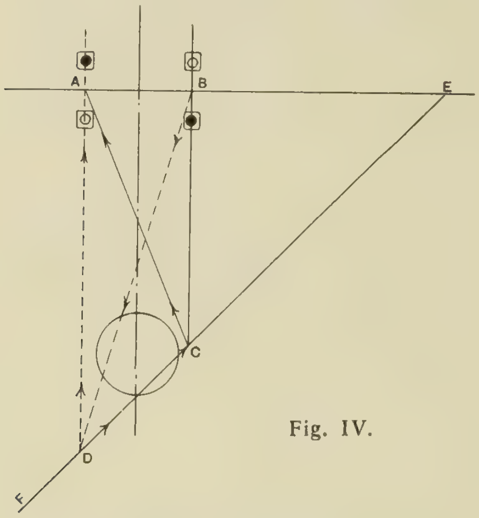


Fig. IV.

course it will be understood that such a condition would be very extreme, and rarely, if ever, reached, but it is probably far nearer the truth as regards what a bearing has to stand than the general idea of assuming that only one-third of the weight of the car is on the front wheels, and that in consequence each wheel has to withstand a force equal to only one-sixth the weight of the car. With such a wide range of possibilities an exact estimate of what will happen to any particular car, in the way of forces on the wheels, is out of the question. It should be remembered, however, that good wooden wheels will withstand a side stress of about 3,000 lbs., yet they have been known to collapse at a corner taken too fast, or more generally when turning short to avoid an unexpected obstacle. (A wire wheel of the latest construction will bear from two to three times this amount.) It is certainly not a serious exaggeration to say that the force F. on one front wheel (not the horizontal component of it) quite frequently equals three-quarters of the total weight of the car.

Fig. I. shows a very simple way of analysing the effect on the bearings of this force F. The force scale is so chosen that the force F. equals D.C., the hori-

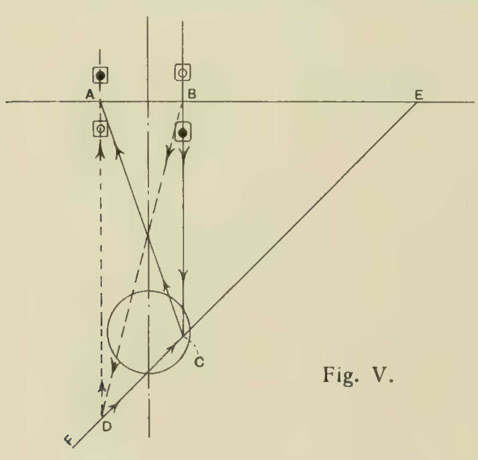


Fig. V.

zontal component of which is equal to A.B., the distance between the two bearings. The line D.C., when produced, intersects the axis at E. Now imagine that a special bearing is provided to take the thrust; this thrust, and the force F., must intersect at the point E. Consequently, their resultant must intersect at the same point and their resultant must be equal to the vertical component of the force F. It is quite clear then that the forces on the two bearings are entirely due to a force equal to the vertical component of F., and act at the point E., and are equal to a downward force at B., equal to A.D., and an upward force at A. equal to B.C. To make the diagram quite clear, the ball which is in compression on each bearing is shown solid black.

Suppose that no special thrust is provided, and the so-called journal bearing A. has to take the end thrust, which it will be remembered is equal in magnitude to A.B. The resulting force will still be upward, but at an angle to the vertical, and will be represented by A.C., while the force at B. will still be equal to A.D., its magnitude being unaltered by the fact that the bearing at A. takes the thrust. Again, if B. takes the thrust the force will be still downwards, but at an angle, and will be equal to B.D. in direction and magnitude, while the force at A. will be vertical, upwards, and equal to B.C. Next to the magnitude of the forces prob-

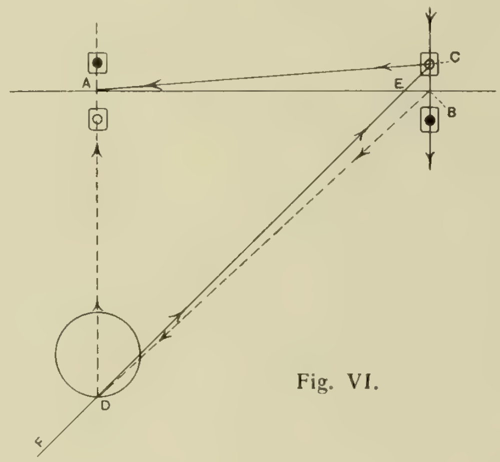


Fig. VI.

ably the most remarkable feature is the very slight inclination that the thrust gives to them. There are, of course, many cars of very high reputation that run without thrusts of any kind on their road wheels, simply relying on the ability of the ordinary single row bearing to deal satisfactorily with a certain amount of end thrust as well as load. This diagram suggests that these firms are not incorrect.

Fig. II. shows the same conditions precisely as Fig. I., except that the two ball races are very much nearer together, being, indeed, only half the distance apart. For convenience of the diagram, although the scale of the force has been kept the same, F. has been estimated to be only half of what it is in Fig. I. Consequently, the distance A.B. still represents the horizontal component of the force. It will be seen at once that the resulting forces on the bearings are almost exactly the same as in the previous case, although F. is only half as large. If F. had been taken to be the same in both figures the bearing pressures would have been almost exactly twice as much in Fig. II. as in Fig. I.; and this shows that it is far more important to separate the bearings as far as possible than to provide additional and complicated thrusts.

I have set out in a table the results of graphic analysis of the variety of bearings shown in the diagrams. The same construction applies to them all, so that detailed description of each one is not needed, though some general comments may be useful.

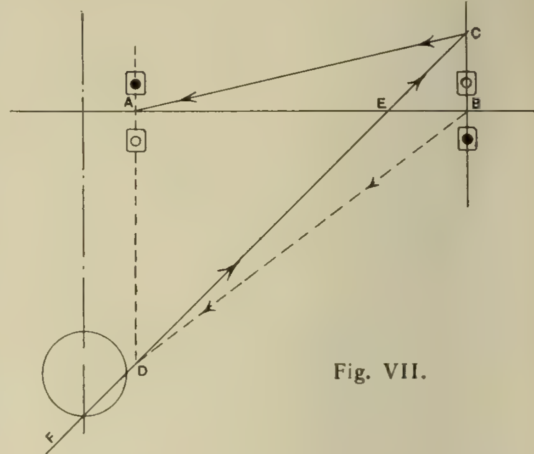


Fig. VII.

I have assumed in front wheels that F. may, under extreme conditions, equal three-quarters W., where W. is the weight of the car. In back wheels F. reaches its maximum when a curve is being taken at so high a speed that only the outer front and rear wheels take the weight, and when the curve is of so large a radius that the proportion of force taken by the front wheels is not much increased. Assuming that the back wheels take two-thirds of the weight, the force F. can reach as a maximum 1.413 times two-thirds of the weight=.942 W., as against 1.413 W. in the case of a front wheel. In the diagrams I have assumed that the car weighs 4,000 lbs. and that F. =3,000 lbs. for a front wheel, and 2,000 lbs. for a rear wheel.

It will be observed with regard to Fig. I. representing an admirable arrangement of front hub bearings that the pressure on the bearings varies from about $1\frac{1}{2}$ W. to 2 W., and that these pressures are not materially reduced by the introduction of a special thrust bearing. It will also be observed that the direction of thrust

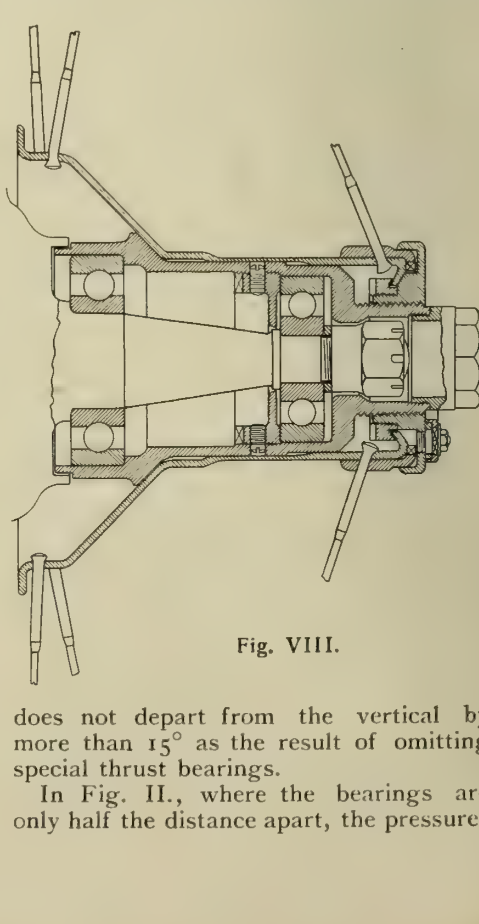


Fig. VIII.

does not depart from the vertical by more than 15° as the result of omitting special thrust bearings.

In Fig. II., where the bearings are only half the distance apart, the pressures

on the bearings in all cases exceed 3 W., while in Fig. III., the bearings being still closer, they are nearly 4 W. on the outer bearing, and nearly 4½ W. on the inner bearing, while the departure from the vertical is still less.

The back wheel, Fig. IV., is a typical example of the two rows of bearings set well apart. The pressures are from ¾ W. to 1¼ W., but the departure from the vertical is far greater, suggesting that if thrusts are needed at all, they are needed here, and yet it is very rare indeed for special thrust bearings to be employed in conjunction with this design, and excellent results are obtained without them. No doubt the low pressures disclosed account for this fully.

Figs. VI. and VII. show another back wheel system, in which a single journal bearing is employed at the hub, and another near the centre of the axle, and the surprising features disclosed are that in no case does the bearing pressure reach ½ W., and that the differences in pressures obtained by employing a separate thrust bearing probably justify the extra complication, for the bearing pressure is then reduced to less than ⅓ W. It is remarkable how little difference there is between the pressures in Fig. VI. where the bearing is in the central plane of the wheel, and in Fig. VII., where it is 77 mm. away from that plane; indeed, the higher pressures are reached in the former case. On the other hand, it must not be forgotten that the advantage lies with Fig. VI. in the case of severe jolts. These jolts are of considerable importance, and their

TABLE SHOWING FORCES ON BEARINGS.

In Front Wheels F equals 3,000 lbs.

In Back Wheels F equals 2,000 lbs.

FIGURE I.—FRONT WHEEL, DISTANCE FROM CENTRE TO CENTRE OF BEARINGS 143 mm.

Force at Bearing A.		Force at Bearing B.		Axial Thrust.	
Magnitude	Direction.	Magnitude.	Direction	Magnitude.	
5,810	90°	7,940	90°	2,120	Thrust taken by separate bearing.
6,170	70°	7,940	90°	—	A takes thrust.
5,810	90°	8,220	105°	—	B takes thrust.

FIGURE II.—FRONT WHEEL, DISTANCE FROM CENTRE TO CENTRE OF BEARINGS 71.5 mm.

Force at Bearing A.		Force at Bearing B.		Axial Thrust.	
Magnitude.	Direction.	Magnitude.	Direction.	Magnitude	
12,400	90°	14,500	90°	2,120	Thrust taken by separate bearing.
12,580	80.5°	14,500	90°	—	A takes thrust.
12,400	90°	14,660	98.5°	—	B takes thrust.

FIGURE III.—FRONT WHEEL, DISTANCE FROM CENTRE TO CENTRE OF BEARINGS 57 mm.

Force at Bearing A.		Force at Bearing B.		Axial Thrust.	
Magnitude.	Direction.	Magnitude.	Direction.	Magnitude	
15,700	90°	17,830	90°	2,120	Thrust taken by separate bearing.
15,860	82°	17,830	90°	—	A takes thrust.
15,700	90°	17,950	96°	—	B takes thrust.

FIGURE IV.—BACK WHEEL, DISTANCE FROM CENTRE TO CENTRE OF BEARINGS 156 mm.

Force at Bearing A.		Force at Bearing B.		Axial Thrust.	
Magnitude.	Direction.	Magnitude.	Direction.	Magnitude	
3,310	90°	4,730	90°	1,413	Thrust taken by separate bearing.
3,610	67°	4,730	90°	—	A takes thrust.
3,310	90°	4,930	106.5°	—	B takes thrust.

FIGURE V.—BACK WHEEL, DISTANCE FROM CENTRE TO CENTRE OF BEARINGS 120 mm.

Force at Bearing A.		Force at Bearing B.		Axial Thrust.	
Magnitude.	Direction.	Magnitude	Direction.	Magnitude	
4,685	90°	6,090	90°	1,413	Thrust taken by separate bearing.
4,880	72°	6,090	90°	—	A takes thrust.
4,685	90°	6,250	103°	—	B takes thrust.

FIGURE VI.—BACK WHEEL, ONE BEARING IN THE CENTRAL PLANE OF TYRE, THE OTHER BEARING AT DIFFERENTIAL. DISTANCE FROM CENTRE TO CENTRE OF BEARINGS 480 mm.

Force at Bearing A.		Force at Bearing B.		Axial Thrust.	
Magnitude.	Direction.	Magnitude.	Direction.	Magnitude	
101	90°	1,312	90°	1,413	Thrust taken by separate bearing.
1,420	4°	1,312	90°	—	A takes thrust
101	90°	1,935	43°	—	B takes thrust

FIGURE VII.—BACK WHEEL, ONE BEARING 77 mm. FROM CENTRAL PLANE OF TYRE, THE OTHER BEARING AT DIFFERENTIAL. DISTANCE FROM CENTRE TO CENTRE OF BEARINGS 480 mm.

Force at Bearing A.		Force at Bearing B.		Axial Thrust.	
Magnitude.	Direction.	Magnitude.	Direction.	Magnitude	
332	90°	1,081	90°	1,413	Thrust taken by separate bearing.
1,460	13°	1,081	90°	—	A takes thrust.
332	90°	1,780	37°	—	B takes thrust.

magnitude is difficult to estimate, as it depends so much upon the springing of the car, but I think they do not exceed three times the usual load on the wheels. These jolts reach their maximum when travelling very fast, and therefore in a straight course, with the load equally distributed. Suppose two-thirds of the weight is on the back wheels and one-third on the front wheels, then roughly each of the front bearings is subjected to a pressure of $3 \times \frac{1}{3} \div 4$ W. $= \frac{1}{4}$ W., which is insignificant compared with the pressures produced in turning corners. The back wheel bearings, if there are two to each wheel, bear pressures roughly equal to $\frac{1}{2}$ W. each, or if there is one bearing to each wheel as in Figs. VI. and VII., the pressure on each bearing due to jolt is equal to W.

With two bearings as in Fig. IV., the maximum pressure due to taking corners fast is 1.2375 W., and due to jolts is .5 W., while with one bearing as in Fig. VII. the maximum pressure due to taking corners fast is 0.447 W., and due to jolts is 1.0 W.

When it is recalled how much more easy it is to put one ample bearing in a hub than two ample ones, the case for the single bearing becomes overwhelmingly strong.

Apart from the favourable conditions under which the single bearing on a back axle works, there are other considerations, such as simplicity, and lightness, which

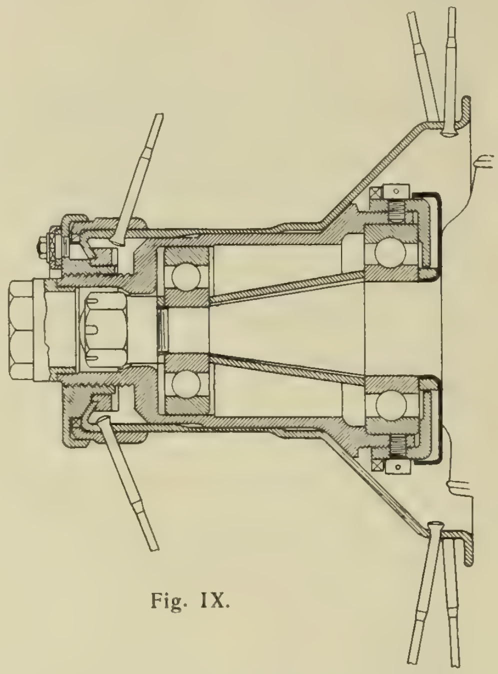


Fig. IX.

give it advantages over the double bearing which, in my opinion, it is destined to replace in a very short period.

It is naturally essential that the hub should be rigidly connected to the live axle, and I know of no method of attachment that gives anything like such satisfactory results as a single key and a taper of about 6° contained angle. No system of parallel fits can rival it, owing to the inevitable variations in manufacture which parallel fits have no means of accommodating.

In conclusion, I would add a little about keeping dirt and water out of bearings. The Continental plan (see Fig. VIII.) is simple, and, as far as my observations go, is exceedingly good; but while it excludes water it does not form a reservoir for oil, and wherever possible I advocate the use of the plan shown in Fig. IX., which retains a large quantity of oil or grease and prevents any water getting in. For experimental purposes I have been running a car with hubs so equipped, in which the bearings are very close together indeed (only 82 mm. between centres), and after eighteen months, and probably 15,000 to 18,000 miles, there is not the slightest sign of water in the hubs, or of shake in the bearings. Meanwhile, the hubs have been well supplied by means of a grease pump with grease of not too thick a consistency, and this has undoubtedly had something to do with the excellent results obtained.

HEATING AND VENTILATION OF WORKSHOPS.

THERE are one or two matters connected with heat which should be kept in mind in designing heating systems for works or dwelling houses. The action of heat is governed by well defined laws, and if we ignore these laws the result is not likely to be satisfactory.

Heat is transmitted in three ways, viz., by conduction, convection, and radiation. It is said to be conducted when it passes from the hotter to the colder parts of a body, or from one body to another in contact with it. By convection, the air in contact with any hot surface has its temperature raised, and consequently expands and becomes lighter—or less dense—than the surrounding air, therefore it rises and carries with it the heat that has been imparted to it, and is, of course, replaced by a fresh supply of cold air which goes through the same process. A text book definition is that heat is transmitted by convection when material particles conveying the heat are carried from one point to another. It is important to understand the effect of radiation of heat, the chief point to be noted in dealing with heating apparatus being that heat rays, or more correctly heat energy, passes from one body to another through air with very small heating effect on that air. Hold a thermometer in front of a fire and even quite a long way off it will indicate a rise of temperature which may be very considerably above that of the surrounding atmosphere. This elementary fact is not perhaps so widely known as it might be, but it is a fact, and applied to the matter of heating apparatus we arrive at the conclusion that air can only be heated by direct contact with a body hotter than itself. Heat units figure in the problem to a considerable extent, and also latent heat—in special relation to steam heating plants—but to engineers it is hardly necessary to define these terms.

In a certain building with a saw tooth roof, radiators were placed at the bottoms of the ridges. What result might we expect from this arrangement? If there were abundant radiator surface, and if it were kept very hot, a current of heated air would rise from the radiator, and being cooled by contact with the cold glass, would be again deflected downwards and would circulate only within the upper air of the shop. In the particular case referred to the radiating surface was totally inadequate, and the heat supply was insufficient, so the only effect was to set up a much smaller area of circulation, which was almost entirely useless, being practically confined to the teeth of the roof. Would the effect have been any better had the radiators been placed at floor level? We should expect it to be a little better because the air heated by the radiators would have to rise to the glass before being actively cooled down, and this would render impossible any small local circulation as in the first case, also the men would profit by the radiated heat. This latter, however, is by no means desirable, for if a man is in a position to be the recipient of radiated heat, the probability is that he will be getting it on one side only, whilst the side which is away from the radiator will be cold, and this is a fruitful cause of rheumatism.

Whatever scheme is adopted, the object should be to warm the workmen by raising the general temperature of the air in which they are working, and not by local heating as from a radiator. And an important point, too, is to provide for warm feet, indicating that a current of warm air should be distributed at the floor level. If we have radiators at or above the floor level, the air which they heat rises vertically upwards, and a current of colder air flows along the floor to take its place. If we could put the radiators below ground, and bring the heated air up through gratings on which the men stand, the position would be more satisfactory, but it is safe to say that in most cases such an arrangement is impracticable.

Of course, if we put in sufficient radiating surface and keep it hot enough, the heated air will not be as cold when it drops down again, after being cooled by contact with the roof, as it was originally, and in time the general atmosphere of a shop will be increased in temperature. It is obviously wrong to put in an apparatus to heat air which—owing to the natural laws of convection—cannot be made use of until it has gone up to the roof, where it is not wanted, and become cooled down again. The efficiency of such an arrangement can hardly be ideal, and as it provides a strong inducement not to let any of the heated air escape, or let any fresh cold air in, the result from a ventilation point of view is bad.

Evidently, convection would be better if worked the other way about, but as that is not to be hoped for, the only thing to do is to use force and make the hot air go the way we want it to go by means of a fan.

Some heating engineers are opposed to the use of fans, because it involves the cost of power to drive them, whereas if natural circulation is depended on this permanent charge is avoided. But it must be remembered that the power required is comparatively small, and as it

enables far more efficient use to be made of the heat, there will be savings in fuel and in the size of apparatus to set off against the power cost. The use of a fan also very considerably facilitates the treatment of the ventilation question, which should be treated as an integral part of any heating scheme. If this is not done the probability is that a separately designed ventilation system will be found to interfere with the heating system and spoil its efficiency, or *vice versa*.

Different Heating Systems.

Although there are quite a number of so-called different systems in vogue, there is really only one principle which is common to them all, i.e., to heat the air of our workshops by bringing it in contact with some convenient heat-giving apparatus, and then keep it circulating in the most efficient way so as to maintain an equable temperature throughout. The different systems only refer to the particular apparatus by which the air is heated and may be divided into three groups, viz. :—

1. Hot water.
2. Steam.
3. Direct stove heat.

In considering which of these systems it is best to adopt in any given circumstances it will be found that convenience almost more than efficiency has to be considered. In dealing with an entirely new factory in which a heating scheme can be planned and executed before any plant or shafting are introduced, the case can be treated on its merits, but where a plant is already at work it is obvious that the most attractive system would be that which required the least amount of disorganisation for its installation. Considered as "Systems" it might be said that there are only two, viz. :—(1) That in which the air is heated by means of pipes or radiators distributed throughout the factory, and (2) That in which the air is heated at central stations and distributed through pipes or ducts.

For the former, both steam and hot water can be used with practically equal convenience. Stoves distributed throughout the shops, can, of course, be used also, but as there are strong objections to their use which are too obvious to dwell upon, it is not necessary to do more than mention them. Direct stove heat is the most efficient and economical for the second system, and when properly carried out so as to avoid excess heat which would over-dry the air, it is undoubtedly one of the best possible arrangements.

Hot Water Systems.

Where the arrangement of workshops favours the adoption of heating by convection or natural circulation, hot water apparatus may be most convenient. There are three principal ways in which it can be installed, viz. :—(1) with large diameter pipes (about 4 in.) which form a continuous radiating surface—practically 1 square foot of surface per lineal foot of tube. (2) With small diameter pipes, and radiators distributed as required, the water being subject to ordinary atmospheric pressure in each case, so that its temperature is not raised above boiling point.

Another system works with high pressure water, and this will be dealt with presently. Number I, with the large pipes, we have long been familiar with, as for many years it has been practically the only system used in greenhouses, for which it is eminently suitable. For workshops it has the serious disadvantage of being very obtrusive and in the way, and also it does not favour an even distribution of heat, as too much is given up near the boiler whilst outlying districts are left cold. Owing to this it is not much used except for small shops or where circumstances are exceptional.

Number 2 system, with small bore pipes and radiators is much to be preferred, and its use, more especially for offices or domestic purposes, is greatly on the increase. In this system heat is meant to be given off by the radiators only, and not by the pipes which supply these with hot water—in fact, it is found desirable in places to lag the pipes in between, so that they do not lose more heat than can be avoided. Pipes vary in diameter, according to circumstances, and may be from 1in. to 3in. or more, and valves are provided, by means of which the supply of heat to each radiator can be regulated, so that it is easy to keep up the temperature in those which are furthest removed from the boiler. A single distribution can be used, which acts for both supply and return. Although it is necessary to arrange this pipe with a gradual rise to a certain highest point—preferably somewhere at about the middle of its length—and from there with a gradual fall back to the boiler, and also to have all the radiators at a higher level than the pipe, it can be understood readily that such a pipe must be easier to instal, and much less in the way than a pair of 4in. diameter. The system has another advantage also, in that the radiators can be wholly or partly enclosed in a ventilation opening, thereby supplying the shop with warm, fresh air. Suitable shutters may be provided by which the fresh air can be regulated or entirely cut off.

The third system works on a different principle to either of the foregoing, in that while in operation the water is under considerable pressure. It is well known that under atmospheric pressure water boils at 212° F., and we cannot raise its temperature beyond that, unless we prevent the escape of steam which begins to be given off after that temperature is reached. As the pressure on the water increases, so does its temperature, and it is this fact that is made use of in the high-pressure water system of heating, only there is no steam generation allowed in it. The fact is made use of that when water is heated it increases in bulk considerably; for instance, when raised from 32 degrees to 212 degrees the increase in bulk is 1-24th, and if raised from 212 degrees a further 130 degrees up to 342 degrees, the bulk will increase for this 1-32nd. In designing high-pressure systems the pipes are arranged to be filled completely with cold water, and at the highest point, above the water level there is fixed a small air compression chamber in the form of a slightly-enlarged vertical pipe, the whole system being then tightly sealed. It will be understood that the increase in bulk of the water that takes place when heat is applied forces it up into the expansion chamber, and compresses the air, which forms a pneumatic cushion on top of it, and which entirely prevents any generation of steam. In this manner the pressure can be raised to almost any degree, but in practice about 600 lbs. per square inch, corresponding to a temperature of 484.5° F., is considered to be the maximum permissible. An automatic valve is arranged to come into operation when the desired pressure is exceeded, which permits some of the water to escape, and so prevents further rise. The water so escaping is sucked back into the system by the vacuum which forms when shrinkage of the water in the pipes takes place on cooling. In the high-pressure system the high degree of heat attained makes it very rapid in its effect; small pipes are used, made specially for the purpose, of one dimension throughout, viz., ⅞in. bore by 1 5-16in. outside diameter. For joining up, the ends are screwed right and left-hand threads, and one face is made flat, whilst the other has a circular V edge, so that when drawn up this bites into the other and makes a tight joint without any packing. In order to ensure rapid circulation, short pipe runs, not exceeding 300 feet to 500 feet, are found convenient. The boiler consists of coils of tubes in multiple sections, but working in series. Water heated in one section goes out and loses its heat in one run of pipes, but on its return is heated in a different section of the boiler. There is only one expansion chamber, and there are no regulating valves and no branch services further than what are provided by the different runs of pipe as described above, so the system is commendable for its simplicity. The chief advantages lie in the much higher temperature at which the water can be circulated, resulting in higher efficiency, and the very small size of the pipes, which permits of their being used in situations where larger ones would be out of the question. The system is very swift in its effect as there is a very small body of water to heat up and circulation is therefore rapid. Disadvantages are that there is a strong liability in winter for the whole thing to freeze solid, and though it is possible to use a non-freezing solution, this remedy is often worse than the disease, as crystals are liable to form during the summer, closing the pipes, and resulting in a burst in the grate; and, as in the case of the 4in. pipes, all the heating must be effected by the surface of the tubes, so that unless these are kept cleaner than is likely to be the case in a busy factory, the efficiency fails. The system does not lend itself very readily to easy ventilation, in which respect it is on a par with the continuous large pipe system.

Heating by Steam.

Steam for heating purposes is utilised in three ways, viz., high pressure, low pressure, and "vacuum," of which the latter is claimed to be the most efficient. In the application of these, very similar apparatus is used for the heat distributing as in the case of hot water, *i.e.*, for natural air circulation systems, but steam can be very conveniently used in connection with forced-air systems, which will be described later on.

Steam is more convenient in some ways than water, the principal advantage being

that the pipes conveying it can be arranged at any desired level, and do not require to be all on the same level. Provision must, of course, be made for dealing with the water produced by condensation, the pipes for which should be arranged to drain by gravitation. These pipes, however, are small, so are not difficult to deal with. What makes steam such a useful heating agent is the fact that it possesses or carries a large store of thermal units in the shape of latent heat, and these it gives up where it is reconverted into water. Now as a British thermal unit is the amount of heat necessary to raise the temperature of 1 lb. of water 1° F., if we raise a gallon of water (10 lbs.) from a temperature of 50° to one of 200°, we impart to it $150 \times 10 = 1{,}500$ heat units, and if in one of the systems described this hot water is circulated and comes back to the boiler at 50°, it will have delivered up these 1,500 units to the heating of the works. That is the limit of the heat-carrying capacity of the water, but it is seldom the return water would drop to its original temperature, and so the average capacity may be taken at a much lower figure. Now, on the other hand, a pound of steam at, say, 30 lbs. gauge pressure, has a temperature of 273°, which, assuming a temperature of feed water of 50° gives a rise of 224°, but this pound of steam contains, in addition to these 224 units, latent heat to the extent of 921 units (which it absorbed during its conversion from water), or to compare it with the foregoing, a gallon of water of 40° turned into steam of 30 lbs. gauge pressure, will contain a total of added heat of $1{,}145 \times 10 = 11{,}450$ units. On re-converting this steam into water, all the latent heat must be given up, so that for each 10 lbs. of steam condensed we set free $921 \times 10 = 9{,}210$ heat units. That is assuming that the pressure of 30 lbs is maintained throughout the system, in which case the heat that is still contained by the water of condensation is not made use of for external work, but is returned to the boiler. It should be noted that the amount of latent heat in a pound of steam at a gauge pressure of 1 lb., is 962 units, and that if we go below atmospheric pressure the proportion is still higher. Therefore, except that higher pressures of steam can be distributed through smaller sizes of pipes, it is more efficient to use low or even negative pressure.

This is particularly useful to remember in cases where exhaust steam from an engine can be utilised. Of course, to attempt to force such exhaust through a long system of heating pipes would set up considerable back pressure, even if the pipes were made of large area, but if a suitable air pump is installed, this difficulty is got rid of, and an extremely economical heating system is the result. The effect is to turn the heating system into a surface condenser, in which the latent heat of the exhaust steam is converted to a useful purpose instead of being thrown away and wasted. Where, therefore, this source of heat is available it will be worth while to make use of it as far as it will go. It is not to be understood, however, that the vacuum system is used only in connection with exhaust steam from an engine. It can be applied equally usefully when steam is fed into the heating mains direct from the boiler, and it is found greatly to improve the general efficiency of such systems. Steam, as is well known, contains a good deal of air, which in a pressure system is apt to collect in various parts of the piping, and interfere with the working. One of the features of the vacuum system is that this air is removed without the aid of objectionable automatic valves. Further advantages of the vacuum system of steam heating are that pipe joints give less trouble, and that there is no water hammer action at all, both of these joints being liable to cause trouble when high pressure is adopted.

As regards ventilation, steam-heated radiators are in practically the same position as those heated by water, except that size for size the former would be more efficient.

The Forced Air System.

This, sometimes called the "Plenum" system, is perhaps the most satisfactory all round for the heating and ventilation of workshops. The essential feature is that pure cold air is drawn from outside the building, heated by contact with hot pipes, or other hot surfaces, and drawn by powerful fans into the various shops and departments through air trunks or underground passages, suitable outlets being provided for the vitiated air, which the fresh supply displaces. This may be carried out by means of one large central heating station, or by smaller units distributed in convenient situations throughout the works. In a comparatively recent modification these units are quite small, and entirely self-contained, requiring no air trunks or passages at all; this will be described presently.

Whatever be the heating system adopted, it is very essential to have a complete circulation of air without any perceptible draughts, and for this reason, as also for the purpose of reducing frictional resistance, it is necessary in the "Plenum" system to have distributing pipes of large area, as may be judged when it is stated that in some cases the total cubic contents of a building are completely changed two or three times in every hour. Quite a moderate-sized works contains over half-a-million cubic feet, so the size of pipes suitable to deal with such quantities can easily be imagined. Obviously, conditions of space, especially in existing factories, may render the installation of such large pipes impossible, in which case the smaller heating stations may be necessary, but it goes without saying that a single large station must be more economical, other things being equal.

The most efficient method of heating air is doubtless to bring it in contact with the direct heat of a stove provided, of course, that the apparatus is suitably designed. In the combustion of fuel a certain definite quantity of heat units are liberated, and if the apparatus is not designed on scientific lines, a large proportion may be wasted.

Messrs. Davidson and Co., have made a speciality of stoves for the purpose, and claim to capture from 80 to 85 per cent. of the total units of the fuel. The firm have carried out some installations of great magnitude, in which most elaborate provision is made by means of filters and water sprays for thoroughly freeing the air from dust and impurities, and providing it with the requisite degree of moisture before it is heated. The stoves consist of a central furnace, from which the products of combustion are led through a series of tubes, arranged horizontally, on either side. The design is such as to cause the hot gases to travel a long distance through the pipes before escaping to the chimney, so that a large proportion of their heat is taken up by the tubes and transferred to the current of air, which is forced to pass around them. The high degree of economy attained will be understood when it is said that it is possible to maintain, with these stoves, the temperature of a building at 30° above that of the outside air with a fuel consumption of 1 lb. of coal per hour for every 10,000 cubic feet of air contents. Although this direct heat stove, as it may be called, is the most economical means of heating air, circumstances may be such as to favour the use of a steam heater in preference. If, for instance, there is a steam plant on the premises, with a considerable margin of power, or the exhaust from one or more engines, a steam heater might be preferable. And again, if the conditions favour the installation of several smaller units instead of one central heating station, the units can be more compact if heated by steam, which also avoids the dirt, inconvenience and extra labour involved by the use of direct heat furnaces.

If there is a steam supply available, these smaller sized heating units, provided with fans, and of course much smaller air ducts, may be found extremely convenient. The latest development in this direction is what is called the "Stanlock" multiple unit system, made by the Standard Engineering Co. This arrangement consists of vertical type heaters composed of solid drawn steel tubes screwed into a circular cast iron base provided with suitable steam supply and drain pipes, and an automatic air valve. Enclosing the steam tubes is a sheet metal cylindrical casing coming to within an inch or two of the top of the base, and provided at its top with a small electric motor-driven fan, and also suitable air connections and dampers whereby either a supply of fresh outside air can be introduced, or the air already in the building can be circulated. This last is useful for rapidly increasing temperature when starting up in the morning, and afterwards the dampers can be regulated to admit any desired proportion of fresh outside air, and so maintain efficient ventilation. It is to be noted that the heated air is forced downwards, so it spreads over the floor and keeps the workers warm and comfortable about their lower extremities; a most desirable feature, as previously mentioned. These units, from their vertical shape, occupy very little space, and seem to be admirable in every way, requiring very little power to work them, and no ducts or air trunking beyond a small branch through the roof for each unit. The makers estimate that with steam at 60 lbs. pressure the units may be placed 60 ft. apart, so that each provides for the heating of 3,600 sq. ft. of floor space. Provided with a suitable trap, the water of condensation can be forced up and returned to the boiler from a high level, so that there is no

need to cut up floors to provide for return water pipes, and this enables the system to be adopted in existing factories where otherwise its use might be barred. Altogether this arrangement appears to be highly convenient for existing works where space is a consideration, owing to its extreme compactness and the facility with which it can be installed.

Amount of Heat Required.

Here are many pitfalls, because there are so many factors to take into consideration. A basis for calculation is provided by the knowledge that one thermal unit will raise the temperature of 50 cubic feet of air one degree. If, therefore, we know the total cubic contents of a workshop it is easy to say how many thermal units must be provided in order to raise its temperature from any point to any other point, provided there were no loss of heat, but, of course, that is an impossible supposition; and it is in the proper estimation of these losses that mistakes are liable to be made. The walls, the floors, the roofs, or ceilings, the amount of glass, the number and size of doorways, and the frequency and length of time they are kept open, and finally the proportion of the contents to be renewed hourly for ventilation, all require to be carefully weighed up in relation to the total cubic contents of the buildings. It need hardly be said that these different factors do not always carry the same value. Walls vary in their cooling effects a good deal according to their thickness, and somewhat also according to their degree of exposure, and the same may be said about most of the other points. It is estimated that a square foot of glass is capable of cooling about $1\frac{1}{4}$ cubic feet of warmed air down to the temperature of the outside air in one minute. Floors should be carefully considered, as manifestly the cooling effect of damp concrete and brick will be very much in excess of close boarding with an air space underneath. Machine men should never be allowed to stand all day on concrete floors. The cost of a grating or standing board is very little compared with the good it does in keeping a man comfortable. For comfort and efficiency almost always go together.

It is very difficult to lay down any general rules in regard to this matter, and generally it is better to take the advice of specialists who have learned by experience what allowances to make. But if a word of advice might be uttered, we should say, see that it *is* an expert that is called in and not simply a self-constituted one.

Ventilation.

So far nothing has been said regarding ventilation as separate and distinct from heating, and it is really impossible to consider the two apart, because they must be arranged so as to cause as little interference as possible with each other. Summer ventilation is not usually a problem at all, in engineering works, because the premises are but seldom of such a nature that they can become stuffy if there are a reasonable number of roof lights that are capable of being opened. It may, however, be mentioned that, in designing a new building, it is well worth while to give some attention to the exact situations of the windows and doors which are most likely to be opened, because it will frequently be found to be possible to arrange the openings so as to obtain the maximum of true ventilation with the minimum of draught.

It will therefore at once become obvious that a great deal will depend upon the style of the shops. Automobile works are usually of very large capacity in proportion to the number of hands employed, so much so that in some departments it is hardly necessary to make any special provision for ventilation; and this would apply especially to modern buildings all on ground level. Where there are upper floors the conditions will be more air-tight, and in such case provision should certainly be made to renew at least one-tenth of the total cubic contents every hour. In paint shops, where there are injurious fumes to be got rid of, and small rooms where the proportion of hands employed is in excess, not less than half the contents should be renewed hourly. A more or less rough and ready guide for the overall capacity of a heating and ventilating plant would be to allow that the cooling effect of the total interior surface of walls, roofs and floors will average about one quarter of what it would if the whole were of glass, and add one-third extra for ventilation, but it will be obvious that this is no more than a very approximate calculation.

From a consideration of what has been described it will be seen that a "natural air circulation" system of heating is from its very nature not the one that gives either the most satisfactory or most efficient results; that if circumstances necessitate this system being adopted, the most economical results will probably be retained by the use of a low pressure or "vacuum" steam arrangement. The forced air of the Plenum system would appear to be all round the most economical and satisfactory, and in this description we include modifications such as the Stanlock, but whether steam heaters or direct heat stoves are employed must depend entirely on local conditions and circumstances.

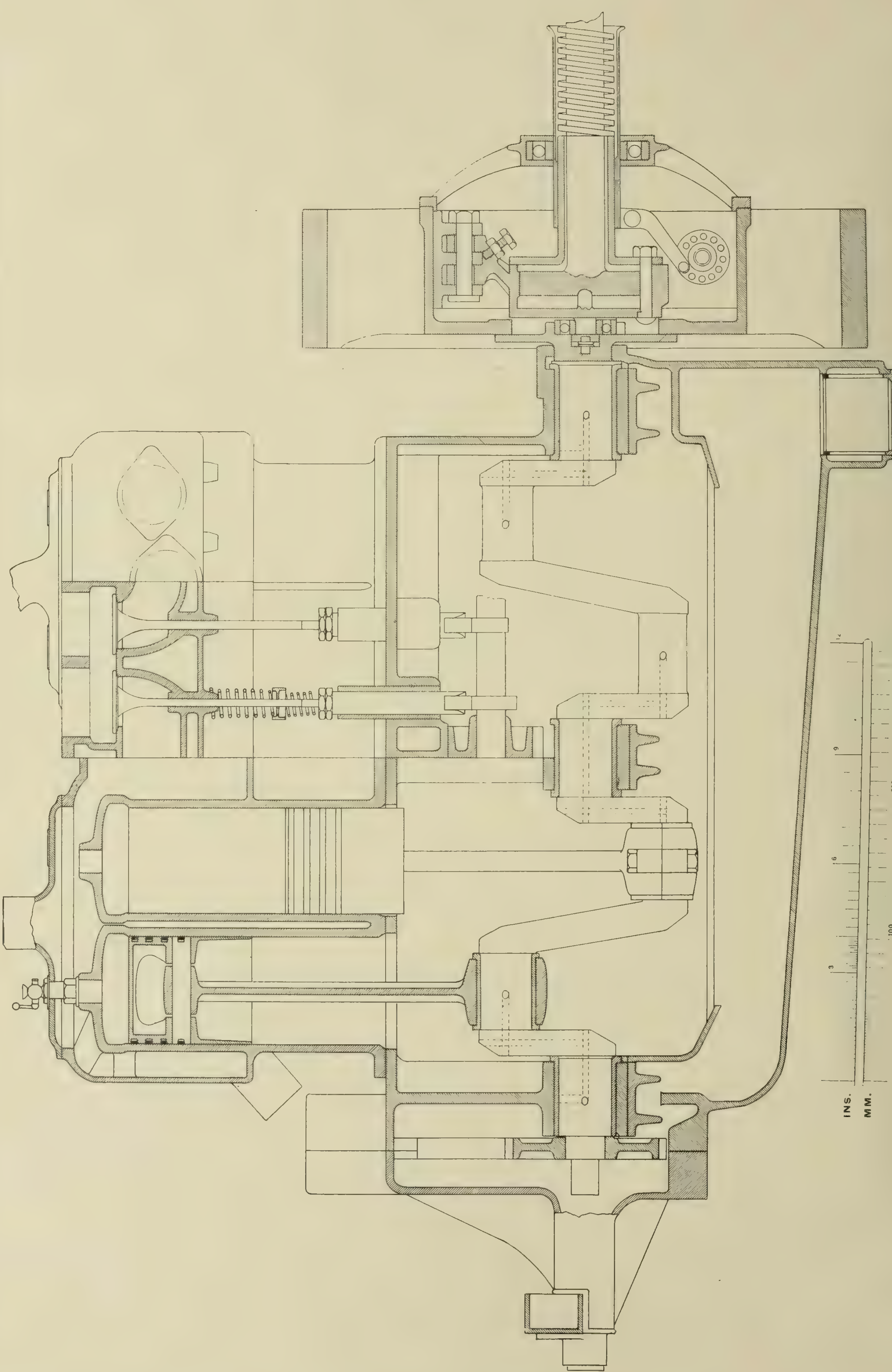
THE 14 H.P. METALLURGIQUE CHASSIS.

This chassis possesses several features entirely peculiar to itself.

JUST as there is an indefinable character peculiar to Italian-made or Italian-designed chassis, so is there a peculiarity about those which originate in Belgium, and the Metallurgique may be taken as typical of the products of that country. Belgian manufacturers share with the Italians the advantage of inexpensive skilled labour, and therefore in a quite general sense, it is not surprising to find that they should endeavour to make the most of lowness in production costs by selling their chassis at a low price also. Thus it is but seldom that a small Belgian car is supplied with those fittings which are between the necessary and the luxurious—optional fittings, such as fourth speeds, ignition point control, and so forth—and the 14 h.p. Metallurgique shares this common feature, though not to a very great extent. Criticism of the small short-comings is difficult if the selling price is kept in mind, but as they are not serious, and mostly faults of omission rather than doubtful practice, they will be referred to in due course, without regard to monetary considerations.

The engine is at present undergoing some slight alterations, and the section illustrated shows the late 1910 type, which is a little different from the earlier model in that the lubrication is forced, and the tappet arrangement has had to be altered slightly—to allow a passage for the oil to be bored through the aluminium between the camshaft and the cylinders. Also the exhaust passage was cast in the main block until lately, but a loose exhaust pipe, with four flanged joints, is now employed. The cylinders are practically two pairs cast together, there being a large central space to permit the use of a central bearing of fair length, and this gives a large central water space, though there is but a small one between the cylinders in each pair; still, the jacket spaces are all large as compared with the cylinder bore. As far as the water connections are concerned, the engine is arranged in a manner precisely similar to that usual for pair-cast cylinders, the inlets and outlets being in duplicate, so it will be seen that the fullest advantage of the monobloc has not been taken. Small openings are left at the top of the combustion chambers, to be closed subsequently by brass plugs bearing compression taps. It should be noticed that there are a number of webs connecting the cylinders and water jacket walls, and that the openings in the jackets are about equal in area to the cylinders beneath them, being closed by the domed feet of the brass outlet piping. This piping is rather above the average of size for that of the cylinders, even though the cooling is accomplished by natural circulation, the internal diameter of the intakes and outlets being 30 mm. at the cylinders, and 40 mm. at the radiator. There is also a good head of water, the depth from the bottom of the outlet to the radiator down to the top of the cylinders being 170 mm. The V-shaped radiator is probably too well known to need description, though it may be remarked that its shape gives easy access to the front end of the engine, enabling the oil pump to be detached with a minimum of trouble. The pistons call for no comment, as their light section is shown clearly by Fig. I. Four rings are used, the lowest one of them securing the gudgeon pin, and acting to some extent as a scraper, to prevent over-lubrication of the cylinders.

Owing to the moderate overall length, the three bearing crankshaft is sufficiently stiff to give reasonable bearing durability, and it is, of course, bushed with white metal. It will be seen that it is offset 25 mm. from the central plane of the cylinders, and the makers claim that this enables enhanced power to be obtained;



THE 14 H.P. METALLURGIQUE ENGINE.

it does not tend to improve the balance of the engine at high speed, but its effect in this respect is, of course, extremely small. The main journals and the big ends are 37 mm. in diameter, and the former have an aggregate length of 175 mm., the big ends being each of them 54 mm. long. There is a flange at the rear end of the shaft, for the attachment of the flywheel, but the shaft is not extended through the clutch, the latter being an entirely separate part, practically complete in itself.

A solid camshaft is used inserted endways with its bearings, in the customary manner, and the oil pump is situated at the forward end in a brass casing, easily detachable when necessary. It is an ordinary gear pump of moderate dimensions, sucking oil from the crankcase sump and forcing it to the main bearings, whence it passes to the big ends through the crankshaft. There appears to be no arrangement for priming the pump, though this is probably but rarely necessary, as the pipes are small, and a moderate amount of oil would be contained in the lead which is connected from the rear end of the main channel to the Bourdon gauge on the dashboard. However, a small filling orifice above the pump could not fail to be an advantage. The filtering arrangements may either be a cylindrical gauze, situated in a chamber at the rear of the sump, and so placed that it can be withdrawn from beneath, or a simple gauze tray may be used, covering the sump, just beneath the big ends, and in this case there is no additional detachable filter. The timing gears are arranged in the usual position at the forward end of the crankcase, a separate large wheel being used for the magneto, the position of which is on the off side of the cylinders.

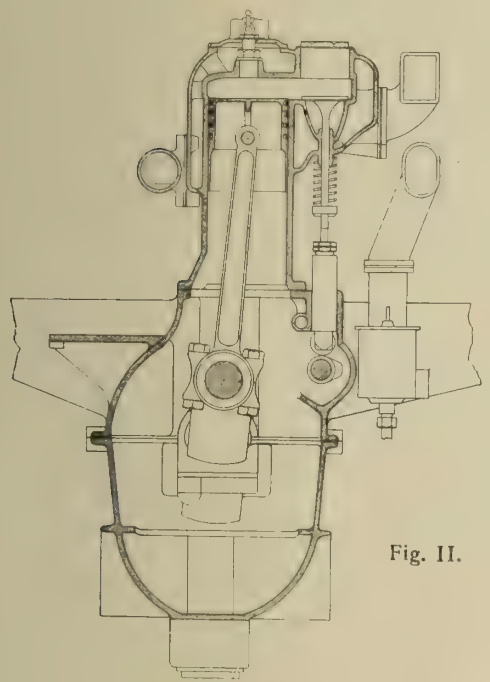


Fig. II.

Unusually large valves are employed, the dimensions being 45 mm. outside head diameter, with an effective diameter of about 35 mm. The valves are light, and the angle of the seating is extremely small, being but little removed from the flat. It should be noticed that strength is obtained by the use of a very large radius where the head joins the stem. Roller-ended tappets are used, and their rotation is prevented by a feather, which slides in a keyway cut in the phosphor bronze guides; a set screw tappet head with a locking nut is provided, and there is a small spring, normally in compression, connecting each valve stem and tappet head. To retain these small springs each valve stem has a groove turned in its lower extremity, and the spring clips into this groove, the purpose of the springs being to silence the action somewhat.

Four crankcase arms, of the usual pattern, secure the engine directly to the frame, these arms resting on hollow packing pieces which lie inside the channel of the side members of the frame, the bolts passing through both packing pieces and crankcase arms horizontally; the whole making an extremely neat job, convenient from the erecting shop point of view.

The flywheel is 400 mm. in diameter, having a rim 98 mm. by 18 mm. thick attached to the outer member of the clutch by six vaned webs, and the clutch itself is probably the most interesting part of the chassis. Its construction is explained partly by Fig. I., but more completely by Fig. III., which is a perspective view of the similar type of engaging mechanism that is fitted to the 40 h.p. Metallurgique. The manufacturers of the Metallurgique are the most notable of the extremely small number of automobile makers who make use of an internal expanding clutch, and the fact that they have continued to use it for a fair number of years, on all types of chassis, is sufficient proof that it has not proved unsatisfactory. It is really a surprising thing that this type of clutch has not found a greater number of adherents, as it has several points of advantage peculiar to itself. Firstly, it is practically devoid of end thrust, and secondly, it easily provides a large, and therefore durable, surface. It does not require copious lubrication, it has no tendency to stick in engagement, and it is easy to provide an adjustment for taking up such wear as does occur. Thus it has most of the advantages of the disc or plate clutch, without its principal disadvantages. Its bad points are that it is rather complicated, and therefore not too cheap to produce, and it is also rather heavy, though this is not of extreme importance, owing to the ability to make it of small diameter. In action it is not as gentle as a multiple plate clutch, and it has practically no automatic slip, but it can be let in more gently than a leather clutch, and is not subject to the troubles that are liable to arise with a plate clutch when it is supplied with unsuitable lubricant.

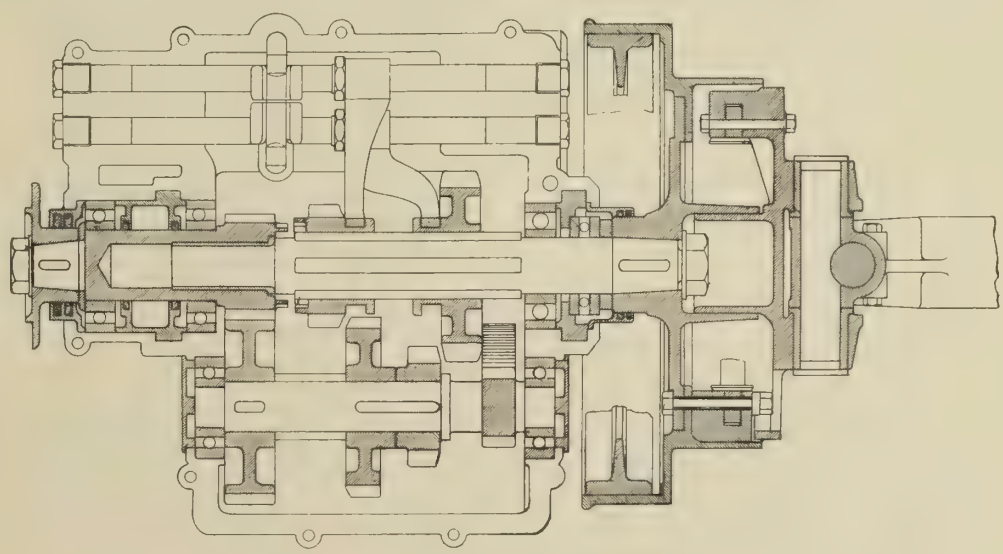


Fig. IV.

The Metallurgique clutch shoes may be either phosphor bronze or iron, the latter material being chosen for the 14 h.p. The shoes are capable of sliding radially on the two main arms of the driven member, to which they are secured by ordinary plain pins of large diameter, which may be seen clearly in Fig. III.

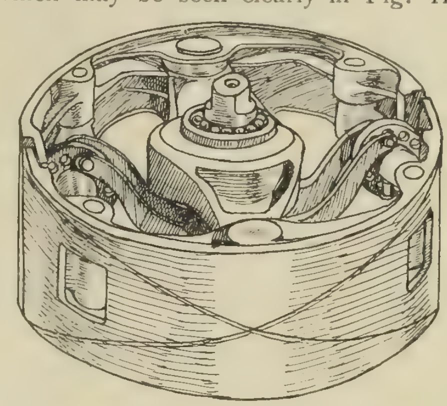


Fig. III.

This portion of the mechanism is shown in the top half of the clutch in Fig. I., and it should be explained that this view shows two sections at right angles. The shoes are connected to each other by means of a pair of short shafts, threaded right and left-hand at their ends, and provided with central discs having a ring of holes drilled in their periphery, so turning these discs expands or contracts the clutch. There is a sliding sleeve operated by the clutch spring, which carries a pair of lugs, that are connected to the discs mentioned above, by short toggle levers, and it can be seen from Fig. I. that expansion of the spring causes revolution of the discs, and so expansion of the clutch, while adjustment can be made by changing the position of the joint pins on the discs. A separate shaft is used for the whole clutch, and is carried in a spigot bearing recessed in the back of the flywheel, and in another ball race carried in a four-armed aluminium ring bolted to the flywheel and also shown in Fig. I. This ring serves to retain the small amount of lubricant necessary. In the top half of the clutch, as shown in Fig. I., there is a small set screw and lock-nut, and this, of course, serves to set the degree of disengagement. Operation

is by the medium of a pair of rollers, which bear against a disc on the lever sleeve, and there is a small end thrust on the shaft when the clutch is held disengaged.

The clutch shaft is connected to the gear shaft by a short coupling shaft hav-

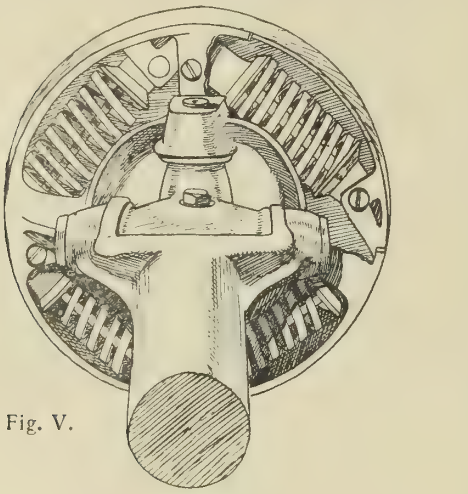


Fig. V.

ing a plain flange at its forward end, which is bolted up to a similar flange on the clutch shaft. At the rear end the coupling bears another flange, but this is star-shaped, and fits loosely in a corresponding recess in the box attached to the gear shaft. The purpose of this arrangement is dual—it allows the clutch to be dismantled without interference with the gearbox, and it also permits a certain limited amount of universal movement. Presumably it is cheaper than most other patterns of universal joint, and it is more easily taken apart, but if called upon to run out of line for any length of time, its durability would probably be found to be poor. Owing to the method of hanging the gearbox it must be difficult to ensure the absolute alignment with the engine, and so it would possibly be an improvement to use a better type of joint.

Almost needless to say, there are only three forward speeds, which is to be regretted, as there is no form of chassis which derives more benefit from the fourth speed than that with an engine giving about fifteen normal horse power. Still this is doubtless one of the omissions

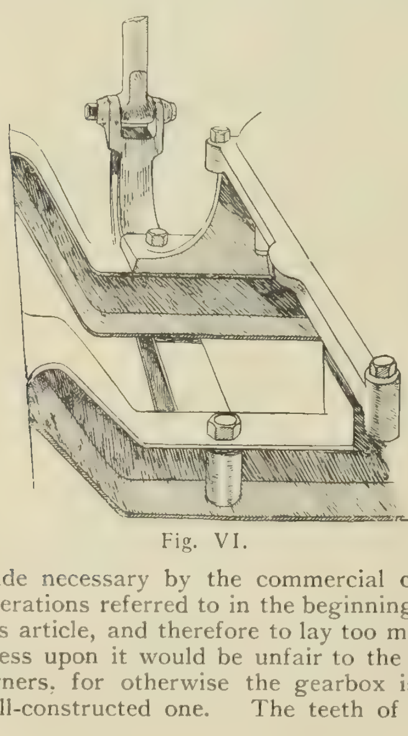


Fig. VI.

made necessary by the commercial considerations referred to in the beginning of this article, and therefore to lay too much stress upon it would be unfair to the designers, for otherwise the gearbox is a well-constructed one. The teeth of the wheels are Module 4, and the ratios are as follows:—First speed pair 15 to 27, second speed pair 23 to 19, and permanent layshaft drive (teeth Module 3) 18 to 38. This gives speed ratios of 3.8 to 1 for the first speed, and 1.8 to 1 for the second speed, for the gearbox alone, and the final drive may be either 13, 14, or 15 teeth in the pinion to 50 in the crown wheel, the pitch being Module 7. Assuming the 14 to 50 ratio the complete ratios from engine to road wheels are 13.6 to 1, first speed; 6.4 to 1, second speed; and 3.5 to 1, top speed. A good point in connection with the bearing arrangements in the box is that the two races at the forward end of the main shaft are spaced well apart, as this renders unequal loading of the bearings unlikely to occur. The thrust bearing to resist any end stress from the propeller shaft is also a commendable and unusual fitting. An equally unusual feature is the mounting of the sliding gears on four long feathers instead of on the six more commonly employed with a splined shaft, and this construction does not give so

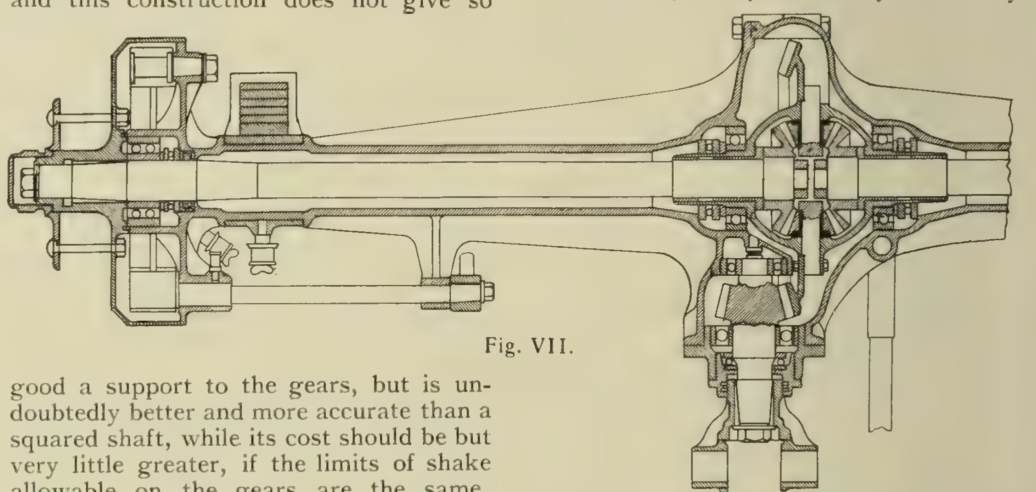


Fig. VII.

good a support to the gears, but is undoubtedly better and more accurate than a squared shaft, while its cost should be but very little greater, if the limits of shake allowable on the gears are the same. Locking of the striker shafts is performed by an ordinary spring catch and notches, and the control is by means of a plain gate and a lever with a sliding tube mounting. The outside portion of the spring cushion and the foot brake drum are made in one piece, and this is attached by bolts to a flange secured to the tail end of the gearshaft, the result being extremely neat, but making the renewal of the brake shoes when necessary a decidedly awkward job. It would seem that there might be some advantage here in making use of the full width of the spring drive casing, and the space between it and the gearbox, for the drum of an external brake, but the frame design makes the support of external shoes a considerable undertaking.

The spring drive, or cushion, consists of four flat coil springs connecting the outer drum with the forked inner member, and the arrangement is explained fully by Figs. IV. and V. Great claims are made for this device, but though it may assist towards the prevention of chatter when using the transmission brake, the range of movement is so small that it can have but little damping effect if the clutch is engaged fiercely or the brake applied with violence.

Both the universal joints are of the same size and pattern, and telescopic motion is allowed to take place between the splined front end of the propeller shaft and the universal fork. A special screw-down grease cap is fitted to ensure lubrication of this slide but it does not supply any of its contents to the pins, which depend upon the grease contained in the wrapping. This wrapping is most frequently a simple leather stocking suf-

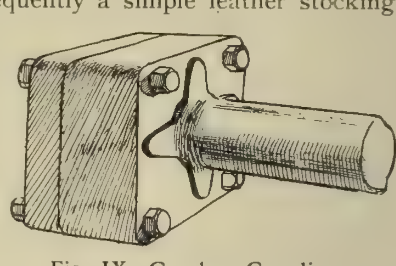


Fig. IX. Gearbox Coupling.

fering from the defects of all its kind, but a metal cap can be fitted over the front joint. In neither case would the lubrication be ideal, as the joint bearings are of a type requiring oil rather than grease, if the latter is not supplied by direct leads to the surfaces. Certainly each pin end is covered by a cap screwed into the bush, but to remove the whole eight caps (with a screwdriver) and to oil each pin separately is scarcely the variety of task in which the private owner delights, especially when the body is fitted.

The motion of the back axle is controlled by a triangulated torque rod mounted in a peculiar manner. The gearbox is supported on two deeply dropped cross members, which pass beneath it, and a heavy malleable casting is bolted up beneath them as shown in Fig. VI., the usual spring ball joint being mounted on the end of it. Although the triangulated torque rod is not an ideal method of controlling, the axle that is a matter which cannot be gone into here, and it was dealt with fully in the June issue of "The Automobile Engineer," but this elaborate bracket enables a cross member to be dispensed with and so makes this portion of the chassis more accessible from above, which is a good feature.

The rear axle is shown in Fig. VII., and is both simple and strong. In so light a car there is no object in increasing the cost by mounting the hubs outside the axle sleeves, the driving shafts being amply strong for the purpose. This method of construction enables the axle case to be a simple iron casting and it may be noticed that the whole of the machining of each half of the case could be performed in a lathe, though it would probably be better practice to bore the housing for the level pinion bearings. As a whole, the axle is probably slightly heavier than a steel axle of similar strength, but the difference would not be very great in so small

a size, and would be out of proportion to the increase in cost. The size and number of the ball bearings is also a good feature of this axle, and the ample provision of thrust bearings is particularly commendable. Four pinions are used in the differential, on the usual spider, and each is bushed with phosphor bronze, shown in black in Fig. VII.

The frame is extraordinarily heavy for so small a chassis, as may be gathered from the principal dimensions, which are as follows :—Material 4 mm. in thickness. Section at gearbox centre line 100 mm. deep and 90 mm. wide. Section at front end 33 mm. deep and 30 mm. wide. Section at rear end 65 mm. deep and 50 mm. wide. Overall width of front end 655 mm., and of portion rearwards from dashboard 760 mm. There is a cross member at the extreme front, dropped beneath the radiator, and of course a similar, but undropped, cross member at the rear. The only other transverse portions are the gearbox hangers, which are also 4 mm. thick and have a section 50 mm. each way, though, of course, the crankcase arms serve to stiffen the front part of the frame very considerably, and the stiffening is increased by bolting the starting handle bracket (which is in one piece with the upper half of the crankcase) to the middle of the front cross member.

The rear springs are rather short, as a transverse one is used. Their length is 1,600 mm. from shackle centres, while they are 48 mm. wide and have six leaves, the top one 8 mm. thick, and the remainder 6 mm. The front springs are 800 mm. long, and three millimetres less in width than the rear ones, there being four leaves of similar thickness to those of the latter.

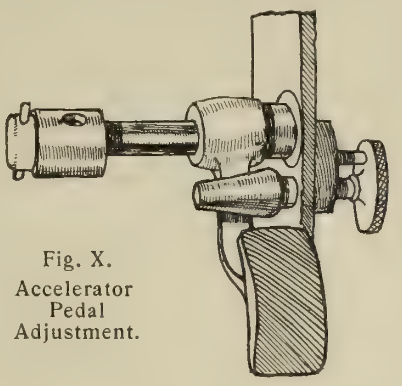


Fig. X. Accelerator Pedal Adjustment.

The spring shackle bolts are lubricated by an uncommon method, which is neat and cheap to make. Each bolt is arranged with the nut inside towards the frame, and the heads are threaded and fitted with knurled brass caps. Each bolt is, of course, pegged to prevent its turning, and a small hole takes grease from the cap to the bearing surface, while small spring catches prevent accidental revolution of the caps. The steering gear has no peculiarity, except that the front axle swivels are inclined with a view to causing the centre line of the pivot pin to intersect the ground at the point of contact of the tyre, this serving to reduce the stress on the connections somewhat. All the joints of both tie rod and connecting rod are of the ball pattern, and one of them was illustrated in the August issue.

Generally speaking, the details of the chassis are well thought out, and are mostly convenient, though the radiator drain plug might be brought outside the undershield with advantage, and studs with wing nuts might well be substituted for the set screws which at present serve to secure the gearbox lid. Still these are but small points, and no doubt the greatest improvement possible would be the provision of a fourth speed, and also of an ignition advance, the latter being of special advantage on so high speed an engine. Taken as a whole, however, the 14 h.p. Metallurgique may certainly be regarded as a worthy representative of its country's automobile products, particularly from the points of view of strength and of true neatness.

POBBLE AT BROOKLANDS.

A short account of the modifications which enabled an increase of speed to be obtained amounting to nearly fifty per cent. of the former maximum.

By Oscar S. Thompson.

THE experiences of Pobble at Brooklands during the racing seasons of 1908-09-10, and the methods by which the car's speed has been improved, from an average of 58 m.p.h. for a single lap in the first-named year to one of 88 m.p.h. for an unlimited number of laps in the last, present certain features of interest, the increased speed having been obtained not by any drastic reconstruction, but by a series of detail modifications without the smallest sacrifice of reliability or comfort. Indeed, it is not too much to claim that if I had known two years ago what I know now, the car could have been put into its present state of efficiency in two or three days, and at a comparatively small cost.

From the very beginning of my racing career I was to a great extent handicapped by the fact that I wanted the car primarily for touring, and was not prepared to try any experiment which would render it impossible to drive in a race one day and through traffic the next. The improvement was therefore slow, but on the other hand Pobble gave very little trouble, and only failed to finish once out of thirty-eight attempts. To-day the chassis is substantially the same as in 1908, having the same engine—four cylinders, 121 × 127—the same flywheel, the same transmission, the same springs, and the same frame. All that has been altered is the following :—

(1) White metal bearings to crankshaft and big ends.

(2) The flywheel has been somewhat lightened.

(3) The pistons have been made a little looser, and the connecting rods a little lighter.

(4) The oil pressure has been materially increased.

(5) The cams are more abrupt.

(6) A different carburettor has been fitted.

It will be seen from this list that the main features of the car are unaltered, and after various experiments with gear ratios we are reverting to the original 21-45 with 820 × 120 wheels.

Now all this sounds very simple, but the correct co-ordination of details has taken a lot of time and a great deal of patience, and it was not until July of the present year that we felt that we had got Pobble into a really satisfactory condition. Taking the years in order, 1908 was very uneventful, being devoted to experiments in gear ratios, carburettors, and tyres. It was not till the last meeting of that year that different cams were tried, and these at once gave another four miles an hour over the previous best. Altogether the car improved 11 m.p.h., or from 58 to 69 for the lap. The next year, 1909, was a bad one. Having fitted a larger carburettor, better induction pipe, and more abrupt cams, we got considerably more engine speed, but owing mainly, as we subsequently discovered, to lubrication trouble and to the pistons being too tight a fit, the car performed in a very in and out fashion. The official improvement was very small, only 3 m.p.h., or 72 instead of 69, but in practice we did laps at from 74 to 75.

Coming now to 1910, the engine was fitted with white metal bearings, the pistons were made an easier fit, the oil pressure was increased, the flywheel was lightened, and a standard White and Poppe 40 mm. carburettor was fitted. As soon as the track was reached it was evident that the car was a great deal faster than before, but as there was no time to make the necessary adjustments the average at the June meeting was only 78 m.p.h. In July a semi-wind-cutting body was tried, and an average speed of over 85 m.p.h. was attained in both flying laps of the 100 m.p.h. Handicap, which was run on the August Bank Holiday, while later in the week several laps were run at a still higher speed in practice. It was evident by now that the gear was too low, and had I raced again I should have increased the ratio from 19-45 to 21-45, viz., from 80 to 86 m.p.h. at 2,000 revolutions.

The outstanding feature of 1910 is the absolute reliability and consistency of the car. There is no trace of overheating, no splashing about of oil, and never once has the slightest mechanical adjustment been needed. This makes it all the more to be regretted that the pistons were eased, the flywheel lightened, the oil pressure increased, and the white metal bearings fitted, at one and the same time, since it is now impossible to allocate exactly the benefits derived from each of these alterations. In 1909 the car was apt to tire after two or three laps at full speed, and used to need quite an amount of attention, though for the above reasons we cannot

be certain how much of this should be attributed to defective lubrication, and how much to other causes. This year Pobble can be driven, in racing trim, at just under 10 m.p.h. on the top speed, can if desired be started on the same, and runs about 16 miles on a gallon. The compression is not particularly high, and the engine starts very easily on the single ignition—a Bosch high tension magneto. Excepting that the cams are rather noisy, the car is now in every way a better touring machine than when first delivered, and in the last sixteen months it has covered some 16,000 miles on the road.

These experiences go to prove that, in view of present-day knowledge, most cars, assuming that they have the necessary reserve of strength, could easily be made a good deal more efficient than they actually are; but whether a number of cars like Pobble, in the hands of reformed coachmen and others of his kind, would conduce to the greatest good of the greatest number, is a point on which I should prefer to refrain from the expression of an opinion.

SPECIALISED EDUCATION FOR AUTOMOBILE ENGINEERS.

THERE was a time when engineering education consisted almost entirely of rule of thumb experience, but that is now so far distant in the past that the necessity for special theoretical study for each branch of the engineer's vast subject is a matter which requires no vindication.

In the days when the value of theoretical training first came to be recognised as something more than a fad, it was regarded as a, perhaps, convenient accessory to workshop experience, but it was scarcely considered possible that theoretical instruction could replace the smallest fraction of the time-honoured apprenticeship or pupilage. After a while, however, it became obvious that a youth who had received two or three years' college training, followed by two years in the shops, was in a better position, and was actually more valuable than another youth who had spent the whole four years or five years in manual work only. The reason for this fact is not far to seek, though many appear to have had great difficulty in recognising it. It is simply that the apprentice learns, firstly, how to do things, and secondly, why they are done, while the student learns the reason first, and attacks his practical work with a good knowledge of the ultimate purpose of his manual tasks.

Following this argument, it would appear that the greater the knowledge of tools and their uses possessed by a student, at the termination of his college course, the better equipped would he be to attack the practice of his profession, and in some colleges effort is made to give workshop instruction in the use of simple hand and machine tools.

There are three methods of conducting manual engineering schools. One is to start with boys of ages between sixteen and eighteen, to train them thoroughly in manual work, and to supplement the workshop instruction by study of the simpler principles of mechanics, with their attendant mathematics and science. This system may last for two or three years, and is suited principally for boys with a moderate general education. Such a course may be followed by a year or two of deeper theoretical work, or the student may take a year's apprenticeship, which will lead him to the possibility of some minor position on a works staff. The second method is intended for boys with a better general education, and usually commences when they reach an age of eighteen to twenty. It consists of two, three, or even four years' study, with a few hours per week spent in manual work, similar in nature to, but less in extent than that comprising the chief portion of the first system. A student taking such a course is qualifying for a much higher position than the student working on the first system, but he will certainly need two years' shop practice before he is competent to take charge of any considerable undertaking. The third system is to give instruction of the same nature and degree as that of the second, but for the student to spend three months of each year in works. For many reasons this is the best of all systems, and many a man who has spent three years in such procedure has been found to be an efficient member of his profession at their conclusion.

The ideal system is yet to be instituted, and there are many difficulties in the way, on account of the concordant working it would require between schools and colleges. There is no doubt that the first system of manual instruction with theoretical supplement produces a good workman in the minimum of time. There is also no doubt that the third system produces the best engineer in minimum time. Obviously, the best procedure would be to make a combination of the two systems.

To digress for a moment, it is useful to examine the ambitions of the budding engineer. If he intends to specialise on automobile work he may wish to design, or he may wish to control actual manufacture, and though it may be thought heretical to say so, the designer has less need of works instruction than the would-be works manager, the difference being that there are many things quite outside engineering which the latter has to learn, and which do not affect the former.

To return to our argument, it is precisely in the extraneous matters that all the present schemes of education appear to fail. The college-trained man is apt to have a total lack of appreciation of the importance of costs, as he has been taught the best way to do a thing scientifically rather than economically, and also the one part of a works with which an apprentice has nothing to do is the organisation—the very section which he probably has hopes of controlling in the future.

Works systems are as essentially a part of the education of an engineer as are the handling of tools or the theory of structures; much concerning them could be learnt in the lecture room, and much more on the spot. Some of the most enlightened men amongst designers are the least successful practically and commercially, just because of that lack of the appreciation of the importance of cost, which comes with some acquaintance with the trade side of a business; and though it would be a sorry day for the profession if cost took first place in a curriculum, yet it ought not to be disregarded wholly. It is omissions of this nature that have led to the loss of more money than almost anything else in the world of engineering. How many motor manufacturing concerns have failed to make ends meet simply and solely because of continual alterations of pattern, which, if quite in accordance with theory, were far too expensive for serious practical consideration?

In the strenuous life of modern times it becomes more and more difficult for a young man to make himself efficient in accordance with the necessities of the times, and anything which will shorten the period of training, or that will make the training more complete, ought to be done. Manual training schools of the best class have proved that it is possible to take a lad of fourteen and turn him out a really good fitter, for small work, at the end of a couple of years. If he then intends to take up engineering professionally, he has to take a short apprenticeship, and climb by slow degrees to the post of foreman or assistant to a works manager, or he has to spend additional years studying theory at an engineering college. It is, of course, a maxim of educationalists that specialised training should not commence too young, and it is perhaps for this reason that the school workshop so often becomes an addition to the playground rather than an extra laboratory. To be of the utmost value it ought to be organised properly, and boys ought not to be allowed to attempt to use tools without proper instruction in the handling of them. The practice of permitting boys to make models and small engines for themselves is an excellent one, and a great help, because it provides an interest which set tasks can never do. This makes the work of the instructor all the harder, but it is obvious that the difficulty is not impossible to overcome and there is no need to consider the matter in detail. To a boy with an engineering bent, the properly directed workshop would be a place of recreation as well as of instruction, and if he continued to study the theory of the profession afterwards he would find that many months of rather irksome practical work had been saved for him, thus enabling him to make a somewhat earlier start on his life's work.

Of course, the value of the school workshop is in most instances limited by the funds available, and it would be neither possible nor desirable that the plant of machine tools should be very

extensive. All that is necessary is that there should be examples of the simpler mechanisms—the lathe, the milling machine, the planing machine—upon which the more elaborate tools are based.

However, to set aside the school and college training and to consider the apprenticeship form of training by itself. It is a matter of common knowledge that only a comparatively small percentage of premium pupils are sufficiently advanced in years to see the necessity for close application. If a boy is taken fresh from school and relieved of the discipline which decrees when he shall get up, when he shall go to bed, how he shall work, and so on, he is ready to take advantage of that freedom to its fullest possible extent, and it is often not until the second or third year of his pupilage that he really settles down to work, if he does so at all. The manufacturer into whose works he goes too often regards the premium paid as a mere fee which allows the boy the run of his works, allows him to learn what he likes very much how he likes, and if he does not get on—well, it is his own fault. This does not seem to be the right view to take. If a man or a company is prepared to take students in exchange for a fee it ought to be prepared to instruct those students. If he or it undertake to teach their business they ought to do so, and ought not to leave the majority of the pupils to their own devices while just the few brilliant ones learn by keen observation, and a still smaller number by a judicious mixture of tact and tips.

How many youths at the end of three years' hard work in shops are worth two pounds a week to the man in whose factory they were supposed to be learning? Granting that there are some exceptional men who really take trouble with pupils, and even discharge them if they will not work or if they show no signs of becoming efficient, still for every man who can truly call himself an engineer there are far too great a number who have failed, not through lack of ability so much as lack of the guiding hand at a time of life when it was still necessary. Even in the best instances of works training the most successful pupil is generally only a good workman; at the end of his " time " he has probably never had anything to do with the organisation or the costing system of the works; that is to say, if he be offered a small position on the works manager's staff he still has a great deal to learn, and, in fact, he has often to erase ideas which he has acquired wrongly through lack of knowledge of the money side of the business.

It would be absurd to suggest that a pupil ought to be given an insight into the purely commercial office work of a firm, but he ought to understand buying, he ought to know why it is not always politic to choose the ideal material on a quality basis, he ought to be made to realise the vital importance of checking and reducing loss of material through wasteful machining, breakages of tools, time spent in setting up work, and so forth. In fact, it is just these things, and the proper appreciation of them, that make the difference between the organizer and controller and the organized and controlled.

That these things should be learnt after the manual training is complete is quite in accordance with reason, but there is an undoubted tendency to rate manual work too highly. The mechanic of the present day is much more a machine minder than an actual craftsman, and knowledge of the capabilities of machines is much more valuable to the budding engineer than is ability as a fitter. One of the outstanding advantages of automatic tools, and of jigs of all kinds, is that they permit the use of a slightly cheaper class of labour. In other words, it does not require so much skill to control an automatic tool as to make a piece by the old-fashioned methods. This in turn means that a pupil will be able to get through the machine shop part of his training more quickly as automatic machines increase in number.

It is known that certain firms who instituted a properly arranged pupil's course, and employed instructors to direct the whole system, found that they obtained some valuable men as additions to their staff in a remarkably short space of time, and this is proof positive that organized instruction pays both the employer and the pupil.

To sum up all the foregoing, it appears that familiarity with simple tools should be gained during school days; that a college course should follow the school; that the long vacations of the latter should be filled by pupilage in works, and that the training should be completed by instruction in organization and control in an actual factory.

Or, on the other hand, if the college training is dispensed with, the works training ought to be arranged in an academical progression, and if a man aspired to the upper ranks of his chosen profession he would be obliged to do some theoretical work in addition. The need for the maintenance of a certain amount of discipline after school days is a strong argument in favour of the college course preceding the works training, and also it is a fact that the knowledge of theory gained at college enables the works training to be shortened. Yet another reason for putting the college first is that theory changes and develops slowly, but practice changes fast, and it is therefore likely that a pupil who finishes his training in works will be more efficient at the end of it than another whose works experience is three years old by the time he leaves his college.

So much for generalities, and now to turn to the special requirements of the student of automobile engineering. His training does not need to differ much from that of the majority of mechanical engineers, but it should be different in a few respects because of the differences which exist between large work and small work. If a man intends to specialise on the building of small engines, and the machinery adjacent to them, he needs to know many things that are not wanted by a man who proposes to devote his energies to the construction of locomotives or marine engines. The automobile engineer uses pounds where his colleagues use tons, uses millimetres instead of feet, uses materials of a strength unheard of in most other branches of the profession, becomes accustomed to quite different varieties of steel, and though such differences are small they are real enough, as any man who has taken up automobile work after some years of heavier engineering can testify. Also the formulæ used by the automobile designer are almost all empirical, they are based on experiment alone owing to the utter impossibility of calculating the forces to which a chassis may be subjected, and there is thus a considerable mass of knowledge which needs to be acquired before leaving any other branch of the profession.

At the same time there is but little in the training of an ordinary mechanical engineer that the automobile engineer can afford to be without, and so it is noticeable that in engineering colleges any course which is provided for students, who intend to specialise on automobile engineering, is usually a post-graduate course; that is to say, it is an optional course that can be taken by any student who has passed the final examination at the end of his general course of two or three years, as the case may be, and it is not a course that can be taken concurrently with the general one. At present there is no doubt that few of the colleges know quite what is most useful to provide, and very many automobile lecture courses consist mainly of descriptions of different designs, with perhaps a somewhat vague consideration of the proportions of essential parts.

The City and Guilds of London Institute are at present the only body in this country holding a public examination in motor construction, and this is not very advanced. There are two grades, "ordinary" and "honours," but the course of study is in each case directed chiefly to acquainting students with current practice, and students are not expected to have any advanced knowledge of either applied mathematics or science. Still, the possession of an honours grade certificate is a guarantee of a very fair knowledge concerning automobile design, and sufficient knowledge of mathematics for most drawing office or workshop calculations. In other words, a boy with a good general education who was able to spend only a small sum on his training would be well advised to take the complete course for these two examinations, at some such place as one of the Polytechnics, working during the day as a pupil in a convenient engineering works (not necessarily a motor works), and perhaps taking a few day classes to eke out his general knowledge. Two years spent thus ought to enable a small appointment to be obtained, though it is worth while to remark that a boy who is doing evening study ought not to do manual work as well for more than seven or eight hours a day at the most. Boys require much more rest than they are usually supposed to do, and it is a mistake to imagine that a lad fresh from school can labour from half-past five in the morning to six o'clock at night, attend classes from seven till nine, and then find time for the necessary home reading, let alone for sufficient bodily and mental rest.

For the more advanced class of students, such as those who could take a post-graduate course at one of the University Colleges, the extra theoretical study may well be combined with harder physical work, as the groundwork sub-

jects are already mastered and the mental effort required is not so great. Occasionally four full years are spent at college, but if the final examination can be passed at the end of the third year there is no real need for the extra year to be given to theory alone.

It is during a fourth or fifth year of training that the commercial side of engineering ought to be studied, after practical work has begun and the surroundings of a works are familiar, and it is amongst post-graduate courses that "costing" and "organization" ought to find a place. Of course, such subjects are difficult to teach because they are so largely a matter of opinion, still there are some basic principles that are of such importance that they ought to be taught, and it is on account of their importance that mention of them is repeated.

The writer has often received requests for advice from the parents of sons who wished to become motor engineers, and it may not be out of place to make a general answer here.

The first thing is to make quite sure that the ambition is not a mere momentary passion and that the boy has a strong natural taste for mechanics. The second step depends upon the means of the parents, and may be either to apprentice the boy where he can spend some of his time in preparation for the City and Guilds examinations, or to send him to college to study mechanical engineering, and, if possible, secure the engineering B.Sc. of the London University. In the latter case the long vacations of the summers ought to be spent in works: any works will do, and a general engineer's shop is as good as, if not better than any, for, if system is lacking, the work (and so the experience gained) is varied. After a successful three years spent in this way an additional year in a good modern factory should complete the training and enable an appointment with good prospects to be obtained.

Finally, it should never be forgotten that in automobile engineering there is no room for men with only moderate or poor intellects. Though this is true of all professions, it is especially true of a new one, where so much of the work is creative, and where knowledge is advancing so rapidly.

A COMPARISON OF VALVE SYSTEMS.

Being a paper read before the Society of Automobile Engineers (U.S.A.) at the recent meeting at Detroit, by Eugene P. Batzell.

INTRODUCTORY NOTE.

At the last general gathering of the American Society of Automobile Engineers, several interesting papers were read, but the most important by far was the one which is reproduced hereunder. The whole matter of the paper is so instructive that to omit more than quite small portions here and there would be inadvisable. It has perforce had to be curtailed somewhat, as the original is extremely long, but every endeavour has been made to retain all the conclusions, and sufficient of the reasoning to enable those conclusions to be appreciated at their true value.

To take up the subject properly we will compare the different new systems with the old poppet valves, and get our results from this comparison. There is more reason for this, because many results are not obtainable in definite figures, therefore it is more profitable to choose for the standard unit of comparison a system with which one is most familiar. The discussion will be taken up here mostly as an investigation of valve-opening diagrams, together with their respective intake gas velocities. Let everything be referred to four-cycle single-cylinder motors of five-inch bore and six-inch stroke running at 1,000 r.p.m., corresponding to 1,000 feet per minute piston speed. The cam-operated poppet valves only will be discussed, not touching the possibility of their being operated indirectly by some means.

The rate and duration of opening of a poppet valve depends on the shape and size of the cam, and the type and size of the cam-follower. The most extensively used forms of cam-followers are the roller, the "V" shaped, and the flat or mushroom valve lifter foot (Fig. I.). The valve-opening diagrams with the roller and the "V"-shaped valve lifters depend upon the radii of the contact surfaces. Identical diagrams are obtained regardless of whether the cam-base circle or the follower-roller or spindle are made of a certain size, providing the relation between them remains the same. Therefore, only the flat and roller lifters will be considered.

As to the cams themselves, they can be made in very different shapes. In Fig. II. are represented three, which will be discussed here, 1 and 2 being designed for use with a roller-lifter, and 3 for the flat.

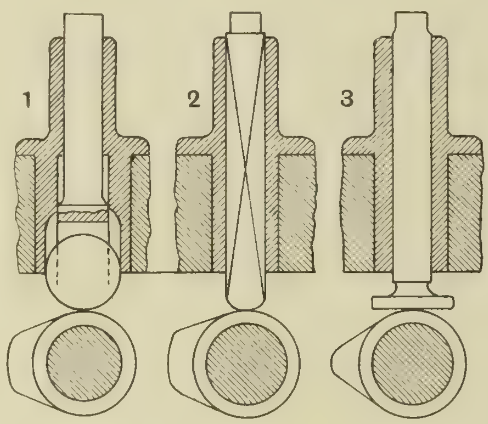


Fig. I.

They are all inlet cams with same lift (five-sixteenths inch), and are shaped for the same duration of valve opening, namely, 210 degrees of crankshaft movement. The inlet could, for instance, start to open 5 degrees after the upper dead centre, and close 35 degrees after the lower dead centre, which is fairly good timing for high and medium speed. With roller-lifter cams there is a one-sixty-fourth inch clearance between the valve stem and the lifter. The cam for the flat lifter is easier to shape with a greater clearance, which is made one-thirty-second inch on the drawing.

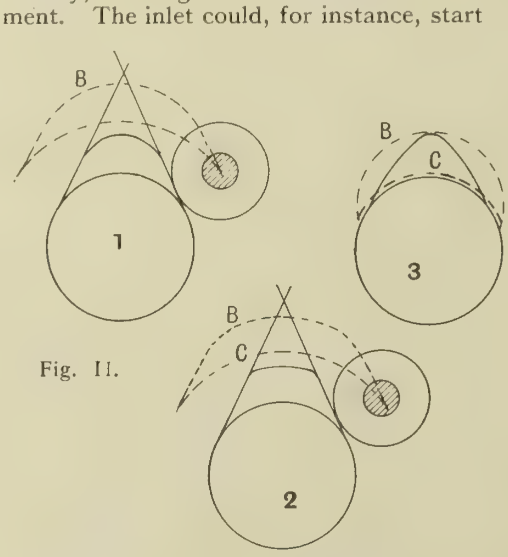


Fig. II.

The cam base circles are all $1\frac{1}{4}$ inches in diameter, and the roller is $\frac{7}{8}$ inch in diameter. The flat contact surface of the valve lifter 3 (Fig. I.) is one inch in diameter, and the cam is rounded in such a manner that contact with the edge of the lifter is avoided.

The difference in the two roller cams may be seen in the drawing. The top portion of cam 1 is a single circle tangential to its straight flanks, which latter are determined by the position of the roller and the valve lifter clearance in the usual way; that is, the lines touching the roller and the cam-base circle at the moments when the inlet begins to open and close. The flanks of cam 2 are determined in the same manner, but the extremity of the cam is formed by an arc concentric with the base circle, and extended until it intersects the flanks. These intersections are rounded into a small radius. On the same drawings the dotted lines B represent the amount of valve lift. For each angular position of the cam the lift is equal to the distance between lines B and C taken in the direction of the centre line of the valve-lifter. For cams 1 and 2 line C is a circle, passing through the centre of the roller at the moment of valve opening, and for cam 3 it is the clearance circle.

The valve lifts obtained in such a manner for different angles between the cam centre and the lifter can be plotted against the angles of crank position, thus giving valve-opening diagrams. Such diagrams for the cams of Fig. II. are reproduced in Fig. III., and are indicated according to the cam numbers by 1, 2 and 3. They all give the following timing: Inlet opens at 5 degrees after the upper dead centre and closes 35 degrees after the lower dead centre. However, the area of valve opening at any particular crank position is a more important thing to know than the valve lift. Referring to Fig IV., and assuming that the gases flow through the valve opening parallel to the valve seat, the valve port area is:

$$S=\pi \times d_1 \times h_1 = \pi\ (d+h_1 \sin \alpha)\ h_1$$
or substituting $h_1 = h \cos \alpha$
$$S=\pi\ (d+h \sin \alpha \cos \alpha)\ h \cos \alpha \qquad (1)$$

When the valve seat is flat the area S is proportional to the valve lift h, as then the second part in the parenthesis disappears. Flat valve seats are seldom used, however, because they are hard to keep tight; they retard rather than promote the gas flow, making the gas change its direction suddenly, and besides they also become bevelled in time. For bevel-seated valves the port area S is a function of the second degree, according to equation (1). However, this area may be considered directly proportional to the valve lift in this case also, because the squared factor in equation (1) is relatively small, and may be neglected. The error thus made is the slighter the smaller the horizontal angle of the valve seat. Assuming, then, that the valve lift and port area are directly proportional, we may also consider curves 1, 2 and 3 of Fig. III. as representing the variation of this port area with the crank position, and it is only necessary to construct a diagram of the proper scale to get the port opening areas in actual figures. We will suppose that the motors in question have valves of 2.7/16 inches clear diameter (d, Fig. IV.), and that the valve-seat has a 30-degree horizontal angle α. As the lift was made 5/16 inch, the maximum valve port area is—

$$S_{max}=\pi\ (2.4375+.1355)\times.271=2.2 \text{ appr. square inches.}$$

If we make the valve lift of 5/16 inch in Fig. III. correspond to 2.2 square inches, the scale for the port area curves becomes known, and it is then possible to draw from these diagrams a number of conclusions as to the motor characteristics with the different poppet-valve lifting mechanism.

As may be seen from Fig. III. the area enclosed by the curve 3 is slightly larger than that enclosed by curve 2, and these areas are both larger than the area enclosed by curve I. The area of these curves is an important factor, determining the quantity of fresh charge which is drawn in by the motor during its suction stroke. It is an undisputed fact that the greater the area of these curves the greater also will be the quantity of fresh charge drawn in.

If H = f (t), is the momentary valve port area in function of the crank position given by time (t), one can write

$$\text{Motor Power} = P\int_{t_2}^{t_1} H\, dt$$

where the letter P is used as an indication of proportion and the limits t_1 and t_2 are the moments of valve opening beginning and closing. However, in some cases, this equation requires a corrective factor, taking account of the elapsing of the intake gas velocities through the valve ports. This theoretical gas velocity is equal to the momentary piston displacement divided by the port area at the period considered.

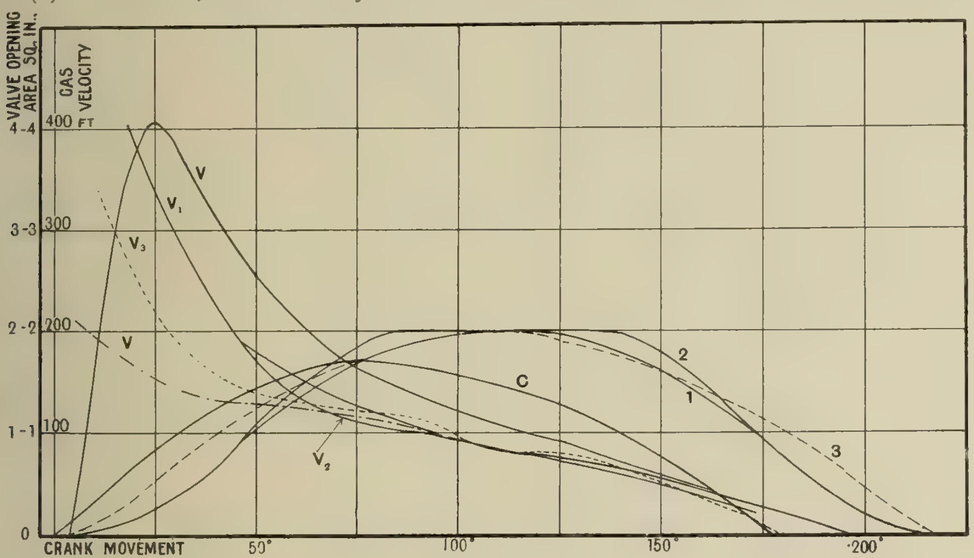


Fig. III.

$$\text{Gas velocity } v = \frac{\text{Piston area} \times \text{shifting}}{s \times t} \text{ or}$$

$$v=\frac{dV}{f\ (t)\ dt} \text{ where } s = f\ (t) \text{ as previously}$$

The piston speed is given by the equations:

$$\text{Piston speed } c = \frac{v \sin (A \pm B)}{\cos B} \qquad (2)$$

$$c = v \sin A\ (1 \pm \lambda \cos\alpha) \qquad (3)$$

in which the letters correspond to those in Fig. V.

Equation (3) is not exact. Assuming that the length of the motor connecting rod e = 13 inches, so that

$$\lambda = \frac{r}{e} = \frac{3}{13} = 0.231$$

the maximum error in equation (3) at the time when angle β is a maximum would only amount to about 2.5%, and may therefore be neglected, especially since equation (3) is the most convenient to use.

The piston speeds for different angles "A" of the crank and for 1,000 r.p.m. of the motor, as determined from the last equation, are given in Fig. III. by curve "C." The cylinder bore being 5 inches, the piston displacement at one foot per second piston speed is 236 cubic inches. Curves 1, 2 and 3 of Fig. III. giving the momentary port area, and curve "c" the momentary piston speeds, the theoretical intake of gas velocity curves can be determined now for each of the cams in question. These gas velocities are designated in Fig. III. as v_1, v_2 and v_3, corresponding to the numbers of the cams.

Comparing these three curves, we notice a maximum difference of only about 10 per cent. between v_1 and v_2. Moreover, this difference occurs when the inlet openings are near their maximum, and when the gas velocity is much lower than at the beginning of the opening. Therefore, in respect to motor power, cam 2 will show no great advantage over cam 1. With slow-running motors, the difference between these cams may be absolutely imperceptible.

With these cams under the assumed conditions, the inlet gases are strongly choked during the earlier part of the valve opening, and consequently there is considerable vacuum inside the cylinder during this period. This vacuum, besides causing direct loss of motor power, also sucks oil past the piston, which tends to cause carbonization. A point in favour of high intake velocity is a more complete vaporizing and mixing of the gas particles with the air when entering the cylinder. But normal gas velocities should be sufficient for that, so that any increase would be a pure loss.

Referring to Fig. III. it will be seen that cam 3, with the flat valve lifter, shows far better results than the other cams, in respect to motor-power. With this cam the valve opening is much quicker at the beginning than with the other cams. The theoretical gas velocity drops much more quickly with cam 3 than with cams 1 and 2, and consequently the intake gases are much less choked with this cam during the early part of the inlet. The difference in velocities with cam 3 and the roller cams, respectively, amounts to 20-30 per cent. during a crank motion of 45 degrees, and, furthermore, the choking effect is proportional to the second power of the gas velocity at least. The quantity of fresh charge admitted to the cylinder during the first half turn of the crank will be considerably greater with valve-lifting mechanism 3, and this will reduce the vacuum inside the cylinder, together with its bad results. This mechanism is decidedly advantageous, except that it requires very strong valve springs, because the first part of the curve of cam lift is very steep. The stronger the valve springs are, the more the valve seat and valve mechanism are subjected to wear, and constructions 2 and 3 are particularly disadvantageous, because of the great sliding motion of the cam against the lifter. Of course, cams for roller lifters can be shaped to give the same quick opening as cam 3, but the objections to quick opening are then greater. . . .

In curves v_1, v_2 and v_3 of Fig III., it will be noticed that the gas velocity also gradually drops during the period when the valve openings decrease, because of a more rapid decrease in the piston speed. From a velocity of 170 feet per second at the moment of greatest valve opening, the gas velocity drops to 50 feet per second during the following 60 degrees of crank travel. At the lower dead centre the inlet valves still have openings 0.7 square

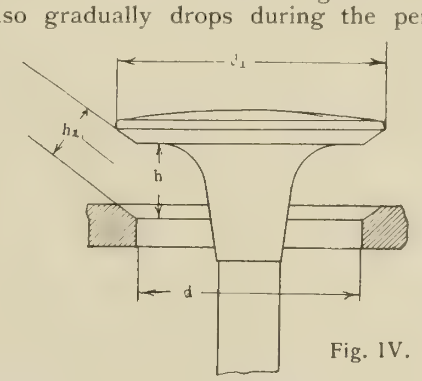


Fig. IV.

inch (cams 1 and 2) and 1 square inch (cam 3), respectively. The last 10 degrees of crank movement before the dead centre need not materially influence the change of gas velocity, and during the following crank movement it can, therefore, not possibly be greater than 50 feet per second, as it was at 170 degrees crank position. By assuming that the gas velocity remains 50 feet per second during the whole period, the inlet valve remains open after the crank has passed the lower dead centre, and by further assuming a mean

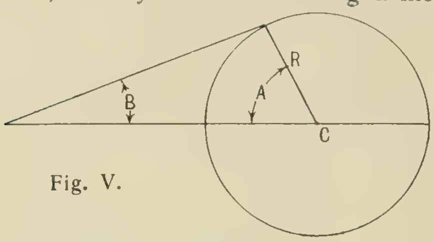


Fig. V.

figure of valve opening area for this time, we can determine the maximum quantity of fresh charge which can be forced into the cylinder by gas inertia if no losses take place.

Mean valve port area for cams 1 and 2=0.45 square inch.

Mean valve port area for cam 3=0.62 square inch.

Duration of 45 degrees crank movement at 1,000 revolutions per minute=0.0075 seconds.

(1) Quantity of gases $50 \times 12 \times 0.45 \times 0.0075 = 2$ cubic inches.

(2) Quantity of gases $50 \times 12 \times 0.62 \times 0.0075 = 2.8$ cubic inches.

The piston displacement being 117.8 cubic inches, these quantities constitute 1.7 per cent. and 2.38 per cent. of the piston displacement volume. These figures are the maximum possible, and they show that a late closing of the inlet valve cannot have any direct influence on the motor power. Only if the motor were choked strongly at high speed with partly closed throttle or by smaller valves so that there would be considerable vacuum in the cylinder, would the supplementary cylinder filling through late inlet closing be of direct value. The main reason for a late inlet closing is that the cam becomes of fuller shape, affecting the valve port area in such a way that the intake gas velocity is kept lower during the piston suction stroke and the quantity of fresh charge then drawn in is increased. This influence of late inlet closing on the motor power is thus indirect.

To obtain an air velocity of 170 feet per second, without considering losses, a difference in pressure of approximately 0.25 pound per square inch is required. The difference in pressure required for an air flow (if this difference is not very great) may be found from the equation :

$$P = \frac{V^2 \times Q}{2g \times 144} \text{ pounds per square inch.}$$

where V=air velocity in feet per second, Q=weight of 1 cubic foot of air (about 0.081 pound), and g=gravity acceleration =32.16 feet. A flow of 50 feet per second requires only about one-twelfth the pressure difference from a flow of 170 feet per second. From this it is seen that without extra resistance the vacuum inside the cylinder necessary for the gas flow is very small, but owing to the existing resistances it may amount to some pounds per square inch in motors only slightly throttled. Therefore, the curve of actual gas velocities through the valve ports differs greatly from the theoretical one, especially at the beginning of the valve opening. This actual velocity will follow approximately curve v, Fig. III., the shape of which is merely a rational guess. At the 170 degrees crank position this curve v shows no marked difference from curves v_1, v_2 and v_3, and there may be seen also the little direct influence of late inlet closing.

From the gas velocity curves it may also be deduced that it is more important, from the standpoint of motor power, to have a quick opening inlet of small size than a slower opening inlet of large size. And it is also better to obtain a certain valve port area through large valve diameter and small lift, than small diameter and high lift. The cams for high and small lift being made of the same type, say, for instance, both after Fig. II., No. 1, will have coinciding valve opening curves on the length of their straight flanks. The valve with smaller lift, but with a larger diameter, will also have a larger port area during this time, and this will help to reduce the high intake gas velocity at the inlet beginning. The influence of a change in the time of valve opening is noticeable by comparing curves v_3 and v_4 of Fig. III. Curve v_4 corresponds to cam 3 with the valve opening starting at "0" degree crank position. It represents a much lower and also much more constant gas velocity at the beginning than curve v_3; and, besides, these curves practically coincide further along. It follows that, theoretically, an early inlet opening is advantageous, and this statement is true for high speed. In practice, however, better results are sometimes obtained with some-

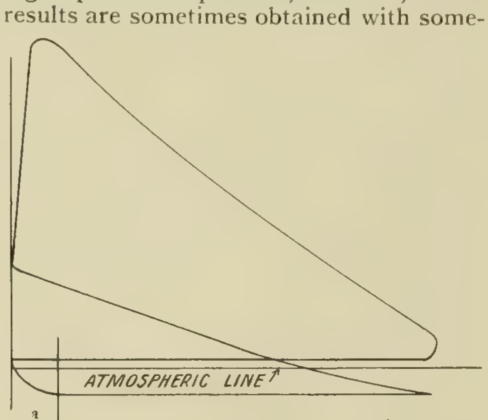


Fig. VI.

what later inlet openings, but if a certain long duration of inlet opening is desired, it is better to start and close early, because a too late inlet closing will result in motor power loss. The inlet valves should close before the moment when the piston in its backward stroke compresses the charge inside the cylinder to the pressure in the intake pipe. With a fairly opened throttle, and at normal speed, the cylinder at the end of the suction stroke will hardly contain more than 80-85 per cent. of a full charge, and the suction line in the pressure diagram, Fig. VI., will be below the atmospheric line, but at the same time the pressure in an intake pipe of sufficiently large diameter will not be much less than atmospheric. Also the amount of supplementary cylinder filling is very small under the conditions cited, and, therefore, there is no likelihood of a return flow until after about 10 per cent. of the return stroke, or nearly 45 degrees of crank movement. A lag of 30-40 degrees in the closing of the inlet valve is, therefore, not harmful in this respect. However, when the throttle is nearly closed, the pressure inside of the intake pipe will not differ much from that inside the cylinder, and then the piston in making 10 per cent. of its return stroke while the inlet valve is open will cause a part of the

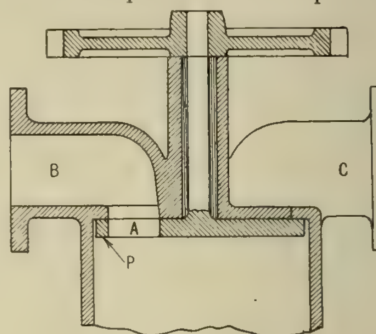


Fig. VII.

cylinder charge to be expelled into the inlet pipe. This results in loss of power and efficiency.

If the compression space is 25 per cent. of the total cylinder volume, and the compression starts at 10 per cent. of the return stroke, the effect will be the same as if the compression space were 27 per cent. ; but with a compression beginning

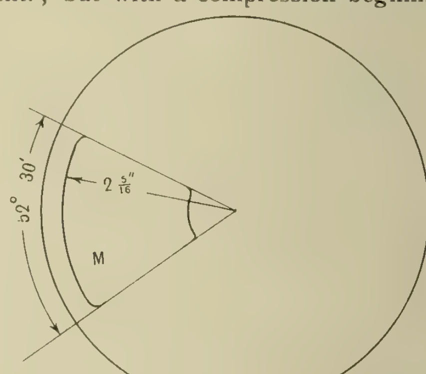


Fig. VIII.

at 2-3 per cent. of the return stroke, corresponding to a lag in inlet closing of 20-25 degrees, the rated compression of 25 per cent. may be figured on. Other conditions being equal, these compression ratios will result in the following compression and explosion pressures, and theoretical efficiencies :

	Explosion Pressure.	Thermal Efficiency Per Cent.
Compression pressure $P_{25} = \left(\frac{V_1}{V}\right)^{1.35} \times 13 = 84.5$ lbs. per sq. in. absolute	300	39
Compression pressure P_{27} = ...76 lbs. per sq. in. absolute......................	270	37.5
Difference ....=8.5	30	1.5

The exhaust gas velocity can be depicted approximately by noting the character of the inlet valve opening and supposing it to be shifted over to the right 20 or 30 degrees. An exhaust cam for 40 degrees lead and 5 degrees lag gives the maximum opening at $72\frac{1}{2}$ degrees after the lower dead centre, or about 25 degrees before the maximum piston speed is attained. The gas velocity curve will, therefore, slowly rise from 0 degree to $72\frac{1}{2}$ degrees ; then rise at a quicker rate to a point of maximum piston speed and gradually drop from this point to zero at the

upper dead centre. The smaller the gas velocity during this period, the less resistance there will be on the piston. An exhaust valve giving a large, quick opening will be favourable not only in this respect, but will allow also of a later exhaust beginning. The motor power can be increased by using large exhaust valves as well as large inlet valves.

It is advisable in many cases to close the exhaust valve after the piston reaches its upper dead centre, and particularly so if the exhaust valve closes gradually. At high speed there is considerable back pressure at the end of the piston stroke, and if the exhaust were closed exactly at 0 degree, the pressure line in diagram, Fig. VI., from this point on would follow an expanding curve during the piston travel "a," and the admission of fresh charge would begin only at the end of the travel "a," consequently the percentage of spent gases remaining in the cylinder would be greatly increased. If the inlet valve opens before the piston has passed through the distance "a," a part of the spent gases may get into the inlet passages and cause back firing, if the charge be poor. Moreover, these spent gases penetrating into the inlet passages cause a reverse flow and reduce the inertia of an uninterrupted flow at high speed. If the exhaust valve remains open a trifle beyond the upper dead centre, there may be time for an equalization in pressure between the cylinder and the exhaust pipe, before the piston moves down perceptibly. In this way the distance "a" of Fig. VI. will be greatly decreased, and the quality of the cylinder charge improved. . . . . .

From what has been said it can be seen that quick closing of the exhaust is also preferable, with respect to back pressure, quantity of spent gases remaining in the cylinder and time of closing. The quicker the valve opening and closing, the greater the power and flexibility of the motor. In this respect the poppet valve is inferior to rotary valves, sliding sleeves, piston valves, etc. In Fig. VII. is shown a sketch of a cut through the cylinder head of a rotary valve motor with a flat disc valve A placed inside the explosion chamber and bearing against its top part. The inlet and exhaust passages are denoted by B and C respectively. The gear is mounted on the valve stem and revolves the valve A. The valve A has a single port, registering with the inlet and exhaust ports formed in the cylinder heads. This valve is of a sector shape (see "m" in Fig. VIII., which figure shows the top view of the valve disc). The valve is designed to rotate at half crankshaft speed in a four-cycle motor; and the same timing will be assumed as for the poppet valves discussed. The duration of the inlet extending over 210 degrees of crank motion, the valve must remain open during 105 degrees of its own revolution, and making the valve port of the same size as the inlet in the cylinder head (whereby the greatest maximum opening is obtained), the radial valve port edges must enclose an angle of 52 degrees, 30 minutes. With this type of valve the prevention of leakage, particularly at the part "p" adjoining the port "m," is not easy. The part "p" must be made of the proper width, say, about 3/16 inch. It must not be much larger than this because that would reduce the area of the port "m." The valve stem can be $\frac{3}{4}$ inch in diameter, its bushing $\frac{1}{8}$ inch thick, and there should also be some cylinder metal at the lower end where the gas ports are located. The dimensions of port "m" are thus approximately fixed as given in Fig. VIII. The total area of the port is

$$S = 2\pi \frac{52.5}{360} \times 1.5325 \times 1.3125 = 2.2 \text{ sq. ins.}$$

or the same as that of poppet valves of 2.7/16 inches clear diameter, 5/16 inch lift and 30 degrees angle of seat.

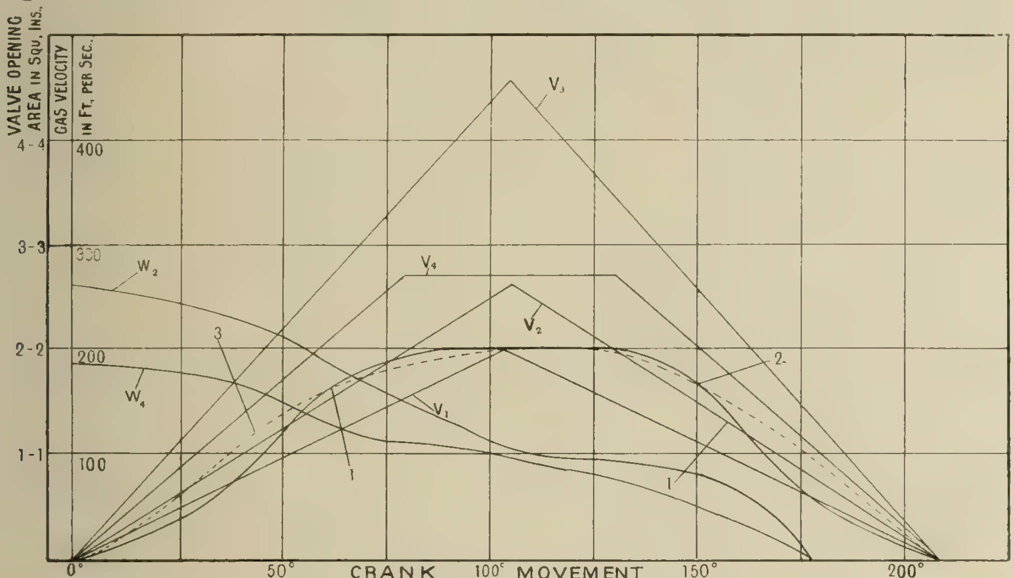


Fig. IX.

It was mentioned above that the maximum valve opening is not important by itself, as the time factor also enters into the problem. In rotary valves driven at constant speed, and with a port shaped to give the maximum possible opening area, the area of opening at any moment is proportional to the angle through which the valve has turned. The area of opening increases uniformly from the beginning of the opening until the maximum is obtained, and then decreases in the same manner. Therefore, the "curve" representing the change of this area with reference to the crank movement will be an equilateral triangle. This curve for the valve shown in Fig. VII. is v_1 in Fig. X. For the sake of comparison, curves 1, 2 and 3 of Fig. III. are reproduced in Fig. IX., but all openings start at 0 degree of crank position. (In this article the cylinders are supposed to be in the same plane as the crankshaft—not offset.) It will be seen that curve v_1 has a much smaller enclosed area than curves 1, 2 and 3. Moreover, the valve opens and closes but a trifle quicker than with cams 1 and 2 Fig. II., and less quickly than with cam 3, Figs. I. and II. The theoretical curve of gas velocity will not be as good with this rotary valve as with poppet valves, but the actual conditions of the gas flow may be better. In poppet valves the gas flows into the cylinder through a comparatively narrow annular opening, which offers considerable resistance. Besides, a part of this annular opening is close to the walls of the explosion chamber, unless the valve is located in the centre of the cylinder head, and that tends to change the flow through the valve. The paths of unequal length which the gas particles passing through different sections of the valve opening must travel, further diminish the value of the poppet valve port effective area. The rotary valve, on the contrary, will allow a practically constant flow.

The flat disc valve can be improved in respect to cylinder filling, if it can be made larger than in Fig. VIII. This can be done as shown in Fig. X. The engine there represented differs from Fig. VII. in having a removable cylinder head and a valve of larger outside diameter than the cylinder bore. It will be noticed that the leakage path around the outer valve edge is rendered more difficult in this construction, if the valve seats are

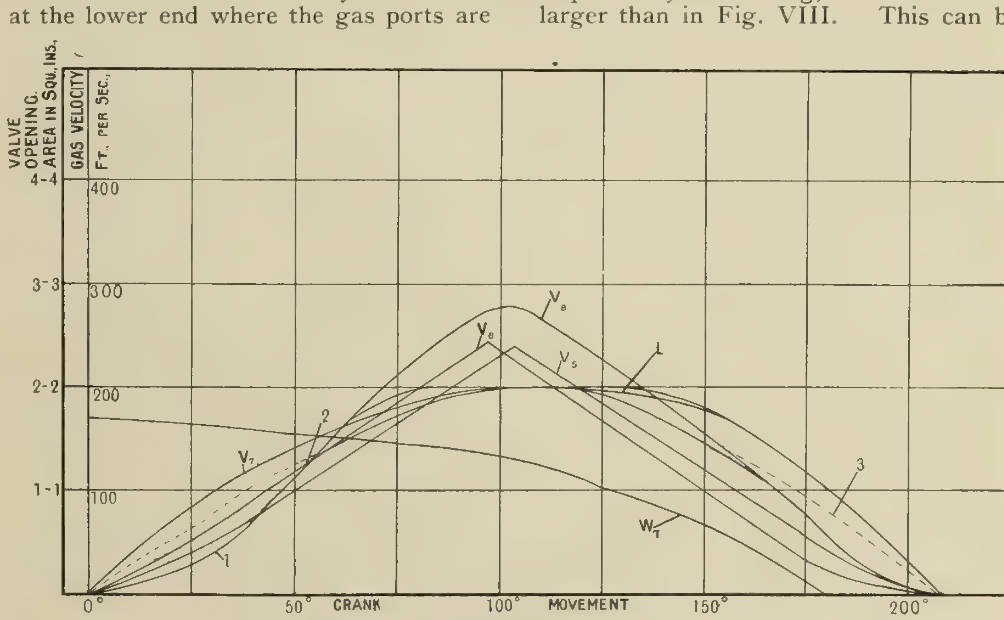


Fig. IXa.

sufficiently tight. Therefore, the part corresponding to "p" in Fig. VIII. can be made narrower here, thus increasing the size of the valve ports. Suitable dimensions are given in Fig. XI. [Fig. XI. was precisely similar in nature to Fig. VIII., which appears on page 146, except that the dimension 2.5/16in. was altered to become 2⅝in.—Ed.] The area of the port is

$$S = 2\pi \times \frac{52.5}{360} \times 1.687 \times 1.875 = 2.9 \text{ sq. ins.}$$

or 32 per cent. more than that of Fig. IX.

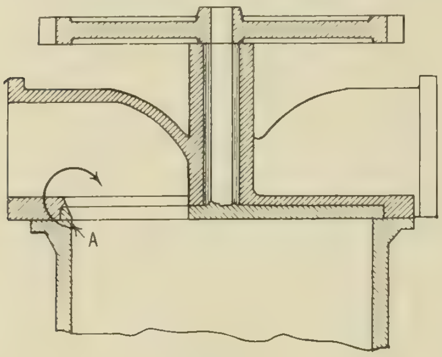


Fig. X.

and of the poppet valves. The lines v_2 in Fig. IX. represent the variation of the inlet opening with this larger valve, and curve W_2 the corresponding gas velocity. The area enclosed by lines v_2 is considerably larger than is obtained with poppet valves. It completely covers that of cam 1. The opening is slightly surpassed by the opening curves of cams 2 and 3 at certain points, but for the most part it exceeds them. With this rotary valve the ports open and close even more quickly than with cam 3, which further improves the construction. Of course the removable cylinder head has its disadvantages, being more expensive, requiring separate means for introducing water into the cylinder head (not shown in drawing) for cooling the valve seat, the exhaust connections, etc. However, the inlet and exhaust ports can be made more easily to register accurately with the valve port with a removable cylinder head than with integral heads. However, the inlet and exhaust ports can be made more easily to register accurately with the valve port with a removable cylinder head than with integral heads.

Larger port areas are obtainable if a rotary valve is made conical, and if made with similar dimensions, as in Fig. VIII., and a cone angle of 120 degrees, the port area would be that of Fig. VIII., divided by the cos of 30 degrees, or

$$S_{13} = \frac{2.2}{.86} = 2.54 \text{ square inches.}$$

Large port area can be obtained by using a double valve, consisting of two discs revolving in opposite directions at half crank-shaft speed. This construction carries with it the objection that a double valve requires double driving means, and the single valve already necessitates a number of gears and other parts. One of such double-valve constructions is represented in Fig. XII. The valve which bears against the cylinder head has a single port "a" (Fig. XII.), which registers in its motion with the inlet "i" and exhaust port "e" of the cylinder. In the outer valve, from the side of the explosion chamber, two ports are necessary, because the inlet period has to follow the exhaust closely. At the moment of exhaust closing the port "a" and the exhaust port of the other valve are in the position illustrated in Fig. XIII., in which the disc motion is transformed to a cylindrical for clearer understanding. If the same port should also be used for the inlet, another port "b" has to be provided in the outer valve, which port "b" could register with the cylinder inlet "i" at the same time with the port "a." The port "a" should not come to the cylinder ports "e" and "i" simultaneously with ports "b" and "c" at other moments than desired. That imposes a certain condition on the the size of the angles, which enclose these ports. For instance, the valve edge "m" has to pass the edge of the cylinder inlet port "n" before the edge of "o" of the other valve reaches it. This necessitates that :—

$$a + d + \frac{b}{2} < x, \text{ where } x = 360° - (2a + d + \frac{b}{2})$$

or $3a + 2d + b < 360°$.

The angle "d," which indicates the interval between exhaust closing and inlet

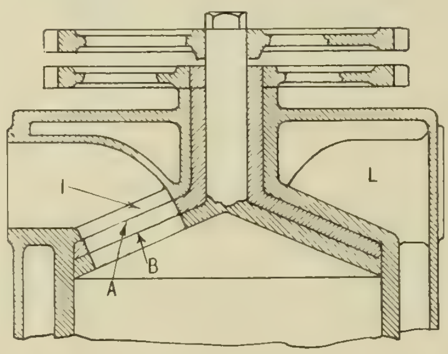


Fig. XII.

beginning, can be positive, zero, or negative, but is generally small.

From the above disparity it will be seen that the double-valve construction in question permits somewhat narrow limits for the exhaust and inlet, decreasing its practical value. If quarter-speed double valves are used, nothing can be gained in the size of port area, compared to a single half-speed valve, but as said, the

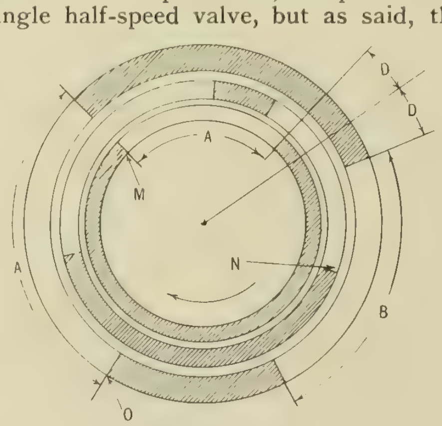


Fig. XIII.

complication in the drive increases.

The plan view of another double-valve construction is represented in Fig. XIV. Here the valve ports for the inlet and exhaust are separate, and are shifted from each other radially. Any desired motor timing can be obtained, but the sizes of the ports are small. With dimensions of Fig. XIV. the port areas in sq. ins. will be :—

$$\text{Exhaust } S_e = 2 \times \pi \times 1\tfrac{1}{4} \times \frac{112 \cdot 5}{360} \times \tfrac{3}{4} = 1.85$$

$$\text{Inlet } S_i = 2 \times \pi \times 2\tfrac{1}{32} \times \frac{105}{360} \times \frac{9}{16} = 2.1$$

Still another different construction of

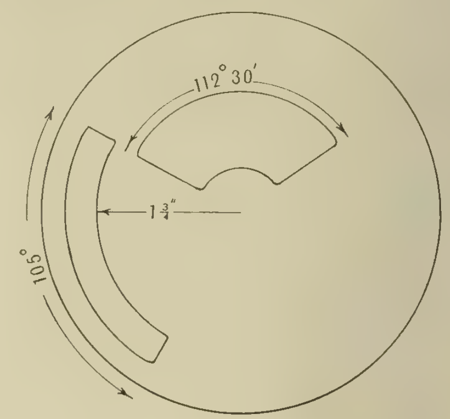


Fig. XIV.

half-time double valves with larger separate inlet and exhaust ports can be obtained by a combination of a cap valve and plate. Such a combination is given in Fig. XV. The outer cap valve has a small play inside the cylinder, and is ring-packed at the ends of its cylindrical surface, to insure greater tightness. The ports are located thus : one in its cylindrical part, the other in the flat top part. The second valve may also be made of the cap form, fitting inside the first one with or without expanding rings with correspondingly located ports; but in Fig. XV. the second valve is shown of the flat disc form, which can reasonably be made, if the port in the cylindrical part is sufficiently large, without increasing it by the double-valve motion. The port in the top part will be about twice the size of that which the conical cap valve permits. In other respects the construction after Fig. XV. appears to be better and more practical.

Other types of rotary valve constructions are those with revolving sleeves. These sleeves can be single, double and oppositely revolving; they can be placed inside the cylinder with the piston bearing against them, Fig. XVI., or outside the cylinder and its water jacket, etc. With the last-mentioned arrangement, good cooling of the valves is obtained, but the

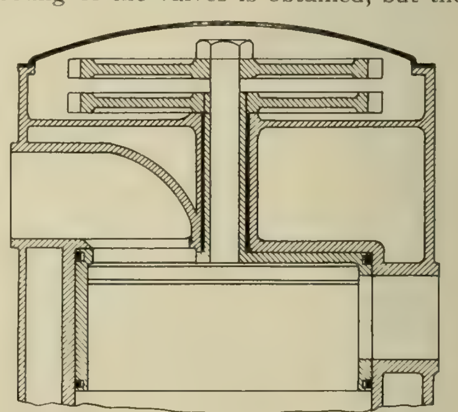


Fig. XV.

motor becomes bulky and of great length in the case of multi-cylinder construction. The size of valve ports in these constructions will, of course, greatly depend on the chosen dimensions. For instance, if we assume a motor after Fig. XVI. to have 25 per cent. compression space, counted from the total cylinder volume, this space will be about 2 inches

high, and the ports could possibly be made $1\frac{3}{4}$ inches high. The maximum area for the inlet port on the outside of the sleeve valve will be

$$S = 2\pi \times \frac{52.5}{360} + 2\tfrac{11}{16} \times 1\tfrac{3}{4} = 4.3 \text{ sq. ins.}$$

Similarly large valve ports will be ob-

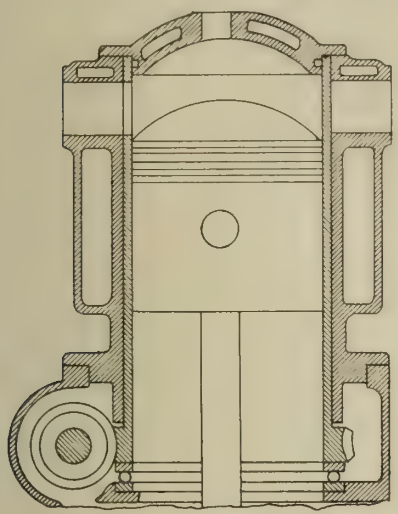


Fig. XVI.

tainable with motors with external rotary sleeves. In all the rotary valve systems, where the valves are driven at constant speed, the same triangular shape of valve opening diagram exists. The rotary sleeve valves have to meet from the practical side the same objections which were directed against the sliding sleeves (mentioned further on, when discussing the silent Knight motor). In the present case some of these objections have greater reason. For instance, if the valve ports remain exposed to the temperatures of the explosion chamber during the whole time, they are more apt to burn out or to work carbon particles in between the valve and the cylinder. The relative motion of the valve and the cylinder occurs at much higher velocity when the valve rotates continuously than when it slides up and down; and more care as to lubrication is required. Of course, the sleeves have not to fit tight into the cylinder, play being left for the lubricant. The sleeves themselves might or might not be provided with expanding packing rings, which in their turn might be round or elliptic.

The question of power required to drive rotary valves presents consider-

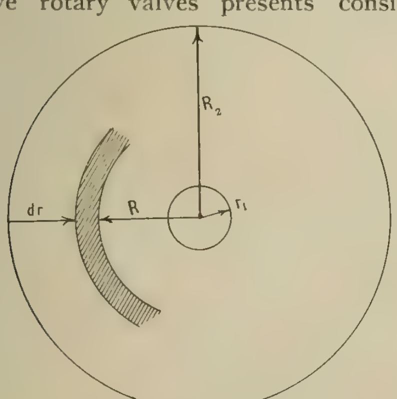


Fig. XVII.

able interest. The rotary sleeves, if properly made and lubricated, may consume very little power, particularly systems with external sleeves, in which the valves are not exposed to the explosion force or to the side pressure of the piston on the other side. Systems with disc or cap valves are subjected to friction, caused by the full force of the explosion, which might lead one to believe that a great amount of power is required for their drive. This amount of power can be figured out approximately as follows :—We will assume an evenly distributed pressure of the valve against the seat of a mean value of "p." One can say that the power necessary to overcome the frictional resistance of an elementary band (see Fig. XVII.) of a radius "r" and a width "*dr*" is

$$p \times f \times 2\pi \times r\,dr \times \frac{\text{r.p.m.}}{2} \times 2\pi \times r,$$

where "f" is the co-efficient of friction and the r.p.m. are expressed in motor revolutions for a half-time valve. For the total valve disc, considered uninterrupted by ports, the required driving power "R" per minute is

$$R = p \times f \times 4\pi^2 \frac{\text{r.p.m.}}{2} \int_{r_1}^{r_2} r^2\,dr$$

or, after integration

$$R = p \times f \times 4\pi^2 \frac{\text{r.p.m.}}{2} \times \tfrac{1}{3} r_2^{\,3}, \text{ because } r_1$$

is negligible.

Substituting $p \times \pi r_2^2 = P$, which in this case might be considered equal to the piston pressure, we will have

$$R = P \times f \times \text{r.p m} \times \tfrac{2}{3} \pi r_2$$

The co-efficient "f" may be taken as independent of the pressure and the velocity. Therefore, the pressure "P" can be substituted by the mean piston pressure the latter being expressed through the developed motor power as

$$P = \frac{\text{H.P.} \times 33{,}000}{E \times \text{r.p.m.} \times 2 \times \text{Stroke.}}$$

Here "E" expresses the mechanical efficiency of the motor, and the motor stroke "S" has to be given in feet. The power consumed for driving the valve thus, will be found to amount

$$R_{hp} = \frac{\text{H.P.} \times f \times \pi r_2}{S \times E \times 3}$$

For instance, our motor, if developing 10 h.p. on the brake at 1,000 revolutions per minute and having a mechanical efficiency of 83 per cent., with a valve friction of 0.1, consumes

$$R_{hp} = \frac{10 \times 0.1 \times \pi \times 2.5}{.5 \times .83 \times 12} = .525 \text{h.p.}$$

The double rotary valves after Fig. XIV. will consume double, or about 1 h.p., and valve Fig. XII. $\frac{1 \text{ h.p.}}{\cos 30} = 1.15$ h.p. However, these figures might be decreased about 15 per cent., because the valve does not represent a full disc, being provided with ports which diminish the area of its surface exposed to the cylinder pressure.

The co-efficient of friction was assumed here to be only 0.1, which generally would be the case if some reliable lubricant were provided; otherwise it will be greater. From all the foregoing there can be drawn the conclusion that the rotary valve systems, with valves placed directly inside of the explosion chamber, use for their drive not a small amount of power, and means to lessen it are desirable. . . . . .

Cylindrical or barrel types of rotary valves also give triangular opening diagrams. The simplest of these is represented in Fig. XIX. Here a valve "a" is formed with a straight central passage "b," through which the inlet and exhaust passages communicate with the interior of the cylinder at the proper moments. This valve rotates at one-quarter crankshaft speed. With an inlet of 210 degrees duration the valve channel can occupy at each end an arc of $26\frac{1}{4}$ degrees

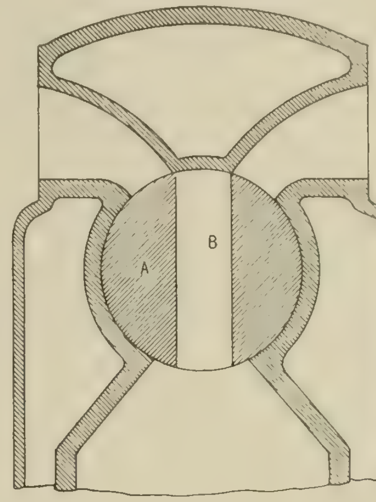


Fig. XIX.

if the greatest possible inlet opening is desired. In such a case the inlet port of the cylinder head will also extend over $26\frac{1}{4}$ degrees, and the exhaust port for 225 degrees duration over 30 degrees. For a cylinder of 5-inch bore the valve could be made about 3 inches in diameter, with a channel 4 inches long. The maximum inlet area being equal to that of the channel passage is approximately

$$S_{24} = \pi = 3 \times \frac{26.25}{360} \times 4 = 2.75 \text{ square inches.}$$

or 25 per cent. larger than with the poppet valves. Such valves sometimes are made to fit with a small play inside of their chamber and to be supported only by bearings at their ends. The play being filled by oil to give the required tight separation between the ports and the cylinders. Lengthwise the valve carries expanding rings, which separate the ports from the valve ends. The oil introduced in the play around the valve does not serve as a lubricant, because the valve has individual bearings. It has to be fed continuously, and as it is not caught back, the whole quantity is led into the cylinder and the exhaust, soiling them. In the design assumed, with 4-inch port length, the cylinder pressure acts on an area of about 10 square inches of the valve..

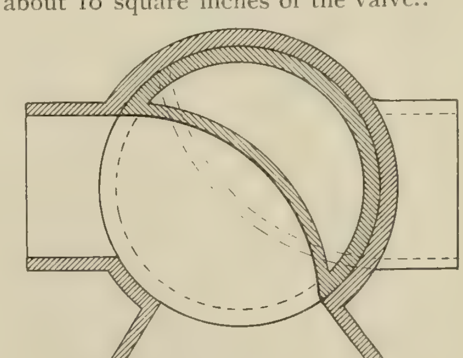


Fig. XX.

The cylindrical rotary valve systems which are probably the most promising for practical work are those comprising two separate single valves—one for the inlet, the other for the exhaust—and comprising two double oppositely rotating valves. A number of different arrange-

ments of this kind are possible. Each valve can be made according to Fig. XIX., Fig. XX. or Fig. XXI., and also Fig. XXII., or it may be constructed to serve simultaneously as a gas passage (Fig. XXIII). The use of this last-mentioned construction for the ex-

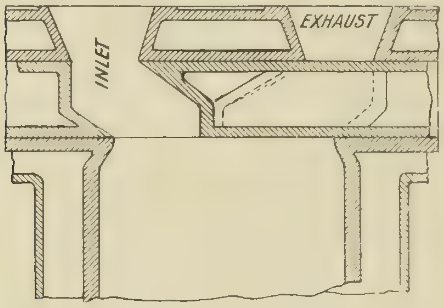


Fig. XXI.

haust does not seem to promise good results, at least, with valves of sufficiently large diameter for high-speed motors without any cooling means, as represented in Fig. XXIII. Generally these separate inlet and exhaust rotary valves can be made of smaller diameter than the combined valves, which give the advantage that their circumferential velocity is smaller at the same speed of revolution, and the problem of an easy-running valve is therefore simplified.

For a cylinder of 5-inch bore, and with half-time valves, each with a 2-inch outside diameter and a port length of 4 inches, the inlet port area will be approximately

$$S=\pi\times\frac{52.5}{360}\times 2\times 4=3.65 \text{ square inches.}$$

The same area is obtainable with a quarter-speed double valve and half of it with a quarter-speed single valve. This arrangement (Fig. XXIV.) allowing under assumed conditions an inlet port of only 1.83 square inches, will not be apt to give as good results in motor power as some other systems with bigger and wider ports. In Fig. XXIV. the inlet valve is shown to be double, but in the particularly referred-to motor construction the inside valve member has a special

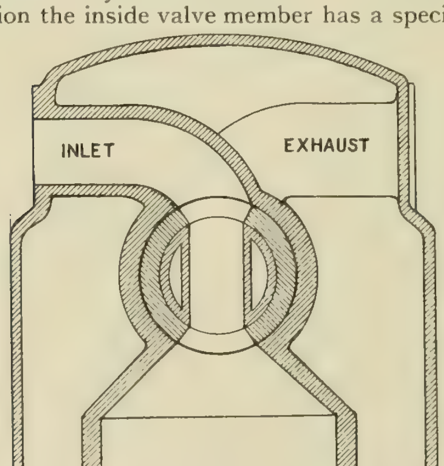


Fig. XXII.

purpose, which will be mentioned further along. If double half-speed valves are used—which are practical only if the valves serve also as gas passages—then the port area can be made still larger; and with a 2-inch valve it will be twice that of the single, totalling 7.3 square inches (counting on the cylindrical surface). Of course, such a valve with an opening equal to 37 per cent. of the piston area would give a very easy gas flow if the maximum choking effect were not transferred to another place. The inside valve of 2-inch outside diameter can give a gas passage of only $1\frac{3}{4}$ inch diameter, or 2.4 square inches area, if the gas flows to the valve port in one direction. The conditions become better when the gas flows to this port in both directions; that is, if the gas enters the interior of the inside valve from both ends. In this case only half of the total quantity of gas flows from each side through the area of 2.4 square inches. Owing to this restriction of the gas passage area, the big-sized valve ports are practically valueless because they form only a relatively short length of the gas path. Here conditions are the exact opposite of those usually encountered with poppet valves.

The possibility of big ports for the inlet permits of reducing the period of their opening. The shortening of the inlet period will in turn slightly diminish the port area, and by further shortening the port length this area may be reduced to about 5 square inches, which equals 25 per cent. of piston area, and will be sufficient for the motor at any reasonable speed. The lines v_3 in Fig. IX. constitute a diagram of a 5 square inch valve

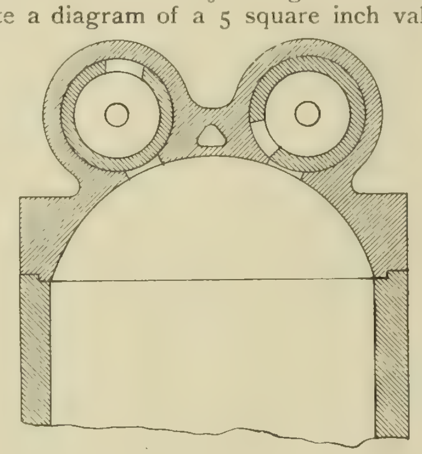


Fig. XXIII.

opening for an inlet period of 210 degrees. This arrangement would be quite satisfactory from the standpoint of motor power, if similarly large exhaust opening and passages were used. But, as stated, it is hardly advisable to subject the exhaust valves to additional heat by making them outlet passages for the spent gases. Something would be gained in this respect by providing auxiliary exhaust ports, controlled by a separate valve or by the piston, through which the hottest gases escape. On the other hand, a half-time valve 2 inches in diameter with a port $4\frac{1}{4}$ inches long giving a 225-degree exhaust period has an opening area of 4.15 square inches, which is probably as much as can be obtained without introducing great complication, and which, besides, is sufficient for most purposes.

Generally speaking, the combination of a single or double-inlet valve serving also as a gas passage, with a half-time exhaust valve, with an elbow or curved channel, cooled from the outside, and in case of larger size also from the inside, appears to the writer to be the most practical arrangement (see Fig. XXV.). Auxiliary exhaust ports uncovered by the piston could be added if desired. In this construction the exhaust valve passage is opened to the cylinder before the exhaust time actually begins, which, though not desirable, is no great disadvantage.

From what has been said concerning rotary valve systems in general the conclusion may be drawn that with but few exceptions they surpass the poppet valves in respect to size and character of valve openings. Conical valves can also be used in place

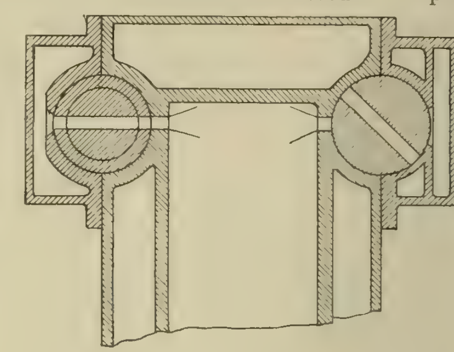


Fig. XXIV.

of the cylindrical ones, but the kind of port openings obtainable will not change greatly. The conical valves require more attention to properly fit their seats. The valve taper is generally small, and care is required to prevent the valve from being pressed against the seat so as to cause it to stick.

The more practical and reliable rotary valve systems, of those referred to in the past, may become strong competitors of the poppet valve, especially if they are connected with mechanical simplicity. In this respect, it should not be sufficient to make the motor look simple by concealing all complication in the crank-case, or by special covers. The proper placing of the valves so as to enable the use of the simplest driving means according to each individual case, will be considered of great importance. It does not matter whether the simplicity is obtained by placing valves in the cylinder head, at its sides, or elsewhere. However, it is advisable to choose the valve place with a view to getting the most favourably shaped explosion chamber. In multi-cylinder motors some barrel valve systems will be found to be the least complicated, because a single driving means may operate the same valves of the whole cylinder row. The disc and sleeve rotary valves for

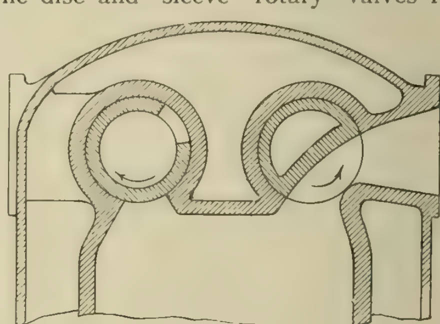


Fig. XXV.

multi-cylinder motors are less simple in their drives, but good results can be obtained by transmitting the drive to only one valve, from which the others are driven in train. This kind of drive with valves in adjoining cylinders revolving oppositely also conforms well with simplicity in location of ports. Generally speaking, it will be found that far more difficulty arises with that part of the valve drive which serves as an intermediate link between the valves and the motor crankshaft. Here the complication increases further if motion has to be transferred to the motor auxiliaries like water and oil pumps, ignition devices, etc. This part of the valve mechanism represents a

subject too spacious in itself to be treated upon in the present article, where it will suffice to have mentioned it briefly. Through the desire to simplify the valve drive probably originated the rotary valve systems, in which a single valve serves for more than one cylinder. The valve has one inlet and one exhaust passage, which register with the cylinder ports at certain moments. These moments are in

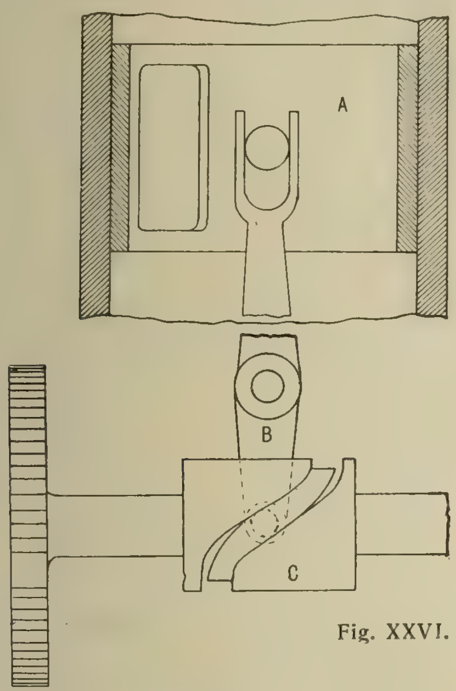


Fig. XXVI.

fixed relation with the cycles in both cylinders, which determines the position of the cylinder ports to each other. The valves are preferably made cylindrical and, depending on their general dimensions, will give openings similar to other systems with the same valve form. These single valves can be located over the cylinder head, at either side, with horizontal or vertical axis, etc. . . . .

The motion of sliding-sleeve valves closely resembles that of the oscillating, the rotating movement being substituted by a straight one. Single sleeve valves with a regular motion cannot be used here for the same reason as was mentioned in the foregoing. With a device impelling them with required irregular motion, the same sort of valve diagram can be obtained as with the system in Fig. XXVI. As an intermediate valve system between the single and double-sliding sleeves, one can be interposed, which uses the main motor piston as a second valve member. There is little to say concerning this combination, as it also restricts the motor timing.

The double-sliding sleeve system is the most interesting in its class, because it includes the much-spoken-of silent Knight motor. It should be of great interest to investigate the sliding-valve motion of this particular construction, though, as pointed out before, it will not differ greatly from that of the double oscillating valve. For the sake of uniformity we will assume a motor of 5-inch bore and 6-inch stroke, and choose our own valve dimensions; so far as the writer knows, the nearest actually built motor is 124 × 130 mm. We may assume the eccentricity of the sleeve-operating eccentrics to be $\frac{1}{2}$ inch each, the length of the connecting-rods to the inner sleeve $2\frac{1}{2}$ inches, and that to the outer sleeve 4 inches. The height of the inlet and exhaust slots in the sleeves may be made $\frac{1}{2}$ inch and their length $5\frac{3}{4}$ inches, which dimensions are pretty nearly the same as those in the 124 × 130 mm. motor. The inlet begins when the piston is at the upper end of the stroke. At this moment the inner sleeve is in its lowest position, and the corresponding eccentric at its lower dead centre. The outer sleeve with its eccentric moves 90 degrees behind that of the inner one. The valve opening may be best found in a graphical way; that of the inlet is represented in Fig. IXa. by curve v_6. This curve v_6 is as accurate as it could be located in a drawing of twice the actual side of the mechanism. It does not show a great difference from curve v_5, and therefore all that has been said about this can be repeated in regard to the silent Knight motor. The shape of curve v_6 confirms the conclusion to which we arrived formerly, stating that the double-valve mechanism operated from eccentrics has no advantage over other valve systems, including some poppet valve types, as regards rapidity of port opening, closing and opening. This circumstance should refute the claims often produced in favour of the Knight motor in this respect. It is true that this motor has many advantages over the common poppet valve type, notably in respect to valve openings; but it remains slightly less favourable for motor power than some rotary valve systems. It ranges with those which give a maximum port area a little smaller than its own; through the shape of valve ports nothing being gained. As to the claimed possibility to make the Knight valve ports as large as the main piston area, it can be said that such an abnormal motor construction will hardly ever be required. The port size obtainable with some of the other systems referred to in the present article is more than sufficient for any reasonable purpose. The conclusion which we have deduced theoretically from the valve diagrams, stating that the Knight and similar motors have no great advantage over poppet valves, as far as motor power is concerned, has been confirmed in actual practice also. Some poppet valve motors have shown as much or even a greater rate of power development, together with not less flexibility.

In the current literature more has been said about the silent Knight motor than any other silent valve system. It is proper to mention some of the points made. Prominently stands the consideration of uncertainty in sleeve lubrication. The thickness of the sleeve added to that of the cylinder walls, and the oil films between, which are heat developers, should render a good cooling of the motor more difficult. Uneven thickness of cylinder walls resulting from core shifting when casting should cause unequal distortion of the sleeves. The wear of the sleeves should be rapid and the amount of power necessary to operate the sleeves should also result in rapid wear in the valve connecting-rods, etc. Carbon particles are easily worked in between the sleeves, also causing wear; and the exhaust ports are liable to burn out in the thin sleeves, etc.* These objections met replies as follows: In regard to sleeve lubrication, it was pointed out that the sleeves were made with a play of 3-1,000 inch to allow for oil. The lubrication system has proved reliable in use; and in tests which were made a piston seized once, but not the sleeves. The sleeves transfer the heat from the cylinder inside on to a greater surface by their motion, making the cooling more effective. No wear was perceptible in the sleeves after runs of thousands of miles. The sleeves are long and present a very large bearing surface, resulting in their working under a small specific pressure. They do not require much power for their operation—less than poppet valves, and the amount of it transferred through the small connecting-rods is lessened because the piston in its motion assists that of the sleeve. (However, in reality the piston counteracts the sleeve motion as much as it assists it.) In fact, the valve operating mechanism stands the work well. The exhaust sleeve ports become exposed to the explosion chamber heat after the piston has travelled through a part of its downward course, when the temperature is no longer extremely high. To take care of carbon particles capable of working out the sleeve ports, their edges are eased off. The sleeves improve balance.

The general mechanical construction of the silent Knight motor valve-drive has also been mentioned frequently, particularly the drive from one half-time shaft, with connecting-rods acting at one side of the sleeves, tending to tip them off from the concentric position with the cylinder, which the provided play allows. This objection is important as far as it concerns the bearing surface of sleeves and cylinder, but will not interfere with the good functioning of the motor piston. Moreover, if one prefers an increase in motor complication the sleeves can be driven symmetrically from two half-time shafts placed on opposite motor sides.

Passing on now to piston valve systems, let us first consider the one shown in Fig. XXVII. There are separate piston valves for the inlet and for the exhaust. During the downward motion the upper edge of the valve uncovers a circular opening made in the walls, which opening serves as inlet or exhaust port. The valves are supposed to receive motion from a half-speed shaft, either through

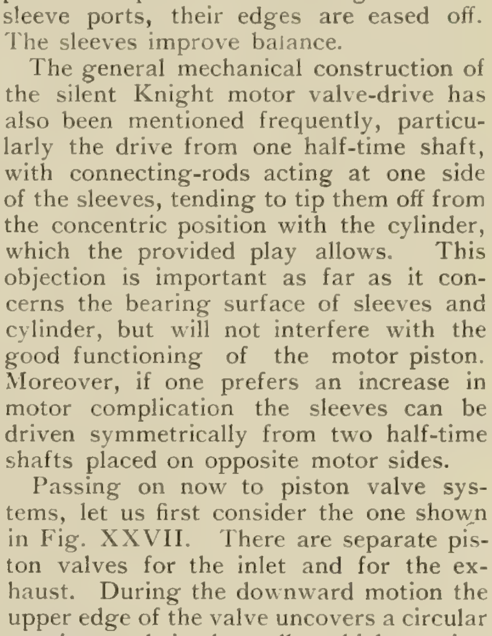


Fig. XXVII.

eccentrics or cranks with connecting-rods. The diameter of the valves may be assumed to be 2 inches and their total stroke 2 inches. The inlet extends over 105 degrees rotation of the half-speed shaft, and to obtain the greatest possible opening the piston must uncover the port 52 degrees 30 minutes, before reaching its lowest position. The travel of the piston valve previous to reaching this point may be obtained from the equation:

Piston travel $X = 2(1 - \cos A) \pm l(1 - \cos B)$

The designations being the same as for Fig. V. A simpler formula of sufficient

* These are very extravagant criticisms.—ED.

accuracy for practical use is the following:

$$X = 2\ (1 - \cos A + \tfrac{1}{2}\lambda \sin^2 A)$$

We may assume that $\lambda = 1/6$, corresponding to l=6 inches. Substituting the actual values and writing the equation in such a manner as to give the piston travel, counting from its lowest position, it is found that:—

$$X = 1 \times (1 - \cos 52^\circ\ 30' - \tfrac{1}{2} \times \tfrac{1}{6} \sin^2 52^\circ\ 30')$$
$$= 0.34'' \text{ appr.}$$

If the piston valve has no packing rings, then the port can extend clear around the circumference, and its area will be:

$2 \times \pi \times 0.34 =$ approx. 2.14 square inches.

In case the valve is provided with rings to ensure tightness, the port must be provided with bridges to prevent these rings from falling into it, or from sticking at the port edges, and the port area will be greatly reduced. The exhaust port will be a trifle large, having a height of almost 0.4 inches for a period of opening 225 degrees crank motion, but the area still remains smaller than for poppet valves, if the piston has rings. Curve v_7 in Fig. IX*a*. represents the variation of the opening of valve with a maximum port area of 2.14 square inches, and curve w_7 gives the corresponding theoretical intake gas velocity. It is noted that this gas velocity is almost constant during the first half of the suction stroke, and then gradually drops to the end, it being assumed that the inlet begins together with the downward cylinder-piston movement.

A constant gas flow has great advantages. It eliminates speed variations in the intake pipes, causing vibration and inertia losses, and it also conforms well to the petrol flow through the carburettor nozzle. The effect of inertia on the petrol flow becomes less noticeable, if the air flows through the carburettor at a more nearly constant rate. In this respect the piston valves are superior to other systems, and with fairly large port openings, they possess real practical advantages. These port openings can easily be increased by increasing the valve stroke. It is also possible, by means of special constructions, to open the port during the movement of the valve away from the half-time shaft. In that case the last member in the parenthesis of the equation for the piston travel will change its sign, and this calculated travel will then be 0.445 inch instead of 0.34 inch, or about 31 per cent. greater. Besides, the opening curve v_7 of a piston valve giving the same size of opening as a poppet valve, has a more favourable shape, and encloses a bigger area than the curve of the latter. However, mechanically the piston valve can hardly be considered a simplification over the poppet valve. In order that the motor may run without vibration, the piston valves have to be properly balanced, together with their mechanism. The additional bearings of the piston valve mechanism involve increased cost of construction and care in use. It must be considered that the pressure on these bearings is equal to that on the piston valve itself. Special "balanced" valve constructions can be provided in which the pressure on the valve is equalized.

Piston valves also allow a variety of arrangements, all more or less similar from a mechanical point of view. The valves can be placed on the sides of the cylinder, on its head, etc. In some piston valve constructions the exhaust valve piston is made of greater diameter than the one for the inlet. This exhaust valve piston will be on its downward stroke with the maximum velocity when the explosion occurs in the cylinder. The inlet piston is almost at its dead centre at this moment. Such being the case, the exploded gases acting on the exhaust piston will transmit power impulses to the half-time valve shaft, thus driving it. In this manner the valves become self-driven, and their gearing to the motor crankshaft is merely necessary for maintaining the correct speed and timing, and to carry their excess of power to it. It is being claimed that this results in motor power increase. However, one should consider that the possible increase in motor power is not due to the fact that two pistons, the cylinder-piston and the valve-piston, move under the explosion gas pressure. With the same quantity of exploded gases the motor will be capable of producing only the same amount of power, independently of whether the expanding gases act on a smaller or a bigger moving area, respectively one or two pistons. If a greater moving area is available in the cylinder, the rate of gas pressure drop therein is increased, and consequently the mean cylinder pressure becomes less. There is possible gain in motor power with the referred to piston valve system, because of its self-driving, whereby no power is lost in the gearing from the crankshaft. But, on the other side, the piston generally receives power impulses greater than required for its drive, and the power excess is transmitted to the motor crankshaft through the gearing. As this latter amount of power also is greater than that possibly necessary for the valve driving, the conclusion is that nothing is gained.

All the piston valve systems are more or less complicated mechanically, and are not cheap. Their advantage over the poppet valve lies merely in the silence and positiveness of the valve operation. Both this silence and positiveness are of great advantage to all silent valve systems, when compared with poppet valves. The positive opening and closing of valve ports generally results in a much more flexible motor and one capable of attaining higher speed. In poppet valves even slight inaccuracy in cams and lifters caused by defective workmanship or wear, results in the valve timing not being the same for all cylinders and not what it was intended to be. This affects the motor power, its balance and silent running. A poppet valve motor intended to run at high speed will be provided preferably with cams of a somewhat elaborate shape, to ensure good cylinder filling by fresh charge. But often this fancy cam shape will become useless, resulting also in an entirely wrong motor timing, if the valve springs should prove to be not strong enough at high speed. The matter is different with silent valve systems. In most of them the ports can be finished accurately to their size in the valve as well as in the cylinder. The timing will be correct and even for all cylinders, and it will start to get out of order only after the parts are considerably worn. The correctly made drives of the silent valve systems also are practically noiseless and remain so as long as the wear in them is taken up. They can be well designed in a manner that almost no place can be found where noise could be generated.

As far as general cost of motors with different valve systems is concerned, one can merely guess at present. The rotary valve systems appear to be the cheapest, eliminating a great many parts otherwise found in motors. The piston and sleeve valve systems, on the other hand, are to be considered more expensive than the poppet because lots of additional work is required with them. The piston valve motors will probably be the heaviest, owing to the solidity of the parts required. Sliding and rotary valves, together with oscillating valves, stand next in regard to motor lightness. They may compete for the place with poppet valve motors. As far as can be seen ahead, the several rotary valve systems (particularly some of them), can be manufactured the most cheaply, can be made with the least weight, and can show motor power equal to any other valve system, besides being of much promising reliability.

Conclusions.

The following conclusions may be drawn from the foregoing discussion:

The quantity of fresh charge drawn into the cylinder during the inlet period depends as much on the character of the valve opening diagram as on the actual maximum size of the opening.

A long period of inlet opening is necessary only in so far as it helps to obtain big inlet ports and better cam shapes. A late inlet closing, up to certain limits, has no great direct influence on the motor power under average conditions, because the additional charge entering during this period is small.

The openings obtained with poppet valves are smaller and their diagrams of openings are also less favourable than with most of the other valve systems mentioned in this article. Poppet valves of large diameter and small lift are preferable to poppet valves with higher lift and small diameter, because, with the same maximum size of opening, the larger valves, depending on the cam shape, will have a better opening diagram.

Rotary valves driven at constant speed all show the same character of opening diagram, and the one which, besides giving the desired size of openings, permits the simplest and most reliable construction, is therefore to be preferred.

The Knight valve must be considered superior to the poppet valve, but not superior to some rotary valve constructions.

From the standpoint of efficiency (which latter is favoured by a hemispherical explosion chamber) poppet and piston valves located in the cylinder head at an angle are nearest to the ideal. The silent Knight motor and motors with certain arrangements of cylindrical rotary valves may be considered next best. A good form of explosion chamber is obtained also with the other rotary valve systems with cylindrical or disc valves. Motors with poppet or piston valves located in side pockets have the greatest combustion chamber cooling surface. These constructions also are unfavourable to quick ignition of the charge and must be considered the worst from the standpoint of thermal efficiency.

THE BALANCE OF ROTATING CYLINDER ENGINES.

By H. Grinsted.

MOST writers in the motor Press have, so far, passed over the question of the balance of the rotary engine, being led, it seems, by the idea that as there is no absolute reciprocating motion, the engine is balanced as simply as a rotating disc. The following consideration will show that this is not the case.

The engine can be divided into two rotating systems :—

1. Pistons and connecting rods.
2. The remainder of the moving parts, i.e., cylinders and crankcase and everything fixed upon them.

The second system can be balanced in the ordinary way for masses in the same plane rotating about a fixed axis perpendicular to their plane, with a common angular velocity. For perfect balance the mass centre of the whole must lie in the axis of rotation.

The first system presents a different case. Supposing the cylinders to rotate

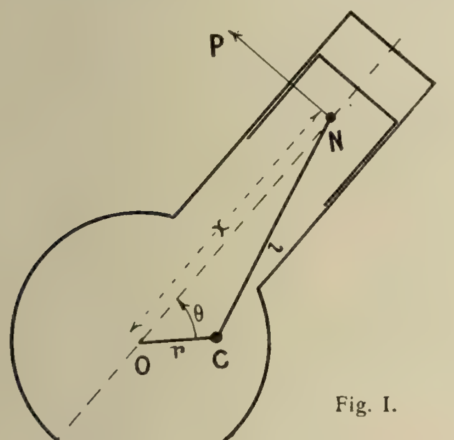


Fig. I.

with constant angular velocity, then the pistons and connecting rods will have varying angular velocities about the crank pin, which is their axis of rotation, and, consequently, varying centrifugal forces, and it is not at all obvious that the system is in balance. The inertia force due to any moving mass is equal and opposite to the force required to give the mass its acceleration, whatever that acceleration may be; so that, in determining the unbalanced force, the acceleration of the mass must first be found.

We will proceed to determine the acceleration of a piston of an engine whose cylinder revolves with constant angular velocity ω.

In Fig. 1. O represents the axis of the crank shaft, C the crank pin, N the gudgeon pin, r = crank radius, l = length of connecting rod.

First suppose that the crank rotates with an angular velocity ω clockwise, the cylinders being fixed, then the piston will reciprocate in the ordinary way.

Let the angle turned through by the crank from the cylinder axis be θ, and the distance of N from O be x; then, as for the ordinary reciprocating engine.

$$x = r\cos\theta + l - \frac{r^2}{2l}\sin^2\theta \text{ Approx}$$

the positive direction being from O to N. And the velocity of the piston

$$\frac{dx}{dt} = -\omega r\sin\theta - \frac{\omega r^2}{2l}\sin 2\theta$$

positive from O to N

Now suppose the whole mechanism has impressed upon it an angular velocity of ω about O in a counterclockwise direction, so that all parts of the mechanism are now given a velocity ωy, where y is the distance of any part from O, in addition to their previous velocities. The cylinders will now rotate with an angular velocity ω counterclockwise, the crank will remain stationary and the conditions are now those of the rotating cylinder engine.

The piston now has

(1) its velocity—$\omega r\sin\theta - \frac{\omega r^2}{2l}\sin 2\theta$ along ON, that is, at θ with the crank (2) a velocity $\omega \times$ ON, *i. e.* $\omega r\cos\theta + \omega l - \frac{\omega r^2}{2l}\sin^2\theta$ along N P, perpendicular to O N, in the direction of rotation of the cylinders, that is, at an angle of $(\frac{\pi}{2} + \theta)$ with the crank.

(Since θ is now being measured positively in a counterclockwise direction from the crank, and ω has been reversed, velocity (1) does not change sign.)

Resolving these two component velocities of N on to co-ordinate axes respectively parallel and perpendicular to the crank, we have

Component of velocity of N parallel to crank

$$= \left(-\omega r\sin\theta - \frac{\omega r^2}{2l}\sin 2\theta\right)\cos\theta$$
$$+ \left(\omega r\cos\theta + \omega l - \frac{\omega r^2}{2l}\sin^2\theta\right)\cos\left(\frac{\pi}{2} + \theta\right)$$
$$= -\frac{\omega r^2}{l}\sin\theta\cos^2\theta - \omega r\sin 2\theta - \omega l\sin\theta$$
$$+ \frac{\omega r}{2l}\sin^3\theta \text{ — — — — (a)}$$

Component of velocity of N perpendicular to crank

$$= \left(-\omega r\sin\theta - \frac{\omega r^2}{2l}\sin 2\theta\right)\sin\theta$$
$$+ \left(\omega r\cos\theta + \omega l + \frac{\omega r^2}{2l}\sin 2\theta\right)\sin\left(\frac{\pi}{2} + \theta\right)$$
$$= -\frac{3\,\omega r^2}{2l}\sin^2\theta\cos\theta + \omega r\cos 2\theta$$
$$+ \omega l\cos\theta \text{ — — — — (b)}$$

By differentiating these component velocities with regard to time, the acceleration components along the two co-ordinate axes are obtained. Thus, acceleration of N parallel to crank

$= \frac{d}{dt}$(a) which reduces to

$$-\frac{\omega^2 r^2}{2l}\left(\frac{9}{4}\cos 3\theta - \frac{1}{4}\cos\theta\right) - 2\,\omega^2 r\cos 2\theta$$
$$-\omega^2 l\cos\theta$$

And acceleration of N perpendicular to crank

$= \frac{d}{dt}$(b) which reduces to

$$-\frac{\omega^2 r^2}{2l}\left(\frac{9}{4}\sin 3\theta - \frac{3}{4}\sin\theta\right) - 2\,\omega^2 r\sin 2\theta$$
$$-\omega^2 l\sin\theta$$

These two components of the acceleration of N would clearly be the components of an acceleration represented by the sum of four vectors :—

Oa of length $\omega^2\left(l - \frac{r^2}{8l}\right)$ at $\pi + \theta$ with the crank

Ob ,, ,, $2\,\omega^2 r$,, $\pi + 2\theta$,, ,, ,,

Oc ,, ,, $\frac{9\,\omega^2 r^2}{8l}$,, $\pi + 3\theta$,, ,, ,,

Od the projection on a line perpendicular to the crank, of a vector of length $\frac{\omega^2 r^2}{4l}$ at θ with the crank

So, for convenience, we can consider the acceleration of the piston as made up of the four parts given above; and for any value of θ the resultant acceleration can be found graphically by setting out those vectors to scale and combining them by a vector polygon as shown in Fig. II.

Now the path of N is a circle about the

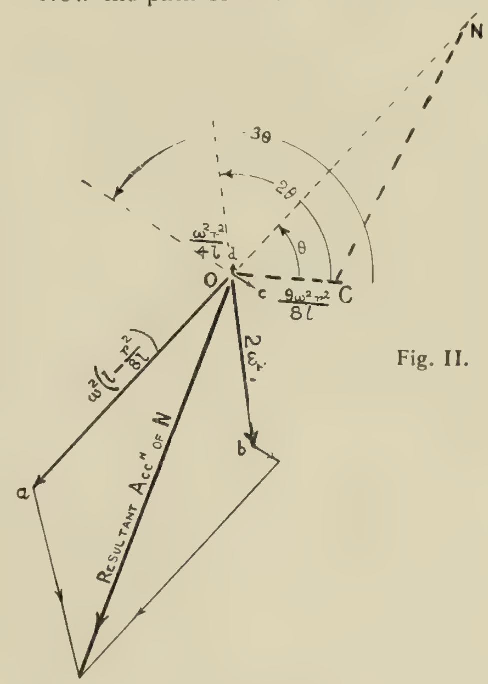


Fig. II.

crank pin, the centre of the big end of the connecting rod being fixed, so the motion of every point on the connecting rod is similar to that of the gudgeon pin N, and we may therefore consider the piston and connecting rod replaced by an equal mass concentrated at their mass centre, distant m (say) from the crank pin, and having similar accelerations and velocities to those of N, but diminished in the ratio of l to m.

As the cylinder rotates, the vector Oa rotates with it at the same speed, and the inertia force corresponding to this portion of the acceleration can be balanced by a mass placed on the cylinder or crankcase; but the vectors Ob and Oc rotate respectively at twice and three times the speed of the cylinder, and the corresponding forces cannot therefore be balanced by any masses rotating with the cylinder. The force corresponding to Od will be reciprocating, and cannot be balanced by any single rotating mass. This force is small compared with the primary and secondary, but if all but this reciprocating force are balanced, its effect will be noticeable.

So the condition for perfect balance in a single cylinder engine is that the mass-moment of the piston and connecting rod about the crank pin shall be zero, and the solution of the problem for a multi-cylinder engine can be deduced from the single cylinder case. If there are n cylinders,

numbered 1, 2, etc., in the direction of rotation (counter-clockwise in above analysis), then if the angular displacement from the crank position is θ_1 for No. 1, θ_2 for No. 2, etc.

$$\theta_2 = \theta_1 + \frac{2\pi}{n} \quad \theta_3 = \theta_1 + \frac{4\pi}{n}, \text{ etc.}$$

The angle between the primary vectors for two adjacent cylinders will always be $\frac{2\pi}{n}$, that between corresponding secondary

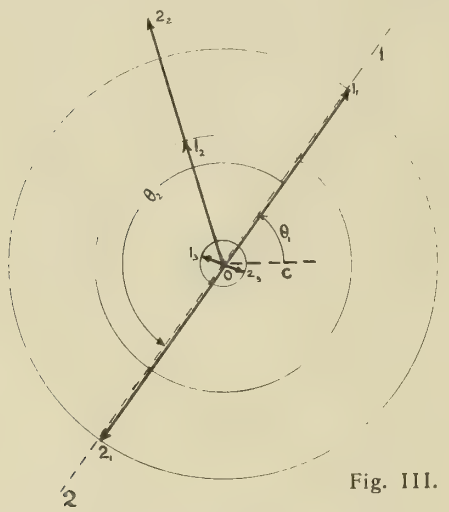


Fig. III.

vectors will be $\frac{4\pi}{n}$, and that between the two tertiary vectors will be $\frac{6\pi}{n}$. In the following examples it is assumed that all the pistons and connecting rods are similar, and that all the latter pivot directly on the fixed crank pin.

In the diagrams, OC in a heavy broken line, is the crank position; the cylinder axes are shown in lighter broken lines, and the full lines represent vectors, the latter being numbered the same as the cylinders to which they correspond with the suffixes 1, 2, 3, to indicate primary, secondary and tertiary respectively. The primary reciprocating component is not shown in the diagrams.

In the two-cylinder engine the primary and tertiary forces balance, but the two secondary vectors are coincident, being at

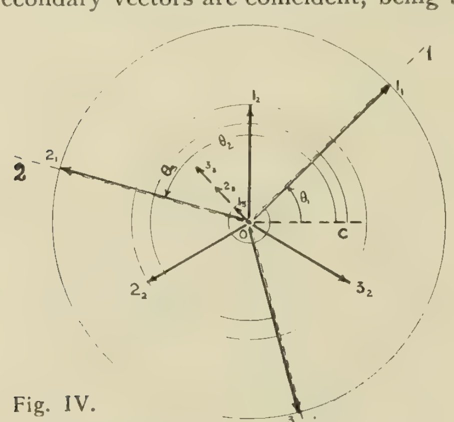


Fig. IV.

an angle of $2 \times \frac{2\pi}{2}$ *i.e.* 2π with one another; hence there is a considerable unbalanced force rotating at twice the engine speed. (See Fig. III.)

In the three-cylinder engine (Fig. IV.) the primary and secondary forces balance, but the three tertiary forces are coincident, since the angle between any two is $3 \times \frac{2\pi}{3}$ *i.e.* 2π. Hence the three-cylinder single crank engine will have a tertiary unbalanced force three times the value of that for one cylinder. In neither of these two cases can balance be effected by alteration of the masses of the pistons and connecting rods, unless the mass moment of each piston and connecting rod combination is made zero.

By arranging a two-cylinder engine with cylinders at 90° instead of 180°, the secondary forces can be brought into balance, but the tertiary forces are then left unbalanced. The primary rotating forces can always be balanced by a mass attached to the rotating crankcase; the reciprocating forces, however, remain unbalanced (Fig. V.).

In engines with more than three cylinders, each set of forces will be found to balance. Let us take, for example, a seven-cylinder engine. The force diagram for each set of seven vectors consists of seven equally-spaced vectors, of equal length, and the corresponding vector polygons are three regular heptagons. The reciprocating forces, being the projections, on one line, of seven equal and equally-spaced vectors, are also in equilibrium. Figures VI. and VII. show the secondary and tertiary force systems (not to the same scale) for one position of an engine.

In Fig. VIII. one position of a seven-cylinder engine is drawn, and at each piston is drawn a vector representing its

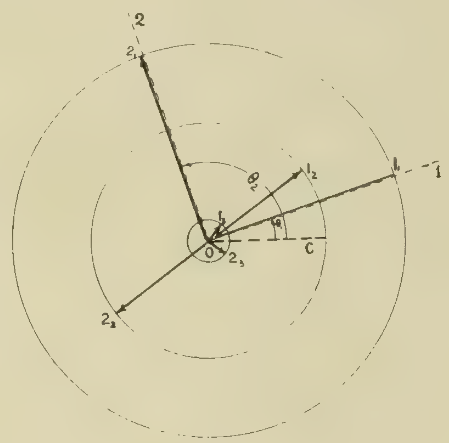


Fig. V.

acceleration to scale. The values given are for a speed of 1,000 R.P.M., the crank radius being 60 mm., and the connecting rod 200 mm. long. The vectors were obtained by vector addition of the primary, secondary, and tertiary components for each cylinder, the small reciprocating component being neglected. The relative values of primary, secondary, tertiary, and maximum of reciprocating components are 100, 60.65, 10.22, 2.27 respectively.

The accelerations can be converted into inertia forces by changing the scale and reversing their directions, if all the connecting rods and pistons are similar. However, if one connecting rod were of larger mass moment than the others, in converting the accelerations into forces not only would the scale be altered, but one vector would be lengthened, and the balance would clearly be upset.

On account of the practical difficulty of obtaining sufficient bearing surface for each big end on a single crank pin, the arrangement usually employed in multi-cylinder single crank engines consists of one large connecting rod, with a cage at its big end carrying pins, to which the other connecting rods are attached. The extra stresses in the "king rod," due to this arrangement, make it necessary to enlarge the section of the rod, and thus cause it to have a larger mass moment about the crank pin than the others. This will disturb the balance of primary, secondary, tertiary and reciprocating forces.

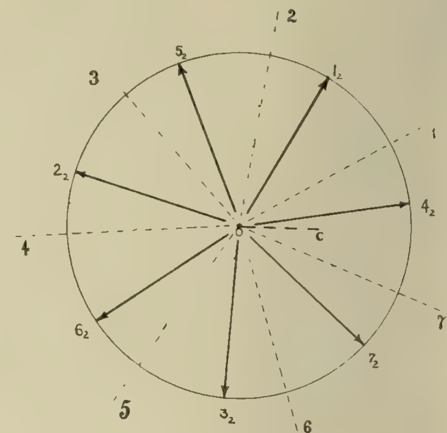


Fig. VI.

The primary system, except for the reciprocating components, can be put in balance by a suitable mass attached to the cylinders, but the secondary and tertiary parts cannot both be balanced for all positions; if the secondary force polygon is closed by giving two other pistons or connecting rods extra mass, the tertiary will be open and vice-versa. It is difficult to foresee the effect of the auxiliary crank pins, but it would be a remarkable coincidence if it were found to balance the effect of the heavier king rod.

Up to the present we have confined our

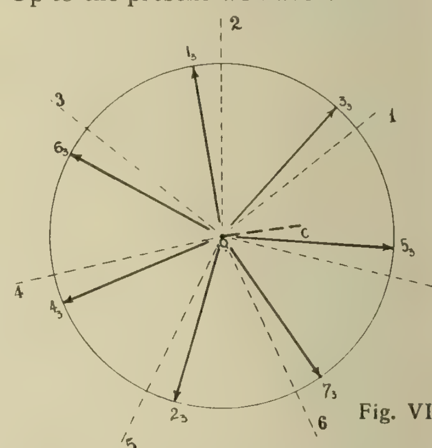


Fig. VII.

attention to single crank engines, and the conclusions arrived at can be applied to the solution of the problem when more cranks than one are used. Each crank

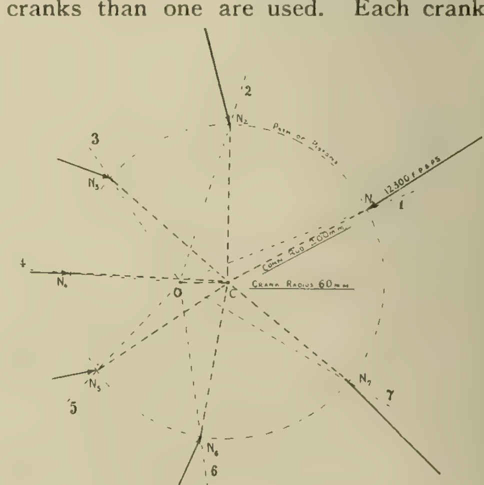


Fig. VIII.

can be considered separately, and the force diagrams combined. We will deal with two-cylinder, four-cylinder, and six-cylinder two-crank engines.

A two-cylinder engine, with cylinders and cranks at 180° can be considered as two single cylinder engines 180° out of phase. Since all the forces at one crank are therefore always opposed to those at the other, there can be no resultant force, but there will be primary, secondary, and tertiary couples. A four-cylinder engine can be considered as two two-cylinder single crank engines, placed side by side. If diametrically-opposed cylinders operate on the same crank, the cranks being at 180°, then the primary and tertiary forces at each crank are in balance between themselves, but the secondary force at one crank is always in the same direction as that at the other; hence there is a resultant secondary force, but no couple.

If adjacent cylinders at 90° operate on one crank, the cranks being at 180°, there is only an unbalanced tertiary couple and reciprocating couple.

If the cranks are at 90°, with either arrangement of cylinders, there is an unbalanced force, secondary in the first case, and tertiary in the second case, and also a reciprocating couple in the second case.

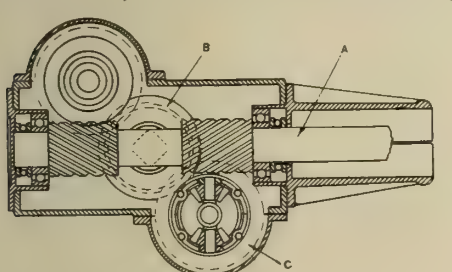
The six-cylinder engine with two cranks at 180°, three connecting rods 120° apart being attached to each crank pin, can be split into two three-cylinder engines 180° out of phase with one another. We saw that in a three-cylinder engine there is an unbalanced tertiary force, so there will be a tertiary force at one crank 180° out of phase with a tertiary force at the other crank; the forces will balance, but, on the other hand, an unbalanced tertiary couple will be set up.

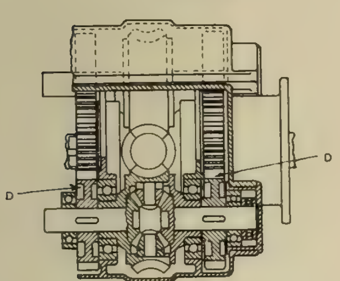
RECENT AUTOMOBILE PATENTS.

By Eric W. Walford, F.C.I.P.A.

Worm Driving Gear.

To obviate the necessity for thrust bearings this inventor duplicates his worm gearing, and provides the driving shaft A with two worm pinions of opposite pitches, each of which drives on to a worm wheel, the two worm wheels being



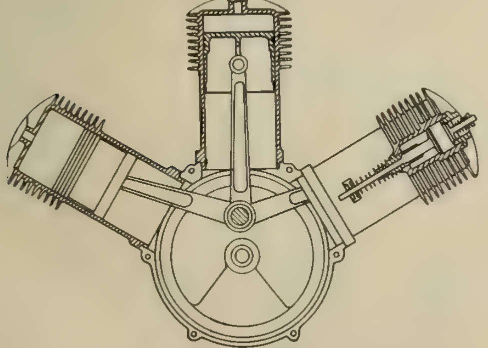


geared by means of spur gearing to a common wheel B on the driven shaft. In the case of a motor car live axle, inside one of the worm wheels, such as C, is arranged the differential gearing, and on each driven shaft of the differential gear is arranged a spur gear D, one of which drives on to each of the live axle shaft lengths. In this manner the end thrust is balanced.

F. Lamplough. No. 23,959/09.

An Aeronautical Engine.

This is the three-cylinder engine used on the Bleriot cross-Channel aeroplane, and the feature of its construction lies

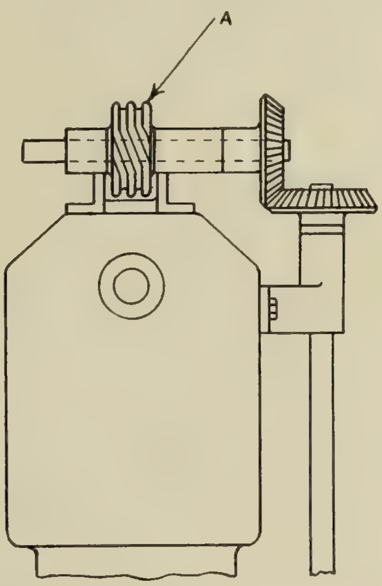


in its compactness. Inside flywheels are used and a single crank pin, upon which one of the connecting rods takes a full bearing, whilst the others have forked bearings upon that of the first connecting rod. The width of the engine is consequently reduced to a minimum.

A. Anzani. No. 28,304/09.

Gearing for Rotary Valves.

This gearing enables a start and stop motion to be transmitted to a rotary valve of any type. The rotary valve spindle is provided with a pin wheel, the pins of which engage a helical cam A, and the pitch of the threads of this alternately

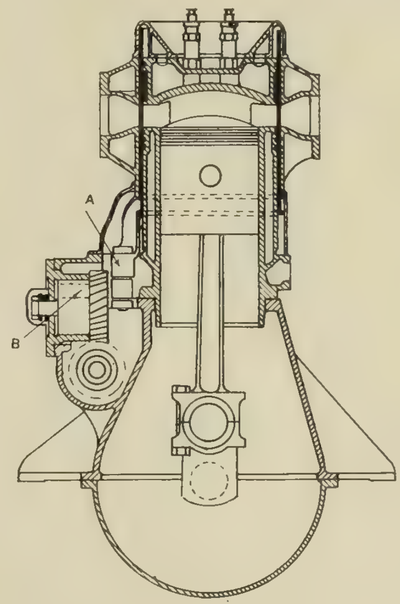


vary between zero and any desired figure. On the zero portion it is obvious that the valve receives no movement.

A. H. Williams. No. 27,618/09.

A Slide Valve Engine.

The patent specification illustrates a number of different methods of carrying out the invention, which consists in giving the sleeve or sleeves a compound movement. Where a single sleeve is used it is given a slight reciprocation, and either a slight angular movement or complete rotation. When two sleeves are employed, one is reciprocated, and the other oscillated or rotated to bring the ports into and out of line at the required period. The drawing shows a single sleeve engine, and the method of operating the sleeve. The sleeve is provided with a lug A, which carries a wrist pin on a bearing, on which is mounted a plunger B, sliding in a driving wheel, the wrist pin and sliding plunger providing the necessary freedom of movement, so that the rotary movement of the driving wheel is translated into the required vertical reciprocation and angular oscillation of the valve sleeve. It is probable that

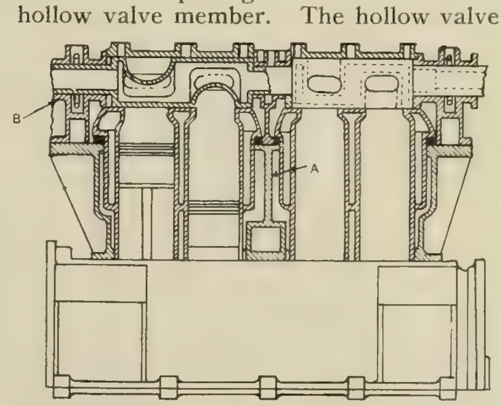


the peculiar movement given the sleeves may have some advantageous effect in the spreading of the lubricating oil.

A. Coats and W. Cameron. No. 16,761/09.

A Rotary Valve Engine.

The cross sectional view shows the particular form of rotary valve used. This has a curved passage A transverse to the hollow valve member. The hollow valve



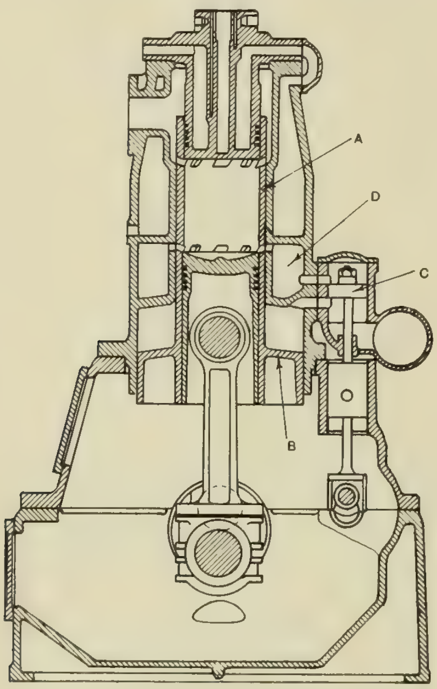
member forms a water pipe return to the radiator, and the valve passage A on rotation is adapted to come opposite the inlet and exhaust passages. The feature of this invention lies in the boldness of its construction. On the crank chamber are mounted brackets A, and these carry the bearings at B for the rotary valve member. From the rotary valve are slung the cylinders which dip into and rest upon trunks, which stand up from the crank chamber. In a modified con-

struction the trunk forms the outer wall of the water jacket, elastic packing rings being interposed between flanges at each end of the water jacket. By slinging the cylinders from the rotary valve there is no tendency for the two to separate, and, as the valve is carried from the crank chamber, the cylinders are subjected to practically no longitudinal stresses.

E. W. Lewis. No. 21,725/09.

A Two-Stroke Engine.

It will be seen that the main piston works inside a sleeve A, the bottom of which is enlarged to form a piston B, the sleeve being reciprocated by eccentric rods driven from the crankshaft. The piston B operates to draw in a charge of air or gas through ports controlled by a

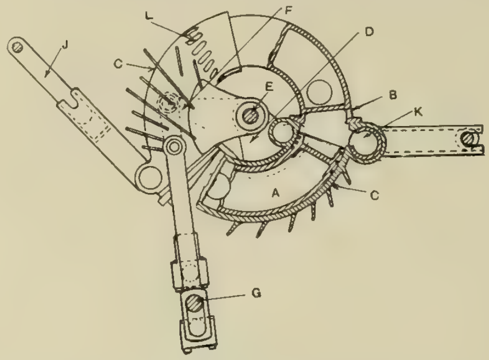


piston valve C, and to compress this into the chamber D. It will be seen that the sleeve A is provided with ports at the top and bottom. When these come into line exhaust takes place through the upper ports, and the fresh charge or scavenging air compressed in the chamber D is admitted to the cylinder, which has the effect of forcing the spent gases out.

Willans and Robinson, Ltd., and J. C. Peache. No. 16,172/09.

An Extraordinary Two-Stroke Engine.

This is probably one of the most extraordinary engines which has been designed during recent years, and it is possible that its operation may be of some interest. There are two reciprocating curved cylinders. The piston A is curved, and its outer end B forms the combustion chamber for the adjacent cylinder C. Each of the cylinders is provided with arms D, which take a bearing upon a stationary shaft E, and the cylinders are adapted to oscillate about this shaft, such oscillating movement being transmitted through arms F, connecting rods, and cranks to a rotating shaft G. To the oscillating cylinders are connected pump cylinders H, the plungers J of which are pivotted as illustrated, these pumps being fuel pumps adapted to fill the combustion chambers through rotating valves K. The cylinder shown in section is deemed to be full of gas, and about to fire. As soon as firing takes place its piston A is forced round, pushing with it the left-hand cylinder, which moves in relation to its piston. Compression consequently takes

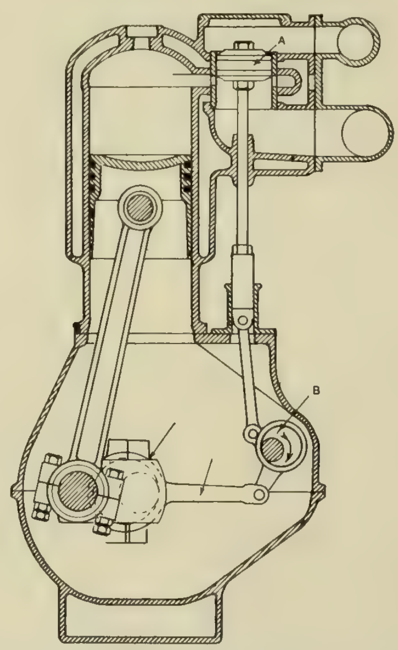


place in this cylinder, whilst exhaust takes place in the right-hand cylinder through the sleeve L. The left-hand cylinder now repeats the cycle in the opposite direction, the cylinders oscillating outside their pistons in a rotary path.

M. Kruse and H. O. Larsen, No. 21,828/09.

Slide Valve Mechanism.

The particular form of valve to which this invention is shown as being applied takes the form of a piston A, which works in a cylinder, in the walls of which are formed a ring of ports communicating with the engine combustion chamber. When the piston A is in the position illustrated, exhaust gas issues from the combustion chamber through the ring of ports, and out below the piston A. When the piston A is in its lower position, inlet takes place, fresh gas passing over the



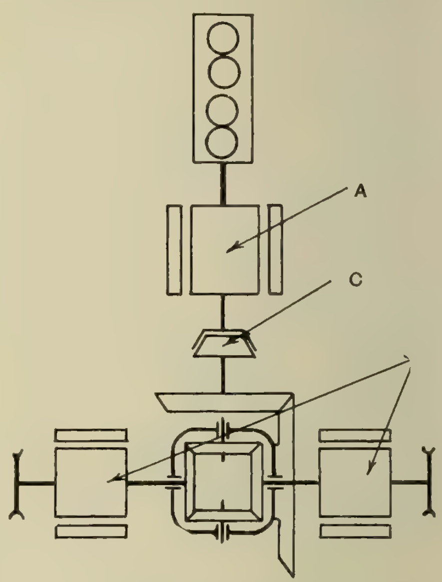
top of the piston valve. The invention deals with the method of operation of the valve piston, and it will be seen to be driven by an eccentric rod coupled to the strap of the eccentric B. The same strap is likewise connected by an eccentric rod to an eccentric on the engine crankshaft. The eccentric rod consequently receives two movements, one due to its own eccentric, and the other due to the eccentric on the crankshaft. At certain times these movements are added together, causing the piston A to be moved very rapidly, whilst at other times they oppose one another, and the piston remains practically stationary. Naturally these operations are arranged to give the quick cut-off and periods of rest required. This seems a clever method of obtaining the advantages of rapid valve action with the two sleeves without complication and without using cams.

A modification is described in which the eccentric strap, instead of being driven from an eccentric on the crankshaft, is rocked from a link attached to the connecting rod.

H. Berry and G. H. Mann. No. 16,420/09.

A Petrol-Electric System.

With this system it is possible to clutch the engine direct to the differential gear and cut out the electric system. The engine drives the dynamo A, which energises the motors, one of which drives

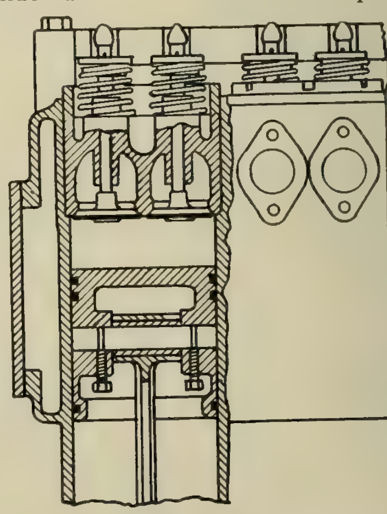


each road wheel. The differential gear in such cases is inoperative, but when it is desired to drive the road wheels direct from the petrol engine, a clutch at C is engaged, so that the road wheels are driven through the differential gear, the motors B having no effect.

Felten and Guilleaume Lahmeyerwerke Ak. Ges. and F. Collischonn. No. 4,681/10.

Cylinder Construction.

The valves are carried in a detachable head, which is slightly larger than the cylinder diameter. The head drops into



place and is secured by a screwed ring. It will be gathered that all the parts are cylindrical, and can be machined all over. Apparently, however, the head cannot be water cooled without employing separate connections.

E. Dombret. No. 6,626/10.

CORRESPONDENCE.

No responsibility is accepted by the Editor for the opinions of correspondents, and the right is reserved to omit any portion of a letter if it is considered desirable to do so. Letters to be inserted with a nom de plume must be accompanied by the name and address of the writer. If publication in the next issue is desired letters should reach the Editor in no case later than the 20th of the preceding month.

THE POSSIBILITIES OF AIR COOLING.

Sir,—In the article under the above title, which appeared in the August issue of *The Automobile Engineer*, Anglo-American does not agree that the circulating fan, forming part of an air-cooled engine system, absorbs a large percentage of the power developed by the engine. When, however, it is remembered that some systems of water-cooling do not involve the use of any mechanical devices requiring the expenditure of power, it is of interest to look more closely into the exact conditions encountered in the air-cooling problem. It must be remembered that in a water-cooled system the jacket water is cooled by air, the radiator being simply a device for providing, in effect, a cylinder cooling surface of sufficiently large area to conduct away the necessary percentage of the total heat generated in the cylinder, the water acting as a good conductor of heat from the cylinder walls to the air-cooled radiator surfaces. If air had the same specific heat and the same heat-conducting properties as water, then it could be used in the cylinder jackets instead of water, circulating at the same low velocity. Such, however, is not the case. Air is a bad conductor of heat; its specific heat at constant pressure is 0.2375 B. Th. units. Thus the problem is simply to circulate sufficient air through suitably-designed jackets or casings, so that the cylinder walls do not attain to a higher temperature than is common to water-cooled cylinders. As an example of an engine to be air-cooled, Anglo-American takes one developing 20 b.h.p., with a heat loss to the jackets of the same amount in equivalent heat units. Now a 20 b.h.p. petrol engine of reasonably high mechanical efficiency, say 85%, will have an indicated h.p. of 23.5, and if its thermal efficiency is as high as 25%, the remaining 75% of the total heat in the fuel, which is heat lost, is divided between the jackets and exhaust gases.

Taking a heat loss to the jackets of 40%, an average value, then 37.6, or practically 38 h.p. = 1,612 B.Th.U's., have to be carried away by the cooling medium.

Assuming the initial temperature of air entering the jackets on a hot day to be 80 degs. F., and the temperature at exit to be 150 degs. F., the rise in temperature will thus be 70 degs. F. Each pound of air will take up $.2375 \times 70 =$ 16.6 heat units, and as the total heat units to be absorbed per minute amounts to 1,612, the weight of air that must necessarily be circulated past the cylinder walls will be $\frac{1612}{16.6} = 97$ pounds. At 80 degs. F. there are 13.6 cu. ft. of air per pound, so the volume of cooling air required per minute will thus be $13.6 \times 97 = 1{,}320$ cu. ft. In order that the circulating air shall take up the necessary amount of heat, the annular jacket space surrounding the cylinder will need to be narrower than when water is used, say, ¼in.

Knowing, then, the quantity of air that must be circulated, and the jacket or annular area surrounding the cylinder, it is easy to calculate the velocity, and from this and the weight of air required, the kinetic energy in the circulating air. Allowances must be made for air friction and eddy losses, and for fan inefficiency, when, even without taking these into account, the h.p. required for driving the cooling fan will be found to be an appreciable percentage of the power developed by the engine.

Using the following data with respect to an engine developing 20 b.h.p. :—

Four cylinder, 3.5in. bore.

Outside diameter of each = 4ins.

Inside diameter of air jacket = 4½ins.

Combined area of annular space surrounding the cylinders = .0928 sq. ft.

Velocity of air through jackets = 237 ft. per sec.

The fan horse-power works out at 2.5, and this neglects the frictional and mechanical losses mentioned above.

One of the leading makers of water-cooled engines at the present time, who were pioneers in air-cooled engines in England, gave up the system because it was found that with even a well-designed air-cooling system, the 12 h.p. engine required 2 h.p. to drive its cooling fans, thus reducing the otherwise useful h.p. by *over* 16%.

JOHN OKILL.

Sir,—The article on "The Possibilities of Air Cooling" in your issue of August forms an interesting discussion on the subject; but some of the writer's conclusions would seem to suggest that he is prejudiced in favour of air cooling; and I here enumerate a few of the points which suggest themselves to me as summing up the situation in regard to the cooling of motor car engines:—An automobile engine must have a certain factor of safety in regard to overheating consequent on being badly driven, through temporary lack of lubrication, defective carburation, or other ingenious abuses, which only the amateur motorist knows how to inflict, and this cannot be provided for to a sufficient extent with an air-cooled engine. To make the statement that air cooling allows of better thermal conditions (and therefore more power economy, etc.) is simply making a virtue of a necessity, as under no circumstances has it the power to reduce the temperature to anywhere near that possible with water cooling, even with enormous expenditure of power in fan driving. Again, the heat efficiency of the modern water-cooled engine could be considerably increased by the abolition of both the fan and pump, but it is not desirable to do this owing to the reasons mentioned above. Apart from other considerations a cooler cylinder induces a greater weight of charge, and the falling off of a power curve at very high speeds is, no doubt, attributable in a great degree to rarification of the charge by the high temperature of the cylinder. High thermal efficiency, therefore, does not necessarily mean increased power, even though it may mean decreased fuel consumption. Tests at the Napier works in 1908 showed that at the rather low jacket temperature of 149 deg., both the greatest horse power and lowest fuel consumption were recorded. To produce even fairly satisfactory results with air cooling means large radiation surface and high velocity of air, we are thus compelled to have fairly heavy cylinders and large horse power absorption by the fan. On the Gnome engine the cooling absorbs about 5 h.p., and the velocity of the air is in the neighbourhood of 80 m.p.h. This gives an idea of what would be required for car work. Air cooling has found its own field, *i.e.*, for motor-cycle and aeroplane work, and it is doubtful if it can ever be applied with complete success to the ordinary heavy passenger automobile.

TOM FORD.

INCHES OR MILLIMETRES.

Sir,—My reasons for suggesting that *The Automobile Engineer* adopt the metric system of weights and measures are that the multiples and sub-multiples being in the decimal system, much time, trouble and energy are saved in reduction, and quantities are expressed as the decimal of a *single* standard (metre), the measurements of area, volume and weight being simply related to those of length :—

10 millimetres^3 = one cubic centimetre = 1 gram.

100 millimetres^3 = 1 kilogram = 1 litre.

1,000 millimetres^3 = one cubic metre = 1 ton.

the weights being that of water at maximum density.

Compare these with the Imperial standard, where 36 inches = 1 yard, 5½ yards = 1 pole, 1^2 pole = 30¼ yards^3.

16 oz. = 1 pound, and 12 of another kind of oz. make another kind of pound, the heavier, ozs. making the lighter pound, and so on, and so on, agreement nowhere to be found.

I am certainly with Messrs. Moore in their desire to have *the best*, but to alter the length of the millimetre, to make 25 exactly equal the inch, would, I am afraid, spoil absolutely the great advantage the metric system has, that is, the simple relation of length, volume and weight.

If Messrs. Moore really want to translate Imperial to metric, why not alter the inch to suit the metre?

I have found that many people will try and translate metric to Imperial and say that 25 millimetres = 1 inch. It surely, Mr. Editor, does no such thing. A centimetre is — so long and an inch —— so long. The one cannot well be translated into the other any more than Latin can be accurately translated into Chinese. To obtain what I shall call the *hang* of the metric system, one must simply *think* in metric.

Mr. Lea and Messrs. Moore champion the cause of the useful 1,000th of an inch. I think, however, that measurements such as 1,000th of an inch and the 10th to 100th of a millimetre are simply relative paper measurements because they are so far removed from the measurements of one's own statue and cannot be fully appreciated as *simple measurements*, but only as relative figures, therefore being only relative figures, surely a portion of a millimetre is in every way as good as a portion of an inch.

The one fault of the metric system is that all variations are by 10ths, but really it is quite probable that this is the fault of mankind in having 10 fingers. If, untold centuries ago, we had been born with 12 fingers instead of the usual 10, possibly mankind would now be reckoning in 12ths, and no doubt the metric system would have varied by 12ths. It really seems such a pity that the young automobile industry does not entirely use metric measures, seeing that tyres, wheels, cylinders, valves, chassis lengths, numerous bolts and aeroplane parts are already expressed in metric, and the full opportunity is there for the grasping.

Concluding, may I ask Messrs. Moore how they arrive at the figure of 80 per cent. of the manufactures of the world being in Imperial inches? I believe I am right in saying that the C.G.S. system of units are used for absolute measures the world over, and that the metric system is compulsory in 20 countries, although in some the system is given another name.

Wishing *The Automobile Engineer* every success.

ARTHUR J. TOBY.

AUTOMATIC ENGINE STARTING.

Sir,—Your contributor, Mr. Bell, in his reply to my letter *re* the above, has apparently misunderstood my remarks.

In the first case I stated that "a dynamo for accumulator charging is always shunt wound"; it may be series wound in addition, but that does not affect the truth of my statements.

In private installations, where the dynamo may be used for lighting direct, in addition to charging, a compound machine is often used, but in the majority of cases the series winding is short-circuited when charging only is in progress.

Where the machine is required to run as a motor in addition to the above, some provision for cutting out one or the other of the windings would be necessary, as if the two were in unison for charging, they would be in opposition for motoring, with detrimental effect to the armature.

The simple solution is a small throw-over switch, so connected to make the machine a shunt wound dynamo, or a series wound motor, as required.

I think your contributor will find his permanent magnet machine for both charging and starting not only very bulky, in comparison with the self-excited article, but far less reliable, owing to the deterioration of the permanent magnets by natural wastage and armature reaction. The effect of the latter on the life of the magnets would be an interesting study, when used frequently for starting an average motor car engine. When used for charging purposes only, at varying speeds, a permanent magnet machine may have advantages in the very small outputs required for motor car battery work, but for all-round reliability what can equal the splendid performances of the "Stone" and similar train-lighting dynamos, of the self-exciting pattern, capable of giving a practically *constant* voltage between train-speeds of 20 and 60 miles per hour?

F. E. SCHOFIELD.

CONTEMPORARY KNOWLEDGE.

Sir,—The article in the September number on the above contains many helpful suggestions towards getting over the difficulty, but I wonder if it has ever occurred to any manufacturers to try the assistance of a working foreman from a repair shop. It must be admitted his experience will be a trifle out of date, but not so much as appears at first sight, e.g., take the case of a man with a sound engineering training in possession of a good ground work in its theoretically side, i.e., mechanics and kindred subjects, coming in contact with, say, 75 per cent. of the different makes of cars on the market, it may be for a few minor adjustments or a complete overhaul. Of course, a car that requires an overhaul has been some time in use, but, in either case, his training has given him an observant eye for all details of construction within range of his job, and he mentally compares the various methods of construction thus brought to his notice.

To take another case:—A car belonging to a customer of his firm is always giving trouble with some part or other, and after putting in the parts, supplied by the makers, with the same result every time, one naturally looks for some inherent fault in the design or construction, and having found it, if he knows the A——— Co. have a better way of doing it, he modifies and adapts the A——— method to suit the circumstances of the case, with usually good results.

I know of several instances of the above, and consider a man of the type in mind to be worth his room in any manufactory.

J.H.C.

Sir,—I should like to say a few words with reference to your recent article on "Contemporary Knowledge," as I consider that the purchase of a batch of competitors' chassis is highly extravagant, especially as every manufacturer has at hand a ready means of obtaining the same knowledge with very little trouble and with no expense.

Full particulars of all new ideas are obtainable from the technical Press and from the various exhibitions as soon as they appear in chassis form—before which it is impossible to buy them. These novelties have generally little value at this stage, and it is only after a prolonged road test that the final opinion can be passed upon them. It is, of course, presumed that the makers have given their "improvements" some sort of test and have eliminated any faults that may have developed before the first chassis is sold.

What is now wanted is precise information as to how the particular features under observation will behave in the hands of the ordinary user as distinct from the engineer. Therefore, from the point of view under discussion, it is correct for the public to buy and test the new features instead of the manufacturer. This is exactly what happens, as it is at this stage that the public do buy and put into service the cars incorporating the new ideas. The only remaining phase is for the manufacturer to obtain the results of these private and unbiassed trials—and to act upon them.

This will present no difficulty if he will only give a little more attention to his repair department, equip it a little better—put it in an entirely separate building if necessary—and lay himself open to *undertake repairs to any and every make of car* instead of limiting himself to cars of his own construction only.

Under present conditions the repair department is often a thorn in the flesh, and many manufacturers do as little as possible of work of this nature; but this should not be so, and would not be so if makers gave a tithe of the attention to this department that they expend on every other. From an educational point of view the repair shop is by far the most valuable part of the factory, and the drawing office should be kept regularly supplied with data from the head of this department.

Who is in a better position to advise as to the strengthening of certain parts, enlarging wearing surfaces, improving accessibility of parts requiring frequent adjustment, etc., etc.? On a host of subjects the observations of the repair man will be invaluable, and if the productions of other firms be included in his work also his views upon improved methods of construction, methods of obtaining economy, silence, flexibility, etc., and even of modifications of design to lower cost of production should be listened to with respect.

For such work a good chief must be appointed—a man who will draw a comparatively high salary: the post demands the highest qualifications, and the ordinary foreman will not do. The man selected must have had a thorough technical training to enable him to accurately determine cause and effect, and to discuss matters freely with the chief draughtsman; he must also have an extensive practical training by means of which he will be able not only to carry out the repairs efficiently, but will be able to appreciate the value of changes introduced to lower manufacturing cost, and, finally, he must be a man of tact. It is highly necessary to eliminate everything that will tend to set one department at enmity with another, and therefore the criticisms of the repair man must be very carefully worded with due regard to the feelings and ideas of the production end of the works.

It is comparatively simple for one man to obtain the knowledge, but it is a totally different matter for this to be tabulated so that the firm can readily make use of it, but with a little thought these matters can all be arranged, and the information thus gained will be invaluable and cannot fail to amply repay the trouble of collecting it.

Make more use of the repair department.

R. PENTONY.

MOTOR CYCLE CONSTRUCTION.

Sir,—In view of the fact that the motor cycle and light "runabout" have, for obvious reasons, possibilities of development immensely greater than has the car proper, and that in their production are really vast potentialities from an industrial point of view, could you not, also, deal with this interesting branch of automobile engineering? At the present time it must be agreed that the design and construction of such machines have by no means reached the standard of excellence seen in the heavier class of automobile. It might almost be said indeed that the motor cycle of to-day stands where the car did ten years ago.

As yet, from some cause or other probably not far to seek, comparatively little *trained* study has been devoted to the motor cycle. The best brains have gone naturally where there is most money. But sooner or later it will pay the engineer equally well, or better, to apply his energies to the improvement to the "motor for the million."

I have thought, therefore, that it might meet the approval of many readers if you would enlarge your province so as to cover the ground indicated. By treating the subject technically in the sound style characteristic of the matter you are giving us now, advancement in the evolution of the motor cycle—two, three, and four-wheeled—would be sensibly expedited.

Perhaps you will allow other readers an opportunity to express their views upon my suggestion. I appeal for support.

D.W.G.

LUBRICATION INDICATORS.

Sir,—In the article on this subject in your September number there are descriptions of several mechanical devices, all, no doubt, more or less effective for the purpose, but for use at night, with the exception of the suggestion contained in your editorial note, they all seem to suffer from the same defect, namely, that they have to be tested by feeling, and as their position must be on the dashboard, this involves a somewhat awkward operation to perform while driving, and one that might quite possibly be the cause of an accident. It is, of course, possible to stop the car to test the lubrication, but this again is inconvenient, and likely to lead to neglect.

Your contributor condemns the ordinary sight drip arrangement, combined with an electric lamp at night, but are there not many good points about it? In the first place, it is much the simplest form of indicator, and therefore the least likely to go wrong. The objection on account of dirty glasses is not a very real one, for they can be kept quite clean by taking them out and washing with paraffin, say, once a month. This is only the work of a few minutes. The electric lamp should be arranged so that the switch is placed close to the driver's right hand, where it can be turned on and off without moving from the seat, or causing any disturbance of coats or rugs.

The presence of an electric lamp is valuable also in case of anything going wrong with the petrol system at night, and it is also useful for seeing other instruments, such as a speedometer or a clock, on the dashboard.

F. H. HUTTON.

HUBS AND ROLLER BEARINGS.

Sir,—In your August number there appeared a letter signed "Rollerace," also another letter in your July issue, signed "Doublerace," calling attention to the lack of durability of multiple row ball bearings. "Rollerace" refers to his disappointing experience with pairs of single row ball bearings placed side by side. He says he is doubtful whether a single row large bearing of equivalent price to a double row, would not outwear the latter considerably. Undoubtedly it would, but he should base his doubling capacity upon dimensions, not upon price. To increase the wear-resisting capacity of a ball bearing, an increase of dimensions is greatly superior to adding another row of balls; for you can then be positive that the single row bearing is taking its full load, but with a double row, one row may still be doing all the work.

"Rollerace" refers to the temptation of restricting the hub bearings to the smallest size, possibly for the sake of a good appearance. This is too true. I know some makers' hubs that should contain bearings twice their present size. Of course, the substitution of a roller bearing with its enormously greater contacting rolling surface in the place of the point to point contact of the ball, would render any increase in present diameters quite unnecessary, and would thus save designers and manufacturers the trouble and cost, in alteration to drawings, patterns, and materials, because the very large margin of excess in load carrying capacity of the roller over the ball would leave a big reserve in favour of the roller bearing when applied to the makers' present standard designs, but "Rollerace" is evidently unaware of the "Timken" taper roller bearing, when he refers to a thrust bearing being required in addition to the roller bearing.

R. F. HALL.

"THE I.A.E."

Sir,—I was very interested to read your remarks in *The Automobile Engineer* with regard to the Graduates' section of the I.A.E. In the Graduates' section now are to be found many talented young men, who not only have a sound practical but also a good theoretical training. These men mostly occupy smaller or intermediate positions in various departments of manufacturing concerns, and are thus in constant touch with first-hand practical experience.

When these men, each having some special knowledge, are brought together at the monthly Graduates' meetings, some most interesting discussions on practical problems result. Young fellows are, however, sometimes reluctant to get on their legs and enter into discussion for the first time, but when once the ice is broken, I have noticed these men often show themselves most fluent on some special branch. They gain confidence as a result of discussion with their fellow workers, and the exchange of ideas results in mutual advantage. The meetings are, as a rule, of a comfortable, free and easy order.

I must say I think that the Graduates' section is a source of large benefit to the young automobile engineer, providing he is within easy reach of some centre.

FRANK H. BALE.

CATALOGUES RECEIVED.

CASE-HARDENING COMPOUND.—A useful pamphlet of hints about case-hardening is issued by Messrs. W. H. Palfreyman and Co., the manufacturers of Palfreyman's hydro-carbonated bone black and of Palfreyman's rust preventive.

VENTILATING FANS, etc.—James Keith, Blackman Co., Ltd., have new catalogues of the belt-driven ventilating fans, their electrically-driven fans, and of the Keith boilers, radiators and their accessories, for heating purposes. Each list is a separate booklet of convenient size and is well arranged.

ENGINES.—The 1911 catalogue of White and Poppe engines is now published and is similar to the previous list as regards style. The dimensions of the engine are given with a sufficient number of measurements to enable frames to be prepared, and the half-tone illustrations are exceptionally good. Types are:—Single-cylinder cycle engine, 85 mm. bore, 85 mm. stroke. One, two, three, four, and six-cylinder engines, 90 mm. bore by 90 mm. stroke. Four-cylinder, 90 mm. bore, 110 mm. stroke. Four-cylinder, 85 mm. bore by 110 mm. stroke. Four-cylinder, 100 mm. bore by 110 mm. stroke. Four-cylinder, 100 mm. bore by 150 mm. stroke. Four-cylinder 110 mm. bore by 130 mm. stroke. And finally, six-cylinder, 140 mm. bore by 150 mm. stroke. The White and Poppe carburettor is described in a separate book, and this contains most careful instructions for fitting and adjustment.

DRILLING MACHINES.—William Asquith, Ltd., who have had a stand at the Brussels Exhibition, have got out a souvenir in the form of a curtailed catalogue printed in English, French, and German, and describing a number of their well-known machines.

SHOCK ABSORBERS.—A booklet descriptive of the Dutrieux shock absorber, with illustrations showing the method of attachment, has reached us from Messrs. Dudgeon and Morren, Ltd.

GAUGES.—The Newall Engineering Co. have recently issued a new list of their well-known gauges, surface plates, and measuring machines. It is profusely illustrated, and the tables of sizes and prices are well arranged, making it easy to find any particular gauge that may be required.

PUBLICATIONS.

AERONAUTICS.—The report of the Advisory Committee for Aeronautics is probably one of the most interesting blue books that has ever been issued to those engaged in the automobile industry. There is no doubt that the investigations of the committee have been conducted in a most practical manner, and though considerable space is devoted to the publication of various theories, particularly in connection with the accumulation of electrostatic charges on balloons, the great bulk of the book will be found extremely valuable to anyone who is embarking on the manufacture of either a gas-supported airship or an aeroplane.

The aeronautical equipment of the National Physical Laboratory at Teddington is first described briefly, and the various main portions of it are well illustrated. Following is a memorandum by Mr. A. Mallock, F.R.S., on some of the questions requiring study, and in this are described experiments made to determine the forces on various surfaces in an air current. The next section is a complete report on the researches made by various other men on the same subject, and the conclusion is that the results of experiments made with a whirling table are unreliable, those made in an air channel appearing to be much more accurate.

Mr. F. W. Lanchester contributes a note on the resistance of planes in normal and tangential presentation, and later on a preliminary report upon the weights and horse powers of various petrol motors for aeronautical purposes. It is pointed out that if the weight of fuel and oil for a long run is included, some of the lightest aero engines make a very poor showing compared to their figures for short runs, which means that a rather heavy economical engine may be better for long distance work than a less efficient engine which is lighter per brake horse power when fuel weight is neglected. The table published, however, is perhaps not quite fair to the lighter forms of engine, because these have so far only been required for short distance work on aeroplanes, and probably their makers have taken no especial pains to make them economical. For instance, the Antoinette is shown to be much less economical than the Daimler engine, but there is no obvious reason why the former should not become as good as the latter, or even better, if as much attention was given to reducing its consumption of fuel and oil, as has already been given to reduction of its weight. The conclusions given are valuable in that they emphasise the importance of economy for all types of flying machine power, but they are misleading if they minimise the importance of light weight of engine.

An interesting section of the book is the summary of the papers relating to the stability of airships and aeroplanes compiled by the secretary to the committee, Mr. F. J. Selby, though he makes no attempt to sum up the conclusions of the different investigators.

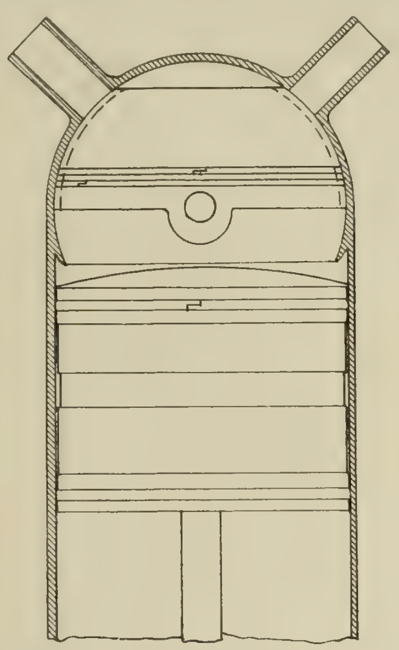
Finally, there is a lengthy consideration of the properties of balloon fabrics as tested at the National Physical Laboratory, and also an account of experiments on plane surfaces and model balloons moving in water. Also many pages are given to observations of wind speeds and variations at a number of meteorological observatories.

It is obvious that much useful work is likely to be done at the National Physical Laboratory during the following year now that the new large air channel and the new motor testing plant are in operation. Thus the next report of the committee will probably be even more instructive than the present volume, though the latter is a work which the earnest student of aeronautical problems could scarcely afford to be without. (Wyman and Sons, Ltd., 8s. 5d.)

POWER AND ITS TRANSMISSION.—All those who have occasion to design the power transmission system in any kind of factory should derive considerable assistance from this small manual by Thomas A. Smith. It is described on the title page as a practical handbook for the factory and works' manager, and this it is undoubtedly. Belt, wire, and cotton-rope transmission, their pulleys and accessories form one section of the work, and there are many practical hints given as to their management, as well as tables of sizes, etc. Another section deals with power installations, such as the gas engine and the electric motor, and this also is full of brief instructions for management. Considerable trouble has obviously been taken to condense the contents as much as possible, and as a result it is but the work of a moment to look through every page for the required heading, and the full index should be needed but seldom. (E. and F. Spon, Ltd. ; 2s.).

AN OSCILLATING VALVE.

The illustration below shows an interesting form of valve for an internal combustion engine patented by W. M. Leedom, of Colton, California. The exhaust and inlet openings are inclined in the cylinder head, and the hollow hemispherical cap oscillates at half piston speed. The design would appear to possess possibilities, as it would not be extremely expensive to make, and there is no obvious reason why it should give trouble, though this could, of course, only be



decided after trial. As shown, it would be impossible to insert the valve in the cylinder, and unless the range of motion was curtailed it would be necessary to split the head of the cylinder in some way, or to re-arrange the ports. The latter would also appear to lend themselves to time adjustments, as it would be possible to alter the nature of the opening by having detachable heads of different patterns, or even detachable port pieces separate from the head.

THE ENGINEERING EXHIBITION.

Quite a large number of the stands at the Engineering Exhibition at Olympia, held during September, were of particular interest to automobile engineers. In the machine tool section there were many exhibits of tools specially adapted for the needs of an automobile factory, and there were also several stands devoted to the display of raw material and finished chassis parts. Amongst the tools, lathes of various types, of course, took a prominent place, some of the principal exhibitors being Alfred Herbert, Ltd., C. W. Burton Griffiths, Ltd., J. H. Storey and Co., the Colchester Lathe Co., Drummond Bros., Ltd., H. Milnes, J. Parkinson and Son, Pfeil and Co., and H. W. Ward and Co., Ltd. Most of these firms also exhibited other tools.

Two of the most interesting machines in the hall were the Sunderland gear generator, shown by Parkinson and Sons, and the Reinecker gear planing machine, shown by Pfeil and Co. The principle of the former is to utilise a rack-form cutter and give a rolling motion to the blank simultaneous with a sliding motion of the cutter, meanwhile the cutter reciprocates across the face of the partly formed tooth. Both sides of the teeth are finished in the one operation with this machine, and it appeared to do very good work. It is comparatively simple and should be easy to handle, while the setting for different pitches is also not a difficult job by any means.

The Reinecker gear planer has been made somewhat on the lines of the already well known bevel gear planer of the same make. The cutter merely reciprocates and operates upon only one tooth face at a time. The blank is mounted on a mandrel borne in a carriage, and it is slowly rotated through a predetermined angle while the carriage is traversed. It will be seen that this compound motion can be adjusted to give almost any pressure angles to the gear teeth, and it is also possible to cut gears with abnormal height of tooth if desired.

There was no great number of grinding machines, though H. W. Ward, Ltd., had a universal cylindrical machine, a universal cutter grinder, and a very well designed bush grinding machine, the latter being undoubtedly the most interesting to our readers. C. W. Burton Griffiths and Co., Ltd., also had a most ingenious completely automatic twist drill grinder.

The increasingly important milling machine was to be found on the stands of many of the exhibitors of lathes who have been mentioned, and Cunliffe and Croom, Ltd., had six of different types, including a useful vertical spindle planer type fitted with a double power gearbox, making it available for either aluminium or other work.

The Britannia Engineering Co., as well as showing a number of small lathes, had several of their marine paraffin engines, including one rated at 60 h.p. The crankcase is cast iron, strengthened by large bolts which are arranged to form columns inside the case, connecting the portions of the casting which carry the cylinders and the crankshaft bearings. This construction permits the walls of the case to be thin, and also allows the use of very large inspection doors, so that once the case is fitted in its bed the whole of the other engine parts can be removed easily without disturbing it. The vaporiser consists of a rectangular chamber, exhaust jacketted, and fitted with baffles which compel all the paraffin to pass over several square feet of heated surface. It is started by means of a blow lamp and no petrol carburettor is supplied. A still larger engine for producer gas is a three-cylinder vertical which was shown by E. S. Hindley and Sons. It is primarily intended for dynamo or pump driving, but is similar in design to some of that company's marine engines.

Yet another marine engine appeared on the stand of W. Beardmore and Co., Ltd. This was a two-cylinder two-cycle vertical, intended for slow speed craft, and adapted for paraffin fuel. On this same stand there were samples of the same firm's pressed steel frames, including one made for the New Arrol-Johnston Co., which had the under-frame and main side members in two single pieces.

Stampings of smaller parts were shown in great profusion by Thos. Smith and Sons, who specialise on all sizes of stamped work. The exhibit included crankshafts, front axles, levers of all kinds, differential spiders, change speed gates, and many other parts. A fair number were shown machined, and can be supplied either thus or in the rough.

Die castings of both a simple and an intricate nature were to be seen on the stand of The Patent Castings Syndicate. While the general nature of this class of work is now too well known to need description, the pieces chosen were excellent for the purpose of demonstrating the possibilities of the process, and certainly suggested that its usefulness is yet to be appreciated at its proper worth.

Acetylene for various purposes was much in evidence. The Imperial Light, Ltd., and C. C. Wakefield and Co. exhibiting flare lamps and house lighting equipment. Messrs. Wakefield had some extremely convenient generators working with the firm's "Carbic," or block-form carbide compound, and the Imperial Light Co. showed an acetylene welding plant in operation, the generator appearing to be of a peculiarly simple and safe pattern, while it was also so arranged that it could be closed up into a comparatively small compass, so enabling it to be transported from place to place easily, and even to be taken through an ordinary doorway.

CATALOGUES RECEIVED.

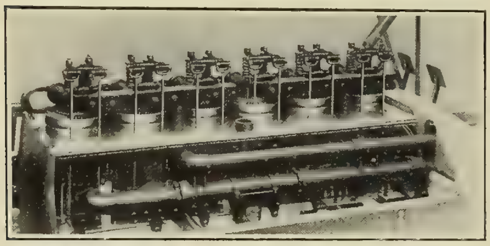
DETACHABLE RIMS.—The instruction book for the manipulation of the Segment detachable and divisible rim is well illustrated and concise in its explanations. It gives a complete description of the device.

IRON ROOFING ETC.—The Bowersfield Steel Co., Ltd., have issued a handy pocket list of their black or galvanized, plain or corrugated steel sheet and the gutters, etc., which come under the heading of roofing accessories. The company are also able to supply weldless steel tubes in a great variety of sizes. This new catalogue is the first issued since the company obtained a gold medal for their exhibit at the Japan-British Exhibition.

THE FRANKLIN AIR COOLING SYSTEM.

Following the article by Anglo-American entitled "The Possibilities of Air Cooling," which appeared in our August issue, we have received a letter from The Franklin Automobile Company describing their well-known system. As this may not be familiar to all our readers, the following passage from the letter may be instructive, as it summarises the claims made, and undoubtedly the Franklin Company have now used air cooling for a sufficient length of time to be able to speak with some authority :—

"The air, instead of being directed on to the front of the first cylinder, and then passing along the sides of each of the others to the rear, is divided into separate currents, one for each cylinder, and each of these is so directed that it constantly envelops the cylinder for which it was



intended, beginning at the top and passing downward along the vertical walls of the cylinder.

"As all of the cylinders have equal currents, and these currents are equally applied to all parts of the cylinder circumference, it follows that all of the cylinders are equally cooled, and that the same temperature is maintained for front, back, and sides of the cylinder. This uniformity tends not only towards greater cooling efficiency, but it serves to prevent such heat-warping of the parts as sometimes destroys their adjustment.

"In the new arrangement the air currents are moved by the flywheel, which is so constructed that it operates as a specially powerful suction fan. There is no fan at the front of the engine. A sheet-metal air jacket encloses the engine, from the top of which extends a funnel of like material encircling each cylinder. These funnels are open at the top, and provide entrance for the cooling air. The suction flywheel creates a semi-vacuum in the jacket enclosing the engine base, and this draws the cool air down along the cylinder walls. In transit the air passes along vertical heat-radiating flanges, which stud the exterior of the cylinder walls. From the point of its admission at the front of the hood the current of each cylinder travels an equal distance before being expelled at the rear of the engine by the suction-fan flywheel. The head is the hottest part of a cylinder, therefore it needs the greatest cooling. In the Franklin system, the air, when it is coolest, immediately after entering, is applied to the cylinder head, and thereafter passes to the cooler parts of the cylinder, which need its effect less.

"The suction-fan flywheel is the only working part of the Franklin cooling system. As an engine must have a flywheel, there is no increase in the necessary number of moving parts, and that means that there is no increase in the number of parts of which there is a reasonable likelihood of getting out of order.

"Franklin air cooling contains for 1911 the chief constructional features which have characterised it in the past. One of these is an auxiliary exhaust. This opens at the base of the explosion chamber immediately after the completion of the power stroke, and through it is discharged 71 per cent. of the hot dead gases, leaving only 29 per cent. to travel up through the cylinder and pass out at the main exhaust. This arrangement gets rid immediately of a large amount of hot burned gases, which if not promptly expelled would tend to heat, warp and pit the exhaust valve, on which compression depends."

We have submitted the letter to the author of the original article, and he replies as follows :—

"It will be seen that the Franklin engine is cooled by passage of air in a direction contrary to its natural flow, and this certainly simplifies the construction, but it must impair the efficiency somewhat at low engine speeds, and it is at low speeds that an air cooling system is always most likely to give trouble. If overheating takes place at all I should expect to find it occur when the engine is revolving slowly with a heavy load, as when running up hill on a high gear, with the throttle wide open, because the flywheel would not then be inducing so strong a draught proportionately as it does at higher speeds, and the air taking longer in its passage through the jacket would become more heated, and would tend all the more strongly to resist movement in a direction contrary to natural convection. No doubt the auxiliary exhaust ports have considerable effect in preventing overheating, but I believe still greater efficiency could be obtained by such an arrangement as I suggested."

INSTITUTE OF AUTOMOBILE ENGINEERS.

Session 1910-11.

Hereunder we give the official list of the papers to be read before the Institute during the approaching winter. Compared with the programme for last session, it would appear that the average importance of the papers will show a decided increase. Putting the Presidential address on one side, the third, fourth, fifth and seventh papers ought to provoke an unusual amount of discussion :—

Presidential Address.

1910.
Oct. 12. "Factors that have Contributed to the Advance of Automobile Engineering, and which Control the Development of the Self-Propelled Vehicle," by F. W. Lanchester, Esq., M.Inst.C.E., President-Elect.

Papers followed by Discussion.

Nov. 9. "Carburettor Action," by Professor W. Morgan, B.Sc., and E. B. Wood, Esq., M.A.

Dec. 14. "The Development of Engines for Marine Purposes," by F. R. S. Bircham, Esq.

1911.
Jan. 11. "The Efficiency of a Two-Cycle Motor," by W. Watson, Esq., D.Sc., F.R.S., and R. W. Fenning, Esq., B.Sc. (Eng.), of the Paddington Technical Institute.

Feb. 8 and 15. Report of the H.P. Formula Committee, with Notes by Dugald Clerk, Esq., F.R.S., M.Inst.C.E., and G. A. Burls, Esq.

Mar. 8. "Airships and the Problems Relating Thereto," by Mervyn O'Gorman, Esq., M.I.Mech.E., M.I.E.E.

Apr. 12. "Effects of Wheels on Roads," by Prof. H. R. A. Mallock, F.R.S.

May 10. "The Use of Pressed Steel in Automobile Construction," by L. A. Legros, Esq., M.Inst.C.E., M.I.Mech.E.

THE WELSH CARBURETTOR.

The Welsh carburettor is an interesting device in that it is different in principle from most other carburettors, as the control of the mixture depends upon the friction between the gas and the pipes through which it passes. In order to understand the action it is necessary to appreciate the construction which is shown in the diagramatic sectional view below.

Starting at the bottom, it will be seen that there is a compound throttle and air valve, which opens a passage to the outer atmosphere, and simultaneously uncovers slots of equal size in the bottom of the float chamber. There is no throttle between the carburettor and the engine. The part seen in the left-hand side of the illustration is secured in the centre of the float chamber, so that air can pass down the two tubes to the slots which communicate with the throttle. Thus, as the throttle is revolved, two separate entrances to the engine are opened, namely, the two tubes and the bottom main air port.

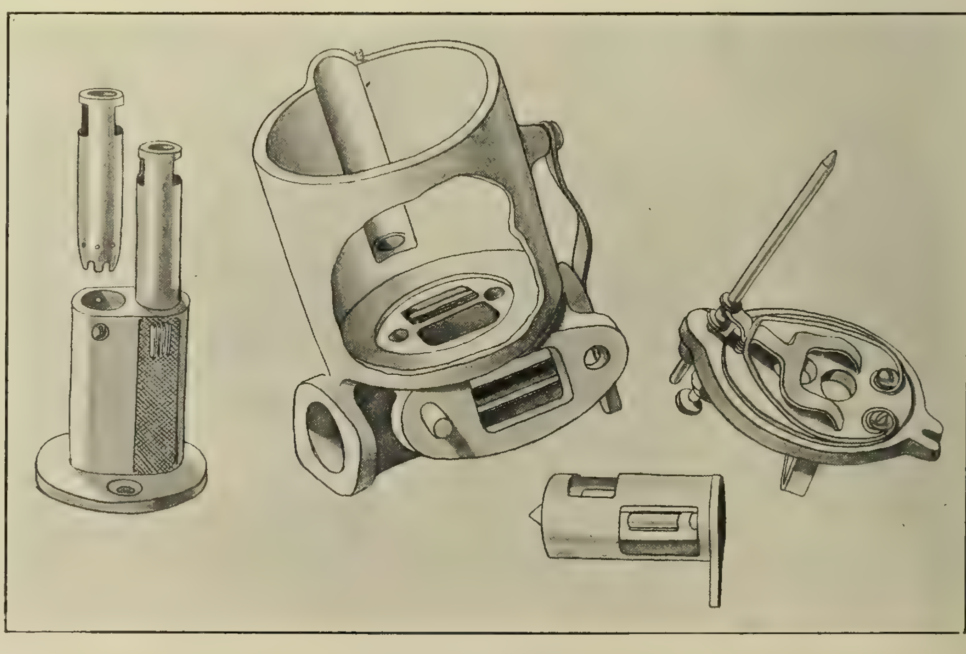
The jets are duplicate, and feed petrol to the central vertical tubes, there being two horizontal holes at the top of the centre piece, which may be made to register with either of six smaller holes in the bottom of the detachable top tubes. The level of the petrol can be adjusted, and should be about equal to that of the jets, while to ensure a supply free from air bubbles there is a vertical channel leading downwards from each jet hole and covered with very fine wire gauze.

It is obvious that the air which is drawn down the vertical tubes will carry with it some petrol sucked from the jets, and that as the throttle is moved the proportion of gas from these tubes to air from the main intake will remain constant unless some means is employed of accelerating or retarding the flow in either branch. Such control is claimed to be supplied by the skin friction between the gas and the small tubes. If a rich mixture is being made, then the friction will be great and the flow slow, while if the mixture weakens the speed increases, so the proportion of petrol in the engine inlet pipe will remain more or less constant. When the throttle is entirely closed a fresh-air port is opened, the gas tubes being cut off completely, and as it is opened, first one gas tube is uncovered, and then both.

It is also claimed that the carburettor is free from overflow troubles when working at small output, as any momentum of the petrol behind the gauze which may remain at the closing of the engine intake stroke causes flow back through the gauze into the float chamber rather than continued flow through the jets.

Likewise there should be no tendency for lag to affect the behaviour of the carburettor, as the jets would supply petrol with very small suction.

Constructionally the carburettor is neat and well made. The float, which is the only part not shown in the illustration, is annular, the adjustment of level is easy, and the jets can be changed very easily and quickly by turning the gas tubes. The main adjustment is the setting of the jets, as the petrol level is supposed to be of use only for changes of fuel density, and it is also stated that once the jets are set for a given carburettor they will probably not require altering even if the engine is changed, providing, of course, that the new engine is within the limits of size for which the carburettor is suited.



The Welsh Carburettor.

THE

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CONTENTS.

ESTIMATING HORSE-POWER.

COMPLAINT of the lack of a satisfactory formula for the estimation of the horse-power of internal combustion engines has been made with such frequency that there is some danger of loss of interest in the matter, on the part of the industry which it concerns most of all. At the present time the incidence of new taxation, on a horse-power basis, has increased the need for a formula very greatly, as previously rating was merely a matter of competition classification, while now it concerns the purse of every owner, and, through him, every manufacturer of taxable vehicles. For the moment a particular equation is in use by the Treasury, but it is obvious that a new method of calculation will be adopted as soon as it is proved beyond all doubt that the present method gives results which average far less than the actual power.

Almost every race, and almost every other form of contest, that has taken place in recent years has been classified in such a way as to encourage high piston speeds and all arrangements whereby the mean average pressure in the cylinders could be increased. It is by no means to be regretted, for, in fact, it is owing to this state of affairs that the weight and consequent cumbersomeness of cars has decreased steadily, while the efficiency, measured in terms of load transported to fuel consumed, has increased very considerably indeed. He would be a bold man who would say that the piston speeds common to-day are a maximum that cannot be exceeded, or even that still greater increase in this direction would be detrimental to the usefulness of the engines for general automobile purposes. However, it cannot be entirely beneficial for there to exist quite as strong an incentive to high stressing of parts as there is at the present time, with both the controlling bodies of competition and the Government using a rating formula in which the cylinder bore is the only variant.

There is no doubt that it will act to the advantage of the industry if the responsible representative bodies can agree upon a better formula than that now employed, so that, when the time arrives, it may be recommended to the Treasury unanimously. It is more than probable that otherwise an even heavier burden may be placed upon the car owner by means of a formula that would over-estimate, rather than under-estimate, the power of an engine.

The Society of Motor Manufacturers and Traders and The Institution of Automobile Engineers have now long been engaged upon a consideration of various formulæ, and the former body, some time ago, published a report which embodied several recommendations. The suggestions were, however, never likely to be adopted, and have doubtless been almost forgotten by now, as the proposed formulæ were somewhat too elaborate for quick reckoning and facile memorising. Few things progress more slowly as a rule than the deliberations of corporate bodies with each other, and, if a new system of power estimation is to be adopted for general use, it is extremely advisable that the pace should be quickened a little, for otherwise the trend of design due to existing conditions will cause many makers to go to extremes which must inevitably be regretted afterwards. The announcement of the engine dimensions of the new cars which will be made during the ensuing year shows how great has been the effect of the method of taxation already, and if piston speeds are to be increased still more, by methods similar to those which have been employed in the majority of instances, then the cost of production cannot fail to be increased; and this is a direct encouragement to the stagnation of improvement of other parts than the engine.

The high piston speeds of modern engines have been obtained in two ways—in fact, piston speed is affected by two things only—and they are by reducing the weight of the reciprocating parts, and by increasing the mean effective pressure. Obviously it is only these two factors which affect the speed of piston travel. Nothing can be said against the first-mentioned procedure, because as long as pistons and connecting rods are strong enough to withstand continued use, then the less their weight the less the power absorbed by overcoming their inertia, and the smaller the vibrations set up thereby, but it seems that the force of explosion can easily reach limits beyond which it is not advisable to go, because, as the mean effective pressure rises, the controllability falls, cooling becomes more troublesome, and the greater heat produced may easily cause lubrication troubles.

Still, apart from considerations of these, perhaps hypothetical, disadvantages, there are mechanical difficulties of more immediate importance. Certainly a few manufacturers have undertaken the task of increasing the power of their engines without an increase of cylinder bore, with a complete lack of appreciation of the nature of their effort, and the fact that some of the oldest and best-known makers have refrained from the production of "high efficiency" engines is doubtless due to their realisation of the magnitude of the work.

The mean pressure of explosion and expansion, of course, depend upon the volume of the charge, and this in turn depends upon the freedom of inlet and outlet for the gases. Freedom of passage depends upon the valve diagram, which is equivalent to the area and form of opening. Thus valves tend to grow in size, by comparison with the diameter of the cylinder, and if of the poppet type, their lift tends to increase also. This means that the valves are becoming heavier, need stronger springs and heavier cams, cam-shafts and driving gears, and as all these features necessitate increase of cost and increased difficulty of silencing, they are to be avoided.

The alternative method of obtaining more power is to prolong the stroke, but this is of little value unless the valve opening is also improved, and it is certainly less easy to make a compact and convenient long stroke engine. Yet a third method is to increase compression pressure, but so far there is no means of using a pressure of over a hundred pounds per square inch, without detrimental effect to the controllability or flexibility.

Each of the means catalogued above have been tried; usually all of them have been tried together, and if effort is to continue in the same direction the effect cannot fail to be bad, especially so for makers, because the difficulties increase very much more rapidly than in direct proportion to the progress already made.

It is for this reason that the adoption of an improved formula for rating, for all purposes, is needed urgently, because though it, at first sight, seems that the use of an under-estimating formula is of advantage to manufacturers, it now cannot be doubted that the price which they will have to pay for it is much more than it is worth. It is not suggested that the Royal Automobile Club formula has yet been the cause of anything other than improvement in engine design, for this is most emphatically not the case, but the limits of the development due to it have equally certainly been almost reached, as will be universally realised before another twelve months has passed.

To be of the greatest possible value and usefulness a formula ought not to encourage development in any particular direction, except improvement in general utility and economy, but such a rating is probably impossible to devise if engine dimensions remain as factors in it. It is also not easy to see how racing is to be conducted except on a basis of power computation, though it might do no harm were the old weight classification once more to be employed; or, possibly even better, a combination of weight and fuel consumption classification. For purposes of taxation, however, there is no reason why power should alone be the deciding factor, especially as the fuel tax penalises the user of a large car to a quite sufficient extent. Still, it is unlikely that the basis of taxation will be altered for some years to come, and it is therefore idle to discuss the possibility at the present time.

If a power formula is to be used it ought to consider both bore and stroke, and it ought to penalise extremes of proportion in either direction. Probably it ought not to encourage a return to the equal bore and stroke proportion, but it ought rather to be equally fair for all engines between a bore-to-stroke or stroke-to-bore ratio of at least one to one and a half, but it appears that it ought to discourage anything beyond this, or at least should not encourage it. To guard against unduly high mean pressure is probably not necessary, because, as has already been pointed out, the flexibility trouble at present limits this, and if this difficulty can be overcome there is no reason why higher pressures should not become common, particularly as they would be likely to improve the mechanical efficiency of engines.

No doubt all these matters have received the attention they deserve at the hands of the committee of investigation, but they may not have been appreciated by the majority of other interested parties, and if a new formula is to be accepted and used quickly, it is essential that it should have the whole-hearted support of manufacturers. It is perhaps unfortunate that it is not possible to evolve a world-wide formula which could be agreed upon as standard in every country where such things are needed, for it would undoubtedly be to the advantage of the whole industry if the same rating was used everywhere. Still, if a sufficiently good suggestion is made, as regards this country, there would be time to consider its international possibilities before its adoption was decided definitely here.

It is a pity that the Royal Automobile Club rating should have been used as widely as it has, because it would have been a great help now if the effect on design of a few other formulæ had been actually tried, and it would certainly be instructive if a few handicap races could be arranged, to be run with the same competitors and as nearly as possible at the same time, using a series of different formulæ for handicapping. It would, of course, be necessary to select a number of cars of widely different types, and they would need to be similar as regards bodywork. Possibly more accurate formula tests could be made with a number of engines alone, on the testing stand, but the former trial would be likely to be more easy to arrange, while cars which were at a disadvantage, through other causes than engine power, should disclose this by taking poor places in the majority of the handicaps. If such a trial could be promoted it would at least arouse public interest, and the results could scarcely fail to be instructive, even if complete reliance could not be placed upon them.

It would be out of place at the present to suggest the whole of the formulæ that ought to be tried, but it may be well to mention a few of them. Foremost of all (on account of its simplicity) is the Dendy-Marshall, $\dfrac{D^2 \; S \; N}{12}$, though it might be found to be unfair to slow-speed engines, still, so far it has been known to give many wonderfully accurate results. It would probably be still more instructive if this formula were to be tried with several different constants. Secondly, the Lanchester formula might be tried in its less complicated form :—

$$1000 \; D^2N \left(\frac{2\;R+1}{R+2}\right),$$

though this even is too complicated for ordinary use, while a simplification of the formula recommended by the Society of Motor Manufacturers and Traders, which might also be experimented with, is : $\dfrac{DN\;(D-1)\times(R+2)}{3}$, where R equals the ratio of stroke-to-bore as in the Lanchester formula, and all other symbols bear their customary significance.

No doubt cars specially prepared for racing would be found to be under-rated by all formulæ, and so would win each race, but the margin by which they did so would indicate the general value of each equation, for it might reasonably be taken that the most accurate formula for all-round purposes would be the one which would produce the closest finishes. Cars with such adventitious aids to speed as special lubrication systems or unusually light chassis should not be included in the selection.

Probably there are many who would consider such a procedure as this suggested trial to be deplorably unscientific, and, while this may be quite true, it is equally an accurate criticism of almost any kind of trial or test that is sufficiently easy to arrange to be practicable. It may be admitted at once that any rating must be inaccurate in the majority of individual cases, but as the new formula is needed for the comparison of cars of different types, there can be few better ways of putting it to the test than by actual experiment on complete cars. The ideal formula is not a formula to give the exact brake-horse-power of any engine to which it is applied; it is a means of classifying *cars* rather than engines alone, and if any formula can be found that is simple, and will enable the order of finishing in a scratch race to be predicted within ten per cent., then it is quite sufficiently good for every-day use.

The practical difficulties in the arrangement of a formula trial are not very great, and the cost would also be small, so small, in fact, that official promoters would probably have no trouble to find a sufficient number of individuals willing to lend assistance in the obvious form, but there is one danger against which precautions would need to be taken, and that is the use of racing machines. It is conceivable that wrong conclusions might be drawn if a large percentage of the participating cars were prepared for high-speed work, not because their engines might be abnormal, but because the remainder of their chassis might be sufficiently different from the standard to give a fictitious value to the engine power as shown by results. That is to say, if a racing car amongst a number of touring cars was found to win every race, whatever the formula, it might not be due even as much to engine differences as to chassis differences. Undoubtedly this fact is the weak feature of such a trial, but if the event was treated as an experimental investigation, and not as a contest—if it could be so arranged that there was no particular honour to be gained by winning each race—the weakness would be of but little account. The governing bodies to whom the conduct of any experiment must fall should have not the smallest difficulty, on an occasion of such great importance, in devising a plan whereby the formulæ would be tested fairly.

THE BALANCING OF AN ENGINE WITH OFFSET CYLINDERS.

By Archibald Sharp, B.Sc., A.M.I.C.E., M.I.A.E., etc.

In the following equations let it be assumed that :—
O^1X^1 be the centre line of the cylinder.
O the centre of the crank shaft.
$r = OP$ = radius of the crank.
$l = PC$ = the length of the connecting rod, centre to centre.
$q = \frac{r}{l}$ = ratio of crank to connecting rod.
a = offset = perpendicular distance of O from O^1X^1.
$e = \frac{a}{l}$ = ratio of offset to connecting rod.
θ = the angle the crank has revolved from the position O X parallel to O^1X^1.
ϕ = the angle connecting rod makes with O^1X^1.
ω = angular speed of crank-shaft.
$\chi = O^1C$ = distance the gudgeon pin C is from O^1, the foot of the perpendicular OO^1.

Draw P A B perpendicular to O X, cutting O^1X^1 and O X respectively at A and B.

Then, $O^1A = OB = r\cos\theta$

$AC = l\cos\phi$

Therefore, $\chi = r\cos\theta + l\cos\phi$ ....(1)

Also, $PA = l\sin\phi$

$PB = r\sin\theta$

Therefore, $l\sin\phi = r\sin\theta - a$

or, $\sin\phi = q\sin\theta - e$

and, $\cos\phi = \sqrt{1-(q\sin\theta - e)^2}$ .... (2)

Expanding the expression on the right hand side of (2) by the Binomial Theorem, and putting $(q\sin\theta - e) = k$, we get :

$$\cos\phi = 1 - \tfrac{1}{2}k^2 - \tfrac{1}{8}k^4 - \tfrac{1}{16}k^6 - \ldots\ldots\ldots(3)$$

Substituting the values of k^2, k^4, k^6,.........we get

$$\begin{aligned}\cos\phi = &\left(1 - \tfrac{1}{2}e^2 - \tfrac{1}{8}e^4 - \tfrac{1}{16}e^6 - \ldots\right)\\ &+ eq\sin\theta\left(1 + \tfrac{1}{2}e^2 + \tfrac{3}{8}e^4 + \ldots\right)\\ &- q^2\sin^2\theta\left(\tfrac{1}{2} + \tfrac{3}{4}e^2 + \tfrac{15}{16}e^4 + \ldots\right)\\ &+ eq^3\sin^3\theta\left(\tfrac{1}{2} + \tfrac{5}{4}e^2 + \ldots\right)\\ &- q^4\sin^4\theta\left(\tfrac{1}{8} + \tfrac{15}{16}e^2 + \ldots\right)\\ &+ eq^5\sin^5\theta\left(\tfrac{3}{8} + \ldots\right)\\ &- q^6\sin^6\theta\left(\tfrac{1}{16} + \ldots\right)\\ &+ \ldots\ldots\ldots\ldots(4)\end{aligned}$$

The even powers of $\sin\theta$ in (4) have the same values for $\theta = \theta_1$

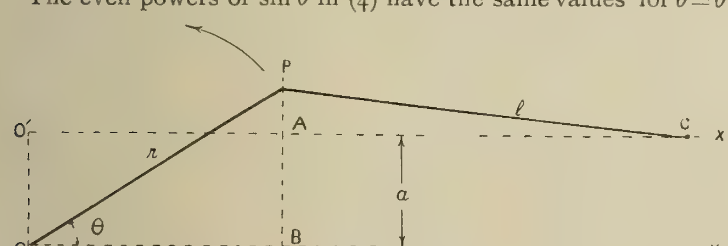


Fig. I.

and $\theta = -\theta_1$, and they can be expressed in terms of cosines of multiples of 2θ. (See Todhunter's Trigonometry).

$$\left.\begin{aligned}\sin^2\theta &= \tfrac{1}{2} - \tfrac{1}{2}\cos 2\theta\\ \sin^4\theta &= \tfrac{3}{8} - \tfrac{1}{2}\cos 2\theta + \tfrac{1}{8}\cos 4\theta\\ \sin^6\theta &= \tfrac{5}{16} - \tfrac{15}{32}\cos 2\theta + \tfrac{3}{16}\cos 4\theta - \tfrac{1}{32}\cos 6\theta\end{aligned}\right\}\ldots(5)$$

The odd powers of $\sin\theta$ in (4) change in algebraic sign as θ changes from the value θ_1 to the value $-\theta_1$, and they can be expressed in terms of sines of odd multiples of θ.

$$\sin^3\theta = \tfrac{3}{4}\sin\theta - \tfrac{1}{4}\sin 3\theta$$
$$\sin^5\theta = \tfrac{15}{32}\sin\theta - \tfrac{5}{16}\sin 3\theta + \tfrac{1}{16}\sin 5\theta$$

Substituting in (4), it may be written

$$\begin{aligned}\cos\phi = &A_0 + A_2\cos 2\theta + A_4\cos 4\theta + \ldots\\ &+ A_1\sin\theta + A_3\sin 3\theta + A_5\sin 5\theta + \ldots\ldots(6)\end{aligned}$$

α, the acceleration of the gudgeon C pin is equal to $\frac{d^2\chi}{dt^2}$, therefore differentiating twice the expression (1),

$$\chi = r\cos\theta + l\left\{\begin{aligned}&A_0 + A_2\cos 2\theta + A_4\cos 4\theta + \ldots\\ &+ A_1\sin\theta + A_3\sin 3\theta + A_5\sin 5\theta + \ldots\end{aligned}\right\}$$

and remembering that $l = r/q$, we get

$$\alpha = -\omega^2 r\left\{\begin{aligned}&\cos\theta + B_2\cos 2\theta + B_4\cos 4\theta + \ldots\\ &+ B_1\sin\theta + B_3\sin 3\theta + B_5\sin 5\theta + \ldots\end{aligned}\right\}\ldots(7)$$

where $B_1 = A_1/q$, $B_2 = 4A_2/q$, $B_3 = 9A_3/q$,.............. the coefficients B_1, B_2, B_3,......involving q and e in their expressions.

The value of e being small in comparison with unity, e^2 and higher powers of e may sometimes be neglected in comparison with unity in the expressions inside the various brackets in (4). Equation (7) may then be written, approximately :

$$\alpha = -\omega^2 r\left\{\begin{aligned}&\cos\theta + q\cos 2\theta - \tfrac{1}{4}q^3\cos 4\theta + \tfrac{9}{128}q^5\cos 6\theta - \ldots\\ &+ eq\left[\sin\theta - \tfrac{9}{8}q^2\sin 3\theta + \tfrac{75}{128}q^4\sin 5\theta \ldots\ldots\right]\end{aligned}\right\}\ldots(8)$$

If both sides of (8) be multiplied by m/g, where m is the mass in lbs. of the reciprocating parts (piston + proportionate part

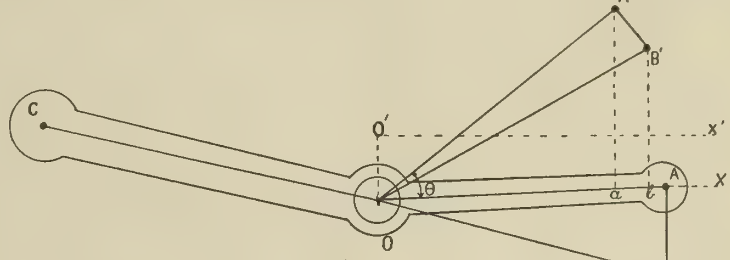


Fig. II.

of connecting rod), the left-hand side is the force required to accelerate the reciprocating parts at the instant the crank is at the angle θ with the centre line of the cylinder. The right-hand side of (8) shows that this unbalanced force is made up of a number of forces of 1st, 2nd, 3rd, 4th, ... orders, which may be considered as due to unbalanced masses revolving at 1, 2, 3, 4 . . . . times the speed of the crank. If the offset a is zero, e is zero, the terms of (8) involving sines of odd multiples of θ vanish, and (8) then becomes the known expression applicable to an engine with its cylinders arranged in such a manner that their axes pass through the axis of the crankshaft.

In an offset single cylinder engine the primary force of acceleration is, from (8),

$$\frac{mr\omega^2}{g}(\cos\theta + eq\sin\theta)\ \ldots\ldots\ldots\ldots\ldots\ldots(9)$$

In Fig. II. let OA represent the crank radius r, when parallel to the axis O^1X^1 of the cylinder. Set off A B=eqr, in the direction opposite to the cylinder axis. Then the primary force of acceleration of C along O^1X^1 is the same as that of a mass m moving with simple harmonic motion along O X, and being driven by the imaginary crank O B rotating with, and at the same speed as, the actual crank O A. For, drawing the crank in position OA^1 at any angle θ with O A, and the corresponding position OB^1 of the imaginary crank, and drawing A^1a and B^1b perpendicular to O X, we have,

$Oa = r\cos\theta$,

$ab = eqr\sin\theta$,

and therefore, $Ob = r(\cos\theta + eq\sin\theta)$, which multiplied by $-m\omega^2/g$, agrees with (9).

The said simple harmonic motion along O X of the mass m is equivalent to a forward rotary motion of a mass m/2 fixed to the imaginary crank pin B, and a reverse rotary motion of an equal mass m/2 fixed to an imaginary crank pin revolving in the reverse direction, the two imaginary crank pins coinciding in position on the line O X. The accelerating force for the former mass m/2 can be balanced by fixing a counter-balance mass m/2 to the actual crank in the position C, in B O produced, O C being equal to B O. This is the best possible balance for a single cylinder engine, the remaining unbalanced force being due to the reverse rotating mass m/2.

The simple harmonic motion along O X equivalent to the actual counterbalance mass, and the imaginary reverse rotating mass do not completely balance the simple harmonic motion of the actual parts which reciprocate along $O^1 X^1$. There remains a disturbing couple, transverse to the crank shaft equal to the primary force of acceleration multiplied by the offset a; that is, equal to $\frac{mra\,\omega^2}{g}(\cos\theta + eq\sin\theta)$. This disturbing couple is of the same nature as that due to the angular swing of the connecting rod, of which the primary couple is approximately $\frac{I\,q\,\omega^2\sin\theta}{g}$, where I is the moment of inertia of the connecting rod. The disturbing effect of these transverse couples is small compared with that of the primary unbalanced force.

Motor-Bicycle Engine with Offset Cylinder.

It will be seen from the above discussion that the primary unbalanced force in a single cylinder engine with offset cylinder is slightly greater than that in a similar engine with central cylinder, in the ratio O B/O A. But this slight increase is so small that it cannot appreciably affect the balance. As an example, take the connecting rod length equal to $2\frac{1}{2}$ times the stroke; that is, $q=1/5$, and the offset a equal to r/2, i.e., $e=1/10$. Then $AB=eqr=r/50$, and $OB=1.0002r$. That is, the increase in the primary unbalanced force due to the offset is about two parts in 10,000. Therefore, no engine builder need hesitate to adopt offset cylinders from the fear that the balance of the engine may be impaired.

Four-Cylinder Engine with Offset Cylinders.

In a four-cylinder car engine with offset cylinders, the four imaginary cranks O B (Fig. II.) are arranged in opposite pairs, and the primary balance is perfect; the terms eq sin θ in the expression (8) for the four cylinders mutually cancelling each other. The same remarks apply to the terms of the 3rd, 5th, ... orders. Thus the largest unbalanced force is of the 2nd order, and equal to $\frac{-4\,\omega^2\,mr\,B_2\cos 2\theta}{q}$, exactly the same expression as for that of the 4-cylinder engine with central cylinders. But it should be remembered that the coefficient B_2 is slightly different for central and offset cylinders, the value $B_2=q$, given in the expression (8) being only a first approximation. To complete the comparison between the two types, the exact values of B_2 for each should be calculated. The necessary data has been given above, and in the following paragraph the exact values of B_2, B_4, . . . are determined.

The primary transverse couples from the four cylinders balance each other. The largest unbalanced transverse couple, of the second order, is equal to the secondary unbalanced force multiplied by the offset a, the disturbing effect of which is relatively small.

Exact values of Co-efficient B_2, B_4, ...

Substituting in equation (4) the values of sin ${}^2\theta$, sin ${}^4\theta$ ...... given in (5) we get:

$$\left.\begin{aligned} B_2 &= q\left(1+\tfrac{3}{2}e^2+\tfrac{15}{8}e^4+\ldots\right)+q^3\left(\tfrac{1}{4}+\tfrac{15}{8}e^2+\ldots\right) \\ &\quad +q^5\left(\tfrac{15}{128}+\ldots\right)+\ldots\ldots \\ B_4 &= -q^3\left(\tfrac{1}{4}+\tfrac{15}{8}e^2+\ldots\right)-q^5\left(\tfrac{3}{16}+\ldots\right)+\ldots\ldots \\ B_6 &= q^5\left(\tfrac{9}{128}+\ldots\right)+\ldots\ldots \end{aligned}\right\}\quad\ldots(10)$$

The expressions in (10) are carried far enough to enable the calculations of B_2, B_4, . . . to be made correct to 4 or 5 places of decimals. In the following table the values are given for engines with connecting rod length 4 and 5 times the crank radius, in each case with a central cylinder, and with offset cylinder, the offset being half the crank radius:—

From the values B_2 in the table it is seen that with connecting rod equal to five times the crank length, and offset equal to half the crank radius, the secondary unbalanced force of a four-cylinder car engine is about $1\frac{1}{2}$ per cent. greater than with central cylinders—an increase small enough to be practically negligible.

	$q=\frac{1}{4}$		$q=\frac{1}{5}$	
	$e=0$	$e=\frac{1}{8}$	$e=0$	$e=\frac{1}{10}$
B_2	.2540	.2604	.2020	.2052
B_4	—.0041	—.0045	—.0020	—.0022
B_6	.0001	.0001	.0000	.0000

Balancing of Sliding Valves.

The results of the above investigation apply directly to the balancing of the reciprocating sleeves as used in the Knight engine. The investigation shows that even with short connecting rods and considerable offset, the primary balance is perfect, and that the secondary unbalanced force is not appreciably increased by the offset. The reciprocating sleeves do not therefore disturb the balance of the engine. I believe it has been claimed by some advocates of the sliding valve that the sliding sleeves *improve* the balance of the engine. But this is not the case. The secondary unbalanced force of the sleeves may be considered as due to an imaginary mass moving with simple harmonic motion and being driven by an imaginary crank rotating at twice the speed of the eccentric shaft; that is at half the speed of the imaginary secondary unbalanced mass of the engine pistons. Whatever be the relative magnitudes of the two imaginary masses, since they produce disturbances having periodicities, one twice the other, they cannot be said to neutralise each other. If the eccentric shaft revolved at the same speed, and in the same direction as the crankshaft, and the eccentrics were set at right-angles to the cranks, then, and then only, by properly arranging the relative masses of the pistons and sliding valves, the secondary unbalanced forces of the systems might be made to cancel each other.

THE 40 H.P. NEW ENGINE CHASSIS.

As preface to the description which follows, it ought, perhaps, to be explained that the N.E.C. chassis has been designed almost entirely from the point of view of the user who desires to have a roomy and rather heavy body, the greatest possible comfort, and sufficient power to maintain a fair speed under adverse road circumstances. In fact, the chassis has been made to suit the bodywork rather than as a mechanism to carry any type of superstructure. Undoubtedly, from the restricted standpoint we have mentioned, the chassis has a number of special claims to consideration, as the large range of relative motion between the frame and the axles, made possible by the spring arrangements, renders the car much above the average of comfort for its type on rough roads. The disposition of the engine allows enormous leg room in the back of the body, without an abnormal wheelbase, and the slow-speed low-compression engine gives extremely smooth running over a large range of speed. The accessibility is generally not by any means so bad as might be anticipated, though, of course, it is not good in view of the fact that everything lies beneath footboards.

Reference to Fig. I. shows the general arrangement of all the parts. It will be observed that the carburettor, the oil filter, and the magneto are all to be got at from the forward end, that is to say, by the removal of the driver's footboards, or the detachable side portions of the front seat. The whole of the transmission is, on the other hand, accessible only through the floor of the rear part of the body, wherefore considerable care must be necessary to protect the upholstery when greasing operations are in progress. An under-frame is employed for the gear box, but the engine is secured to cross members of the main frame, and it is therefore to be supposed that the alignment may cause some trouble in erection. Each of the four springs is clamped securely to a socket bolted to the frame, this being shown inset in Fig. I. Radius rods of a deep and narrow section are situated beneath each spring, and are slung from bolts held in the spring-end boxes, the purpose of the peculiar section of these radius members being to reduce the possibility of a severe wrenching stress, applied by a side-slip or a glancing collision with a curb, causing their fracture, as they will bend rather than break.

The petrol and oil tanks, which are above and behind the radiator, may be regarded

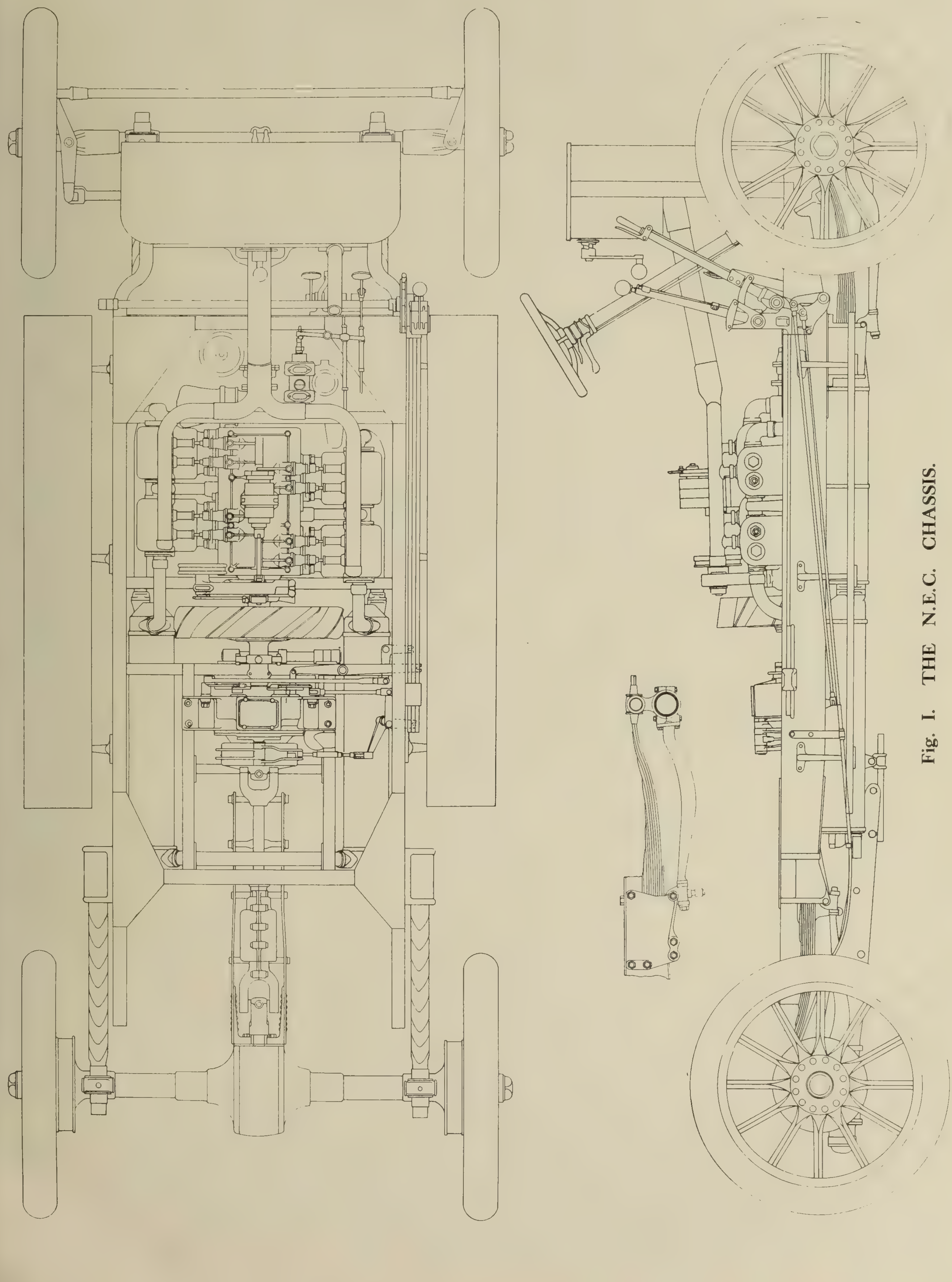
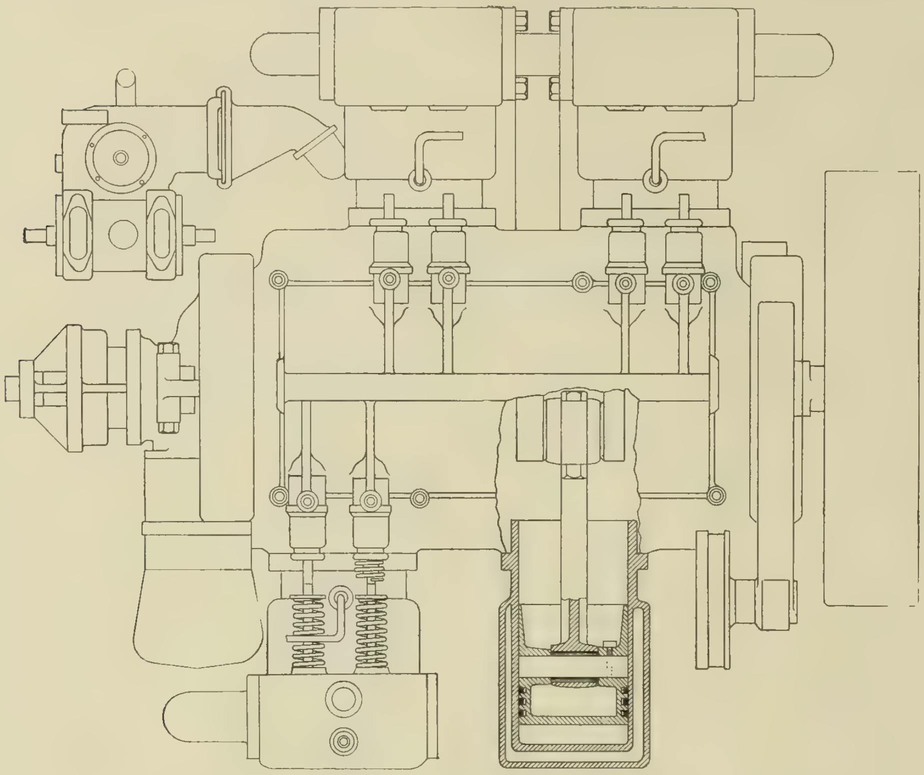
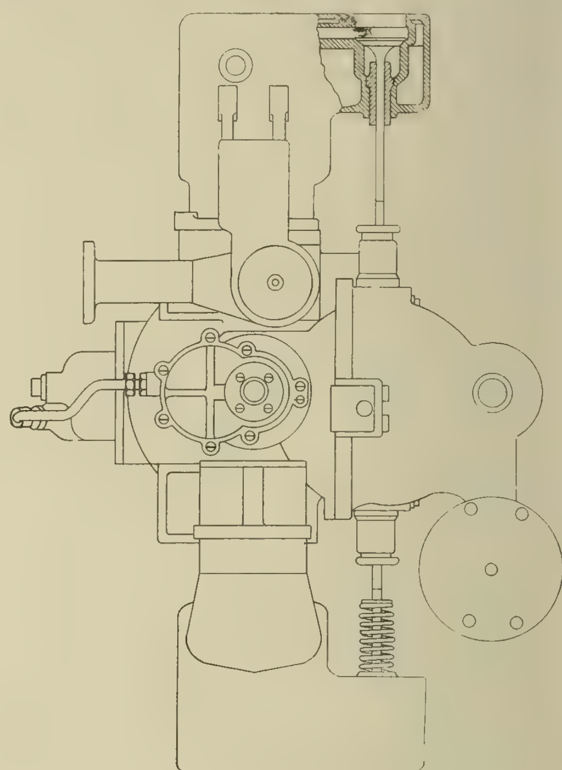


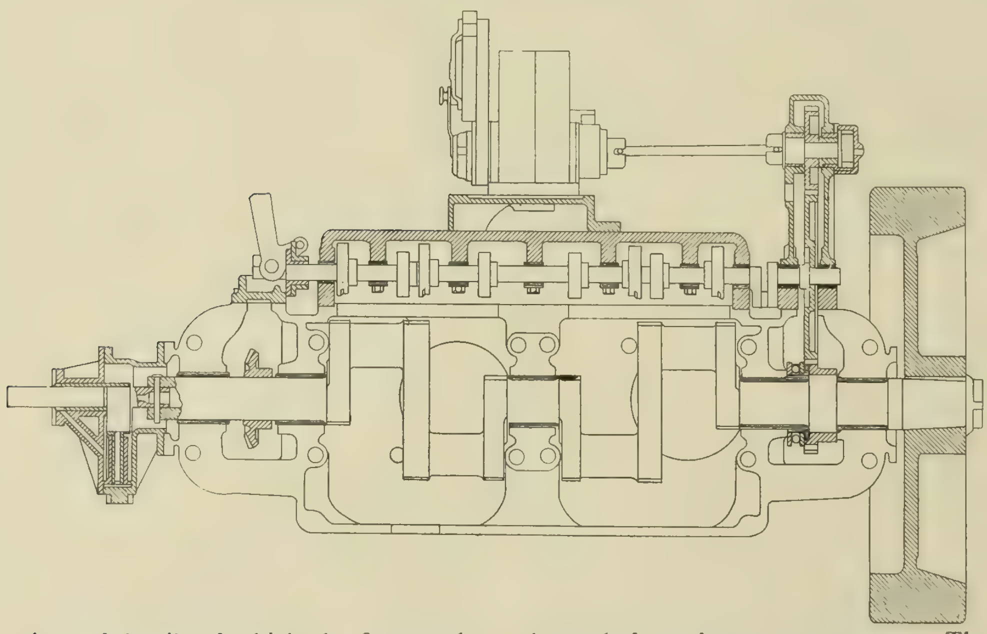
Fig. I. THE N.E.C. CHASSIS.

as a part of the frame unit, as they serve to carry the starting handle and the dashboard, so the frame as a whole is distinctly elaborate, nor do the attachments lend themselves to the most simple fitting. Still, there can be no doubt as to the strength, while the frame should also possess great rigidity.

The engine is almost fully explained by Fig. II., but has a variety of interesting







points of detail, of which the first is the lubricating system. There is a rotary pump driven off an eccentric on the forward end of the crankshaft, which draws its supply both from the bottom of the crank case and from the oil tank on the dashboard, which is supplied with a drip feed to maintain the necessary quantity of oil in the circulation system, without the need for occasional filling up of the crank case direct. The pump forces oil through a hollow shaft to each main bearing, whence it, of course, falls to the bottom of the crank case, but only a quite small quantity remains in this position, the major portion of the lubricant in circulation passing to the filter, which contains almost as much as a sump of average dimensions. This filter appears in Fig. III., the gauzes are separated by brass discs, and the outlet is, of course, at the bottom. Thus there is a very large area of gauze, and so a free supply to the pump, while the effect of deposits of dirt will be slow, as the choking of one or two of the lower gauzes would not restrict the flow perceptibly, and the lower gauzes will, of course, choke long before the upper gauzes. There is a draw-off plug, enabling the system to be emptied or the filter to be cleaned out, but the gauzes can also be removed from above without causing any loss of oil. There is a second pump driven off the camshaft, which also sucks oil from the filter, and forces it to drips on the dashboard, and from these there are leads which drop oil on each big end, and also supply each piston with a very small continuous feed. This second pump is of the ordinary gear pattern, but the main pump is a quite unusual design, and is shown in Fig. IV., the eccentric being fixed to the crankshaft. Aluminium is used for the stationary case, and as the eccentric revolves it forces the plunger to reciprocate. The eccentric is hollow, and there is a slot in the periphery, which comes opposite the end of the hollow plunger when the latter is on the compression stroke. At the same time the slot is also uncovering the outlet from the chamber to the crankshaft, and so a charge is forced to the bearings at each crankshaft revolution.

Another important part of the engine is the governor which controls the carburettor and the ignition timing. Fig. V. shows both the latter and the governor itself. The drive is from the crankshaft through a pair of bevel wheels, and the speed of revolution is twice that of the crankshaft. The control of the firing, as applied to the high-tension battery ignition, is performed by causing the governor arms to traverse a very wide fibre cylinder bearing the contact piece; the latter is V-shaped, and is set as shown in the illustration, so the advance is automatic with the speed, while the magneto is automatically controlled by moving the make and break in the usual manner, but by the governor, and not by hand. The brush can be moved for starting purposes only, and it is connected to a half compression device, so that the engine can scarcely be turned except with both ignitions in a perfectly safe position. At the bottom of the crank case in Fig. V. another control rod will be seen, also worked from the governor, and this is connected to the main air intake of the carburettor, causing a piston to increase the area as the speed rises. Thus the velocity of the air entering the carburettor is constant at all engine speeds, within the limits of accuracy of the governor, which is certainly of assistance in maintaining a regular mixture.

The carburettor is shown in Fig. VI. The two pistons are rigidly connected, and can be moved longitudinally by hand, thereby opening or closing the air supply and the throttle simultaneously, while the governor mechanism causes rotation of the pistons. The throttle position is unaffected by the rotation, but the air piston

is so shaped that the rotation causes variation of the size of the passage quite independently of the hand control. Thus both actions together affect the volume of entering air, and will act similarly or

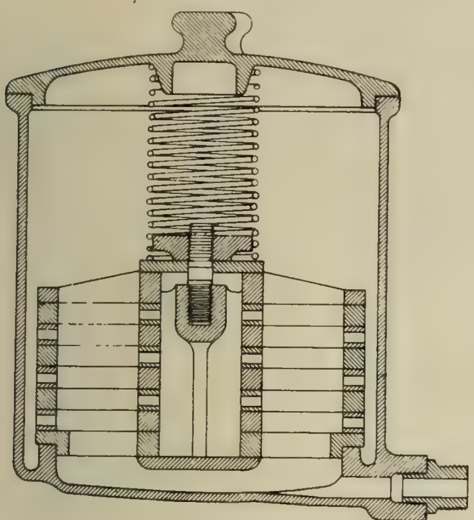


Fig. III.

otherwise, according to whether the throttle is opened as speed increases or as speed decreases, and *vice versa*. There is an air admission grid in the air pipe which is connected to the half compression lever, and is closed for starting purposes.

Another small but ingenious feature is the means whereby the top of the crankcase can be attached or detached without trouble. Owing to the valve arrangement and the horizontal placing of the cylinders, obviously the valve springs will grip the top piece of the case, as the latter carries the tappets, and so either the valve springs must be held in compression or the tappets removed, previous to detachment of the whole piece. To ob-

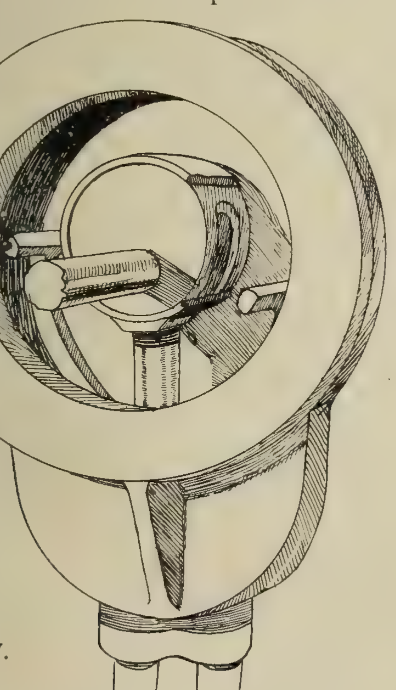


Fig. IV.

viate this difficulty the tappet cases are constructed in such a manner that the actual striker can be allowed to move sideways out of engagement with the end of the valve stem. As this is but the work of a moment, and as there is an easily detachable coupling between the camshaft and its driving wheel, both camshaft and crankshaft can be inspected in their entirety in a very few minutes.

As regards the transmission, the clutch is a leather-faced one, and can be detached without disturbing either the engine or the gear box. It possesses no special features of interest, but is not supplied with a universal coupling to the gear box.

The gear box is one of the most peculiar parts of the whole chassis. The reason for its unusual arrangement being to enable the use of very short and therefore stiff shafts, and also to dispense with the sliding change. Fig. VII. is a diagrammatic view of the gears and shafts, and Fig. VIII. shows the operating mechanism, which can also be observed in the chassis plan. To follow the scheme of operation it should first be observed that the central shaft is divided and spigoted, the front half carrying two gears, which are thus driven by the crankshaft (and will be called the driving gears). The rear half of the middle shaft carries two driven gears, which transmit motion to the propellor shaft. For the fourth, and highest, speed the foremost of the driven gears is slid forward and engaged with an internally-toothed ring cut inside the rearmost of the driving gears,

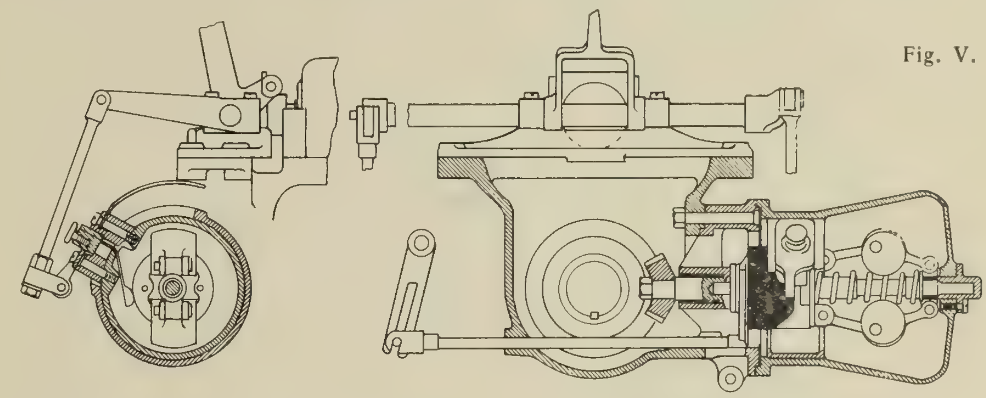


Fig. V.

and there are then no other gears in motion. All the other ratios are obtained by a mechanism exactly similar to the back gear of a lathe. Each of the four subsidiary shafts, which lie round the main shaft, is mounted on bearings set eccentrically in their housings, and each carries a pair of gears secured to it. Thus by turning the bearings the larger layshaft gear is brought into face engagement with one of the driving gears, and the smaller layshaft gear into engagement with one of the driven gears. As there are two driving gears and two driven gears, there are four possible variations, that is to say, if the four be called A, B, a, b, connection may be made between A and a, A and b, B and a, or B and b. In practice B and b give the lowest speed, A and b the second speed, A and a the third speed, and the last combination is not used as a forward speed (the capital letters representing the driving, and the small letters the driven gears). For the reverse, motion passes from B to b through the reverse train of three gears, and the method of supporting the extra pinion as well as the means for engaging it are shown in the end view of Fig. VII. Thus there are thirteen gears in all. The advantages are the stiffness of the shafts and the fact that no idle gears are in operation on any speed; also it appears that there is a much smaller liability for the gears to become damaged by careless changing than is the case with sliding change speed. On the other hand, the price seems a somewhat high one to pay, for the disadvantages are, great cost of production owing to the number of gears to be cut, the number of shafts, and the complicated operating mechanism, while the transmission is not noticeably a quiet one, although it is cer-

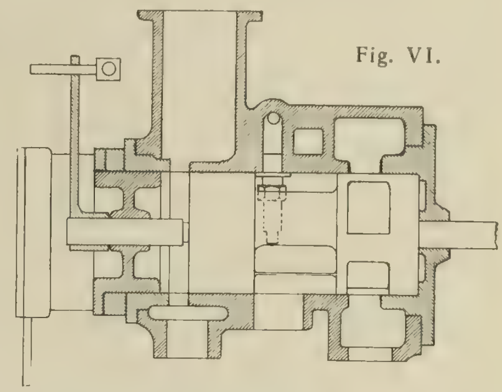


Fig. VI.

tainly by no means noisy. In any case, the increased silence is not a proportionate return for the great amount of labour. The gear box ratios are normally in proportion of two, three and four to one, and including the axle reduction this gives engine to road wheel ratios of 15.5 to 1, 7.7 to 1, 5.2 to 1, and 3.87 to 1.

The method by means of which the eccentric motion is conveyed from the change-speed lever to the gear shafts can be followed by reference to Fig. VIII. The gate is precisely similar to that used for a sliding gear, and the striking rods lie along the chassis frame, as shown in

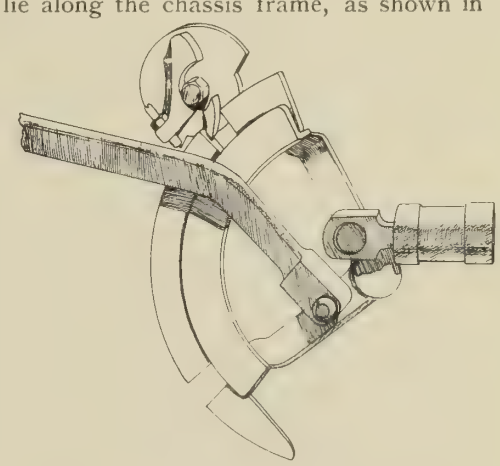
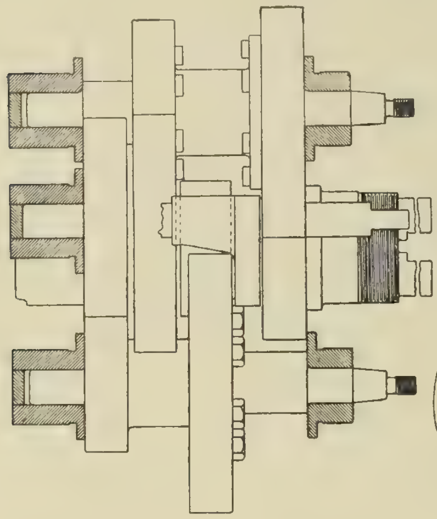


Fig. VIII.

the plan of the chassis. From their rear ends links connect to the actual strikers shown in Fig. VIII. The figure, of course, shows only one of the eccentric shaft ends, but each quadrant controls two of them, the fourth speed being operated by a fifth and separate direct slide, which is connected to a separate rod from the gate lever. Normally the projecting pawls of the shaft pieces lie against the periphery of their respective quadrants, and it is then impossible for the gears to

come into mesh. Movement of the quadrants in either direction causes a slot to pick up one of the pawls, and further motion then rotates the eccentric until the



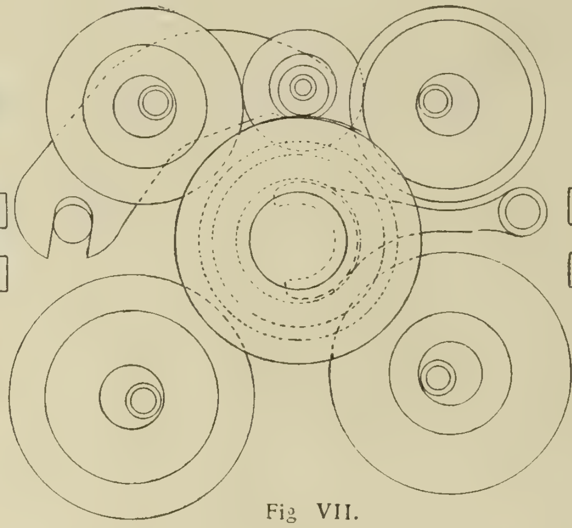
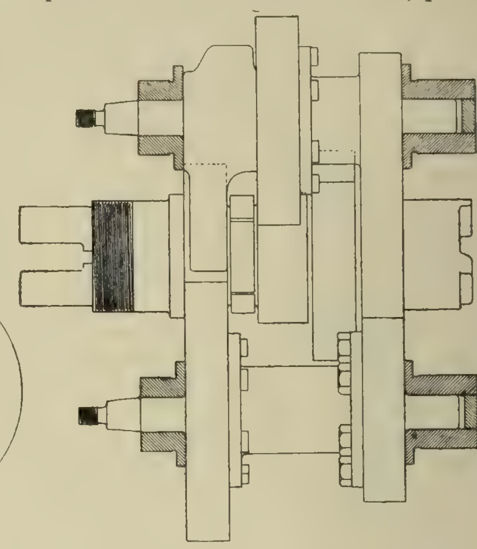


Fig VII.



adjustable stop shown comes into contact with the quadrant. Reversal of the quadrant motion, of course, reverses all the other motions, and so brings the gears out of engagement again. In order to start the rotation in either direction, there are stops on the back of the quadrants, which hit the steps in the periphery of the eccentrics, thereby knocking the projecting pawl into the quadrant slots, and ensuring the picking up of these parts.

Behind the gear box a foot brake of normal external design is situated, and thence the drive passes to the propeller shaft, which has self-lubricating joints of the pattern shown in Fig. IX. The universal joint is by no means the least ingenious part of this highly original chassis. There is sufficient space in the central opening (which is, of course, closed by a pair or flat plates) to contain enough lubricant for months of ordinary use, and as all grease passes outwards it keeps the bearing surfaces clean, while owing to centrifugal action, it has no difficulty in reaching the pins.

Towards the rear end of the shaft there is an extremely substantial telescopic slide, consisting of a square block in a grease-tight casing, which slide is shown in the chassis plan.

Concerning the rear axle, there is not much requiring more explanation than is given by the sectional view of Fig. X. Perhaps the most interesting feature is the manner in which the thrust from the road wheels is transmitted to the differential case through small thrust washers, and from the differential case to the main thrust bearings supporting the worm wheel.

Undoubtedly the use of slotted bolt-heads and nuts for the long bolts that secure the halves of the axle is to obtain a neat appearance, but the objections to the sacrifice of so much efficiency in order to please the eyes of the uninitiated are obvious. Perhaps there is no difficulty in obtaining sufficient tightness in erection, and also there may be no great difficulty in taking down an axle in the manufacturer's works, but designs ought to be such that any good engineer can execute repairs to any part, and there can be no doubt that these screw heads would be very troublesome in most repair shops, after the car had been in use some time, and slight rusting had taken place, as take place it must eventually. The dismemberment of the axle is rendered the more frequently necessary, too, by the absence of any drain plug, this making it very awkward to clean out the worm gear and differential efficiently, so encouraging laziness on the part of the driver.

As regards the front axle and the steering, there is but little requiring more explanation than is given in the chassis plan. The axle itself is tubular, provided with the same type of radius rods as the back axle, and also supplied with a central stay rod, which gives it the necessary lateral stability, the springs having too little lateral strength to resist all side stresses. An ordinary worm gear is used for the steering box, but the steering arm, of course, moves in a direction at right angles to the usual line, the connecting rod lying as shown in the plan of the chassis. All the steering joints are well proportioned, and there is an excellent angle of lock giving a good turning circle for so long a chassis—an especially good one for so large and comfortable a body as the chassis can easily support.

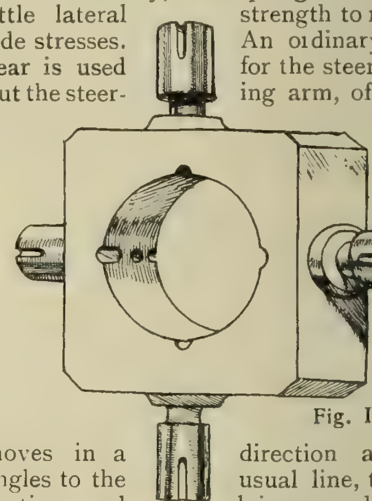


Fig. IX.

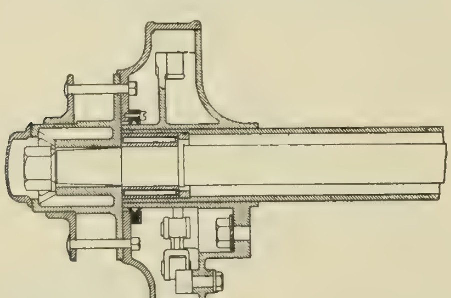
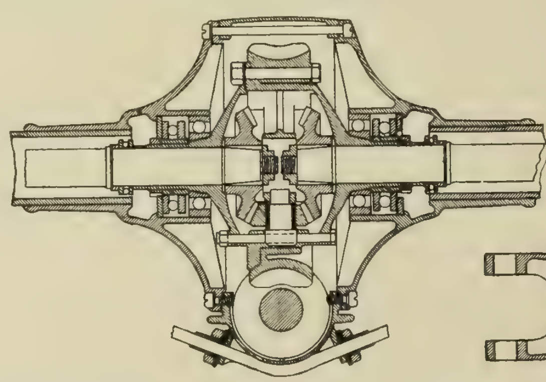
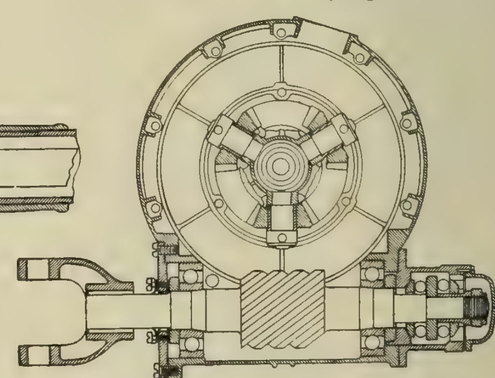


Fig. X.





THE MACHINING OF CAMS AND CAMSHAFTS.

FROM a designer's point of view solid cams have the advantage of smaller dimensions which allows of greater compactness. It is claimed also that being solid with the shaft there is no possibility of their moving out of position. Makers of special cam milling machines for this type lay considerable stress on the absolute uniformity with which they can be made, and by some engineers they are considered to be more silent in working than the loose type, but this last is a very debatable point. A great deal depends, of course, upon correct design, and if camshafts are made of unduly small diameter, and are insufficiently supported by bearings placed too wide apart, twist

must result, whichever type may be used.

It will be admitted readily that the solid cam is—or can be—an excellent job, but nevertheless it does not follow that therefore the loose cam is inferior. On the contrary, the loose type possesses advantages of its own which are not shared by the solid type. The objection so often raised against it that it is liable to work loose on the shaft—can only be upheld in cases of inferior design or workmanship, and really owes its origin to the time when cams were only pinned to their shafts and not keyed. When they are keyed on with Woodruff keys they do not work loose.

Then, as regards accuracy, there is no difficulty at all in making a loose cam as thoroughly accurate as the solid type. It is a very simple matter to make a jig by which all the keyways can be cut to absolutely accurate angles, and the cams themselves can be made to equal exactness for there is no reason why they should not be ground up in place on the shaft, as is done in the case of solid cams. As a matter of fact, however, this is not found necessary, for if the keyways in the shaft are milled in a good jig and the cams ground after hardening, they are found to fit quite accurately. It is an advantage that the shaft need not be hardened, for there is no distortion to fear, and it is to be noted that in the case of the solid type there is always perceptible distortion after the hardening process, which necessitates grinding as a finishing operation for both cams and shaft.

If it were not that the hardening process has reached a high degree of excellence and durability, an objection might be urged against solid cams that if wear takes place it necessitates scrapping the whole shaft. In any case, it must be admitted that the facility with which a loose cam can be changed or renewed is an advantage. Coming to the question of expense of production, the loose cam has decided advantages, so all things considered, it will be seen that the loose cam can hold its own very creditably.

Another point which crops up—mainly one of design, though it affects manufacture considerably—is whether the profile of a cam should be concave or convex on either the opening or closing, or on both sides, for to make a profile which is slightly concave necessitates the use of a grinding wheel of very small size, otherwise the radius would not correspond at all with the plunger roller and there would be very little gained. It would be necessary also to use a milling cutter of small diameter for the same reason.

At the same time the fact must not be overlooked that the cam with concave flanks is a valuable adjunct to engines for certain purposes. It is now being employed largely by manufacturers of engines from which the last ounce of power is desired to be obtained. For this reason it is highly probable that concave faced cams will increase in number, despite the manufacturing difficulties. Of course, the efficiency of an engine must always depend upon the absolute accuracy of the form and setting of its cams, and for this reason it is perhaps surprising that cams with a much larger mean diameter have not so far been employed. Certainly such design would simplify the task of making cams, and would improve their durability, but it might sometimes be difficult to accommodate a more bulky valve mechanism without increasing the size of the crankcase very considerably. Of course, the larger a cam is the greater is its distortion in hardening likely to be, but this matters very little, because of the cheapness and rapidity of the final grinding operation almost invariably employed.

Probably the mild craze for solid camshafts has exerted some influence upon the choice of small diameter cams, and if the normal size was to be doubled the solid shaft would become very expensive. Therefore if a concave cam face is called for by designers, there would seem to be many good reasons for urging the use of loose cams and an increase in size in order to facilitate the machining to absolutely accurate profile.

A simple way of making loose cams, and one which does not require a special machine for the purpose, is as follows: A steel plate about $\frac{1}{4}$in. thick is cut to the exact form of the cam and is fitted with a position stud, and a hole corresponding with the camshaft, this being then carefully hardened. The blank from which the cam is to be made, drilled with a hole to suit the position stud, is dropped on a stud in a vertical milling machine, the stud being just long enough to take the former plate, which is firmly nutted down. A cutter with a plain cylindrical shank is used, and is arranged so that the plain portion comes opposite the former plate. All the operator has to do is to feed up the blank and rotate it against the cutter until he reduces it to the shape and size of the plate. This may appear rather a slow process, but it is surprising what good speed can be made after a little practice.

Fig. I. shows such a machine by C. W. Burton, Griffiths and Co. This machine has a former shaft at the back, revolved by worm gearing in unison with the work and having a keyway cut along its entire length. On this shaft are placed as many cast iron formers as there are cams to be cut, and these control the shape, turning and position of the cams longitudinally. Once a set of formers has been fitted, the machine will go on producing any number of shafts, which are exact duplicates of each other, without any further adjustment being required. The machine will generate its own formers, all that is needed for this purpose being to have a master former which is an exact duplicate of the cam to be produced. This is mounted on a suitable arbor and carried on the former shaft driving spindle, the former shaft having been previously removed. With the machine, arbors are provided for the master former, with four keyways or six keyways, according to whether four-cylinder or six-cylinder camshafts are required, the keyways being correctly spaced for the angular relation of the cams, so that it is quite a simple operation to finish former blanks after they have been bored

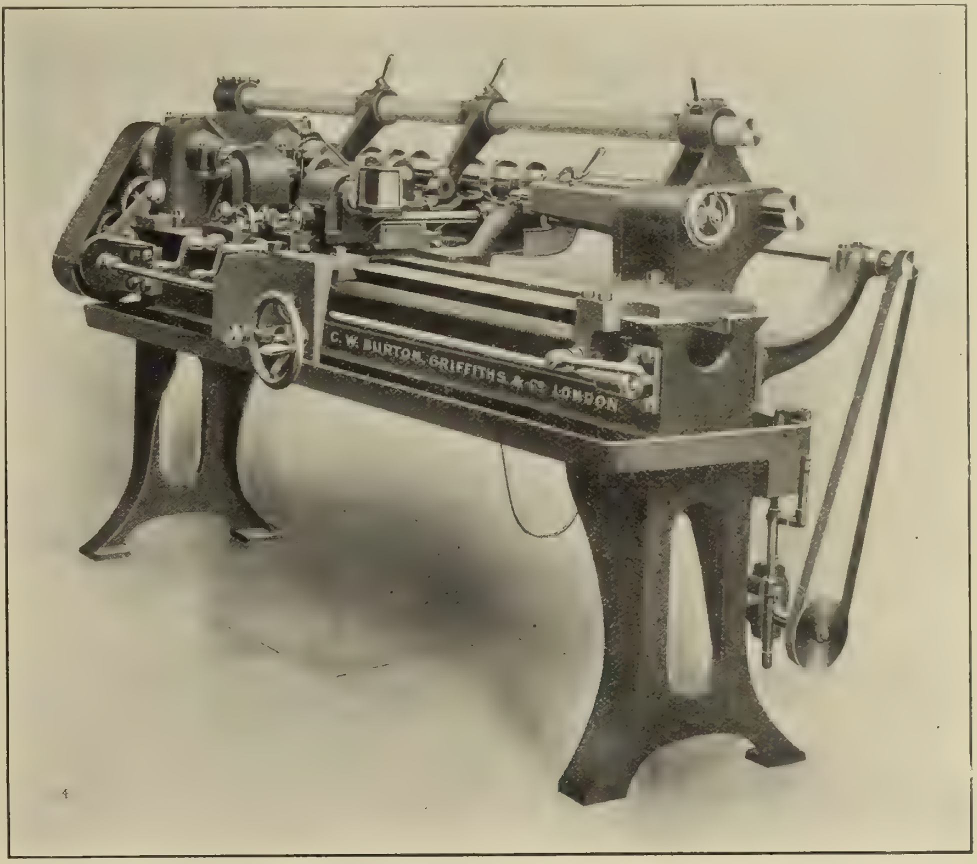


Fig. I.

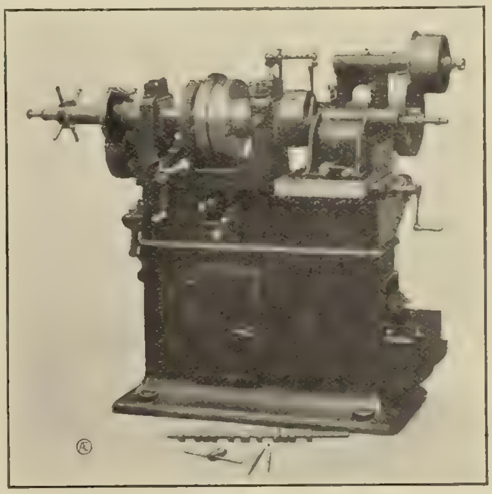


Fig. II.

and have had their respective keyways cut in them.

The saddle carries two milling heads, each positively driven by bevel gear from a driving shaft at the back of the machine. One of these heads carries a high-speed end cutter, for rapidly roughing out,

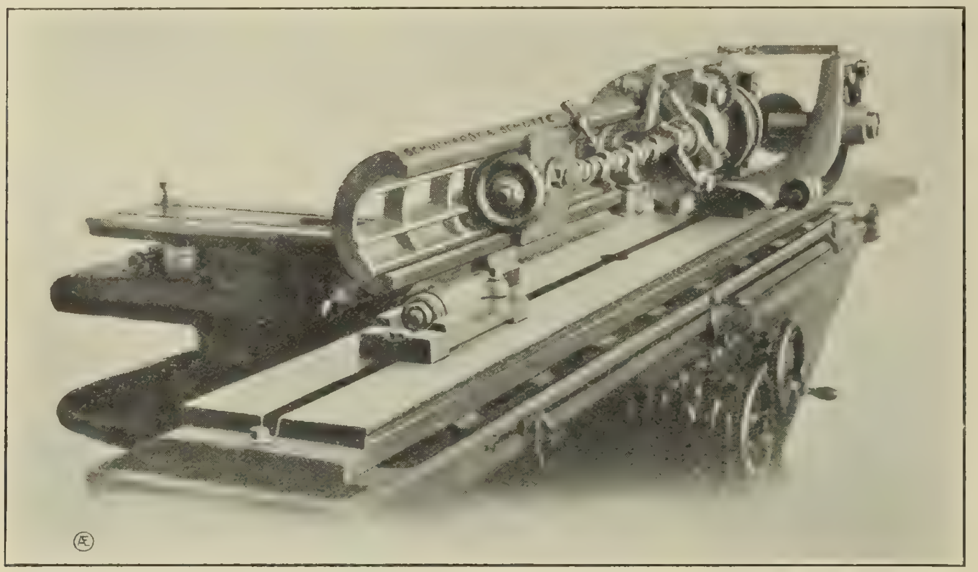


Fig. III.

whilst the other carries a face cutter, which finishes the work accurately and smoothly. The saddle has a former traverse, operated by spur gearing from the head-stock, but there is also a quick traverse by hand. A shaft, shown along the top of the machine, carries any desired number of steadies, of which two are included as part of the standard outfit.

This machine is designed to take a grinding head for finishing the camshaft after hardening, but the desirability of such an arrangement may be doubted, unless under special circumstances. Combination machines, as a rule, are not so satisfactory in regard to economy of output, as separate machines for each distinct operation. At the same time there are many occasions where, owing to small output or other similar causes, there is not work enough to keep a single purpose machine fully employed, and the attachment might thus often be extremely useful.

Fig. II. illustrates a machine of a somewhat different type; supplied by Schuchardt and Schutte. This can be used for either loose or solid type cams, but when used for the former it is recommended that the cam blanks be keyed and fixed on to the shafts before milling, so as to ensure absolute accuracy of position.

The formers on this machine are carried on the main spindle, and a space is provided for carrying six at once; much economy of time is gained, as the necessity for changing the formers for different cams is obviated. Formers can be scribed out on the machine by fitting a finished cam in the usual way, and substituting a roller for the cutter, and a scriber point where the former roller should be. By then pulling the machine round by hand the correct shape is marked on the former blank, which can then be taken out and machined up to shape.

The camshaft is prepared in the lathe in the usual way, leaving collars to the finished width of the cams, and of a diameter sufficient to allow of a slight cut being taken from the highest point of the cam. In milling, the shaft is held in a sleeve that passes through the main head-stock spindle, and which can be moved longitudinally. A three-jaw chuck is provided on the nose of the spindle, and in this the shaft is gripped close to the cam that is being cut. This chuck has to be released to enable the shaft to be moved longitudinally, but the shaft is held all the time by the internal tube, so that the indexing motion, which is connected with the latter, gives the correct positions of the cams. A steady is provided to support the camshaft on both sides to ensure accurate work, and is undoubtedly of great assistance in this respect, particularly for rapid work.

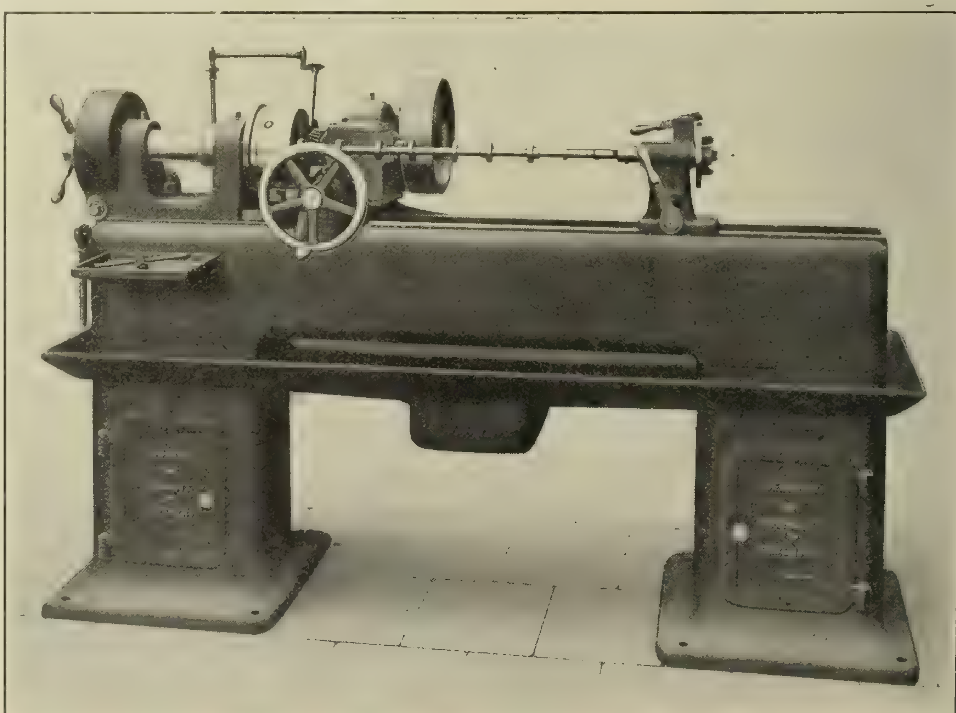


Fig. IV.

The cutter head is held up to the former by means of a weight, and there is a hand lever for moving the cutter away from the work when this is desirable.

No matter how accurately cams may be cut in their soft state, it almost invariably follows that after hardening there is found to be perceptible distortion, and the only way to deal with this satisfactorily is to leave a sufficient margin for a final grinding operation, for which a special type of machine, or a special attachment to fit to a standard type of grinder, is just as necessary for economical production as is the cam milling machine itself.

Fig. III. is an illustration of Schuchardt and Schutte's automatic grinding attachment, which can be used with plain cylindrical or universal grinders. It consists of a self-contained frame, with a line head centre, and an adjustable centre head, suitable for carrying the camshafts or arbors between the centres, as may be seen in the illustration.

The frame is pivoted in centres so as to oscillate truly and freely without backlash. The work spindle is driven from the overhead motion in the same way that the ordinary line headstock is operated, and all the normal automatic motions, feed, and delicate adjustments of the grinding machine apply in every way as in cylindrical grinding. The former cams are carried on a sleeve fitting the spindle at the outer end loosely, and engage with a division plate, fixed to the spindle, which sets the relative angular positions of the cams on the shaft correctly throughout. Four shapes of cams can be used on the sleeve, and can be readily removed, without disturbing the spindle. The former cam is held up in contact with an idle roller by a long flexible helical spring, placed in a mean direct line over the formers. Owing to the arrangement of pivoting the work frame in centres attached to the machine platens, the possibility of oblique stresses or side movement of the work is reduced to a minimum.

Webster and Bennet, Ltd., are the makers of the machines shown by Figs. IV. and V., which are respectively cam milling and cam grinding machines. In the milling machine (Fig. IV.) the camshaft is held between centres as shown. The cutter spindle is carried on a cross slide, which moves to and fro at the back.

The copyholder is mounted on it, and is pressed against the former cam by means of a strong spring, while there is a micrometer adjustment for gauging the work as it progresses. The former cams are

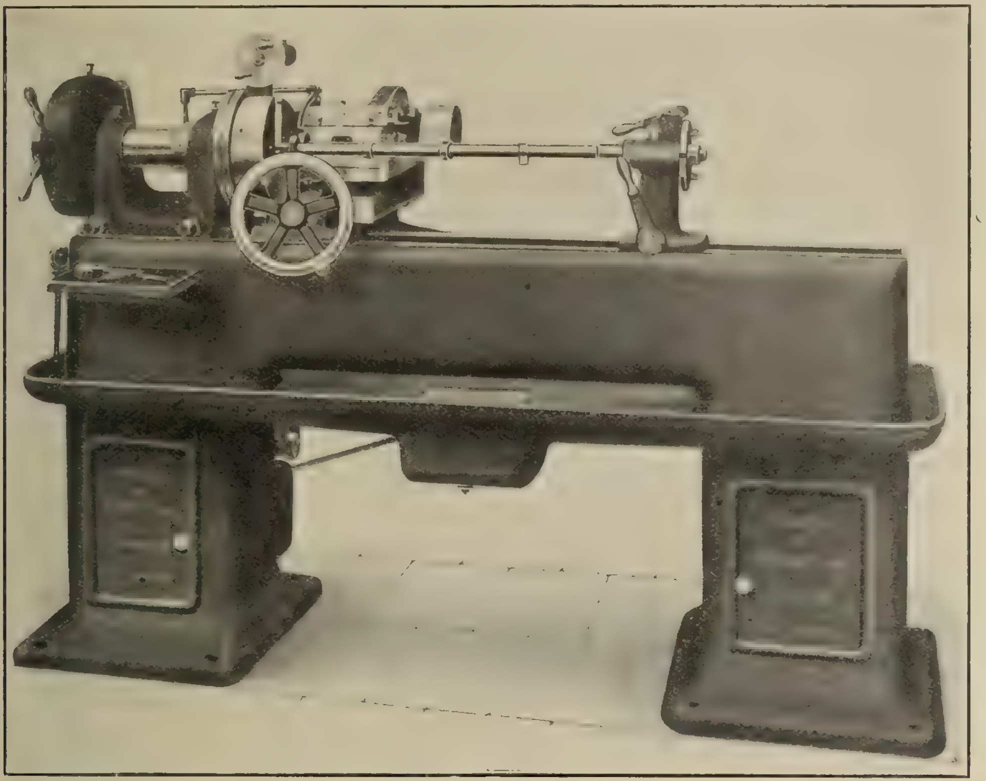


Fig. V.

mounted on a hollow spindle, and are of various outlines to produce any desired shape of cam. The work is revolved against the cutter by a worm gearing,

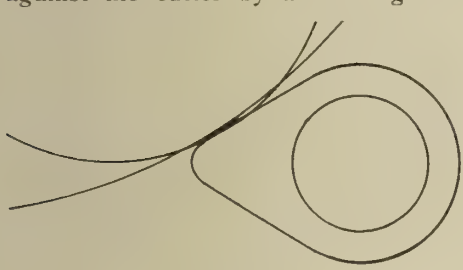


Fig. VI.

controlled by a friction plate on the hollow spindle, and if the cutter belt should break, the feed stops. For the first revolution of the work the depth of cut is regulated by the hand-wheel, shown in front, but on the second or finishing revolution the cutter is controlled solely by the former unaided by any hand adjustment.

The grinding machine shown in Fig. V. is designed to finish camshafts only. The general construction is very similar to that of the milling machine, and the arrangements for fixing the work and operating are so similar as to hardly require detailed explanation. A useful feature is to be noted in the reciprocating motion given to the emery wheel to keep its face true.

In the article on "Grinding Machinery," which appeared in our first issue, the possibilities of grinding in supersession of milling and various other operations was dealt with at some length, and we therein illustrated the Landis grinder, which is specially suited for cam production. This machine will grind shafts with eight cams from the rough at the rate of three per hour, which is perhaps a more rapid output than could be obtained by milling. A machine that will do such work at such a speed is bound to make its influence felt, and the fact that it can be used for a large variety of other work, giving equally satisfactory output, is an additional recommendation, if one were needed.

Attention is called by the makers to the fact that in grinding cams peripherally the quality of the work must alter with the gradual reduction in diameter of the grinding wheel. This effect is shown by the diagram, Fig. VI. Even granting that this is shown in an exaggerated way, it must be acknowledged that in the nature of things the work on the cam with that class of grinder cannot be as good as is obtained by face milling, see Fig. VII., which is the special characteristic of the machine under notice. But it will be seen that if a cam is required with a concave profile this machine will be at a disadvantage, for it can only deal with straight or convex profiles.

Of course, the use of this machine in place of a miller does not obviate the final grinding after hardening, but the setting up of the work is very simple and easy to do, and only a very fine allowance need be left for finishing, so that the latter operation does not take long.

For rapid production it is questionable whether the grinding machine will not be found to beat the milling machine on equal terms, and we look to see further

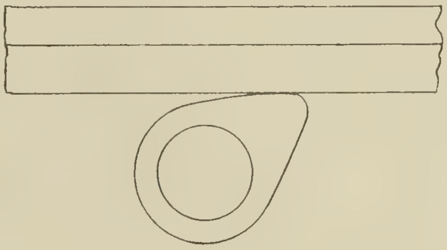


Fig. VII.

development along this line, having regard to the fact that accuracy of work is in no way sacrificed.

GASEOUS EXPLOSIONS.

An abstract of the recent work of the Committee of Investigation.

SPEAKING broadly, there is but little of direct practical value to automobile engineers in the third report of the committee of the British Association, who are engaged on an examination of the physical and chemical characteristics of gaseous explosions. The experimental work which has been carried out during the past year was principally similar to that performed during the previous year, being largely connected with the investigation of heat radiation from flames.

Probably one of the most interesting practical conclusions was that the nature of the internal surface of a vessel in which an explosion takes place has a detectable effect upon the temperature and pressure of the gases immediately after explosion, and also upon the *rate* of loss of pressure and temperature during the expansion, which follows explosion in an engine cylinder. If the inside of an explosion chamber is black and rough, forming a surface which absorbs radiant heat readily, there the loss of heat is most rapid. On the other hand an increase of pressure of three per cent. was obtained by polishing and silvering the inside of the vessel.

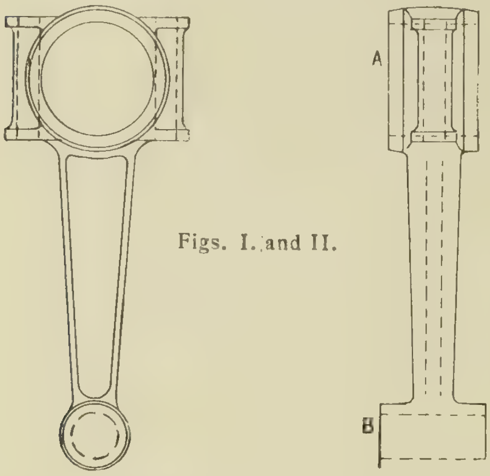
It is also pointed out in the report that the loss of heat by radiation increases with the richness of the explosive mixture, and at a greater rate, which means that a greater percentage of heat is likely to be lost in an engine using a rich mixture than in even a slightly larger engine using a weaker gas.

Other experiments have shown that a burning gas is transparent to its own radiation; that is, that a flame is transparent to radiant heat. This transparency is, however, not very good, so the lesser efficiency of engines with small cylinders, as compared with those having larger dimensions, is due almost as much to radiant loss as to loss by conduction with the walls. Professor Callender stated that the percentage of heat lost by radiation alone might safely be taken to vary inversely with the diameter of the cylinders for otherwise similar engines.

Referring to the suggestion that the nature of the surface of a combustion chamber has an effect upon the mean pressure of explosion, it, of course, follows that the efficiency of an engine is likely to increase as its internal cylinder finish is improved, but it is also obvious that the difference between one cylinder and another is hardly to be noticed as soon as a few explosions have taken place, as the smallest possible deposit of carbon would transform the most carefully-polished surface to the opposite extreme—so far as radiation is concerned.

JIGS & TOOLS FOR MACHINING CONNECTING RODS.

THERE are several points of similarity in the design of the great majority of connecting rods used by automobile engine builders to-day, a notable exception being the tubular type as used by the Lanchester Company. Generally speaking, however, the design most adopted is very similar to the sketches shown in Figs. I. and II. herewith, which



Figs. I. and II.

show a drop stamped steel connecting rod for a 20 h.p. to 25 h.p. four-cylinder engine.

The accurate machining of these drop forged rods presents many problems to the machine shop superintendent or works manager, and affords ample facilities for the jig and tool designer to devise a rapid and accurate means for their cheap production, as they are generally put through the shops in batches of not less than a hundred at one time. Most petrol engine builders have had difficulties with their connecting rods at different times, the stampings or forgings being of such a light, springing nature that the great difficulty has been in boring out the two ends, for the crank-shaft and gudgeon pin, in dead alignment to each other. This is of paramount importance, as if the two ends are not parallel to each other great trouble will arise when assembling the engine. It is quite safe to say there are very many ways of machining connecting rods, each method being performed with varying degrees of success, but the methods here described have been found

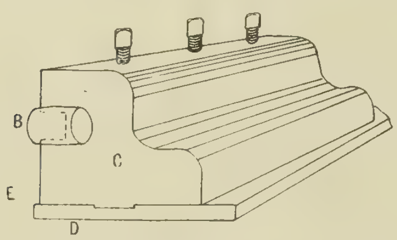


Fig. III.

to give better results than former practice, whilst the work will pass a very rigorous inspection for accuracy.

After the forgings have been inspected for soundness, and issued from the rough stores to the machine shop, they are taken to a 30 inch disc grinder, and the two faces, A and B, in Fig. II. are ground up true, forming the basis for subsequent operations. As the stampings are now made within fairly fine limits it will be understood that very little stock will be removed in this grinding operation, which occupies about eight minutes, including clamping and taking out of the fixture, as shown in Fig. III. The cast iron block C has a tongue planed on the bottom which fits in a slot in the table D, as shown. The block C simply has a slot milled across the front, as shown dotted. The forgings fit in this slot, and the three clamping screws, shown on top, secure it in position. It will be observed that these screws grip the stampings on their edges, and therefore there is no tendency to spring the rod out of truth, as is the case when it is gripped on the flat face, for milling. The two faces, A and B, are ground to the flat steel gauge, Fig. XII., the two straight faces, E and F, being the register distance. Each face is ground separately by swinging the table and fixture round its axis by means of the usual bottom lever supplied for the purpose on nearly all disc grinders.

After being inspected for the grinding operation, the stampings now pass on to the drilling section and are drilled and faced for the bolts. The small end is centred in an upright drill, in the box jig shown in Figs. IV. and V., where F is a light box casting planed top and bottom and machined on the faces G and

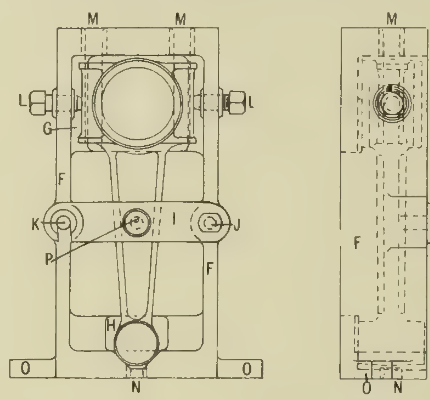


Fig. IV. Fig. V.

H to locate the rod from the ground faces A and B. I is a swinging steel clamp, pivoted on the pin J, and secured by the stud and wing nut K, whilst P is a knurled screw for keeping the work on the machined faces G and H. L L are two adjusting screws for securing the work in an upright position in the jig. After the first rod is set in the jig, only one screw is slackened each time so that no further setting is necessary. M M are the two hard steel guide bushes for the high speed twist drills (N is a similar bush for centring the small end, which rests in a vee as shown), and together with the two clamping screws L L keep the stampings in a rigid position in the jig; O O are two lugs for securing the jig to the drill press table if necessary. The two bolt holes are drilled and reamed in this jig, and the small end centred. By drilling the bolt holes before the large end is bored there is a clear, straight path all the way through solid material, which is not the case when the rods are bored first, for then the drill runs out into the previously bored hole, the holes are not then straight, and broken drills are the result; while with the method here shown these defects do not occur.

After the rods are drilled and reamed they are faced on the same machine in the jig shown at Fig. VI. The rods are simply dropped on the two hard steel pegs Q Q, which fit in the bolt holes, the long end hanging down through the slot R, and the two holes are faced with a four-lipped facing cutter, or counterbore, with a pilot end fitting in the holes. When one side is completed the rods are simply turned upside down on the pegs Q, and the opposite sides are completed.

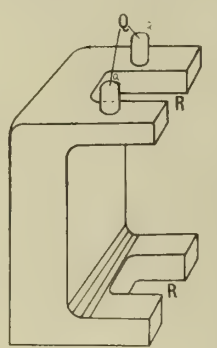


Fig. VI.

The time for this operation complete, including setting up, will be about twenty minutes per rod.

After another inspection for the previous operation, we now come to the biggest and most important machine operation concerned, that of boring and facing both ends, and this is where trouble generally commences, by springing the stampings; but if ordinary care is exercised, and no brute force exerted, the stampings should not be sprung if bored in the fixture shown in Figs. VII. and VIII., whilst Fig. IX. also shows it mounted in position on the capstan lathe faceplate. S is a cast iron plate planed up on the back face, while T, Fig. VIII., shows the lathe faceplate with S attached to it, and a stamping in position for boring. U is a stout steel pin secured in S, nearly half-way between the two brackets X and Y, which are cast solid on S. V V are two tapped holes, and W W are two plain reamed holes for a locating peg, which determines the position of S when boring the rods, the large pivot bolt and nut U, and one set screw V, and one locating pin in the hole W, which passes through both S and T, keeping the fixture and work secure on the face plate T. Z is a hard steel set screw passing through the bracket X, and A and B are two shouldered, hard steel setting or locating pins, driven tight in X

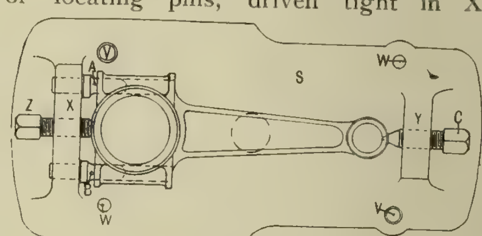


Fig VII.

exactly the same distance apart as the bolt holes in the rod end, into which they are a good sliding fit without any shake whatever. C is a hard steel pointed set

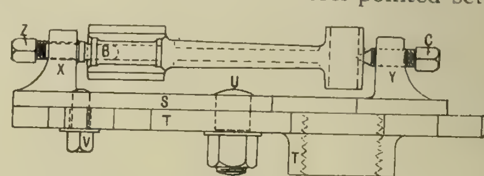


Fig. VIII.

screw, which fits in the centre hole in the small end of the rod as shown (which also acts as an oil hole when the rod is completed), and forces the forgings tight up against the screw Z. Thus the rod is set central by the two adjusting

screws Z and C, and held firmly by the two pegs A and B. After one end is bored, reamed and faced, the set screw V is slackened, the plunger peg W withdrawn and the fixture turned round half a revolution on the face plate, when the plunger peg W will locate it accurately for boring the opposite end. Thus all the clamping strains are at the ends of the rods and not at the flat sides.

The boring is best done on a capstan lathe, either with the head and capstan

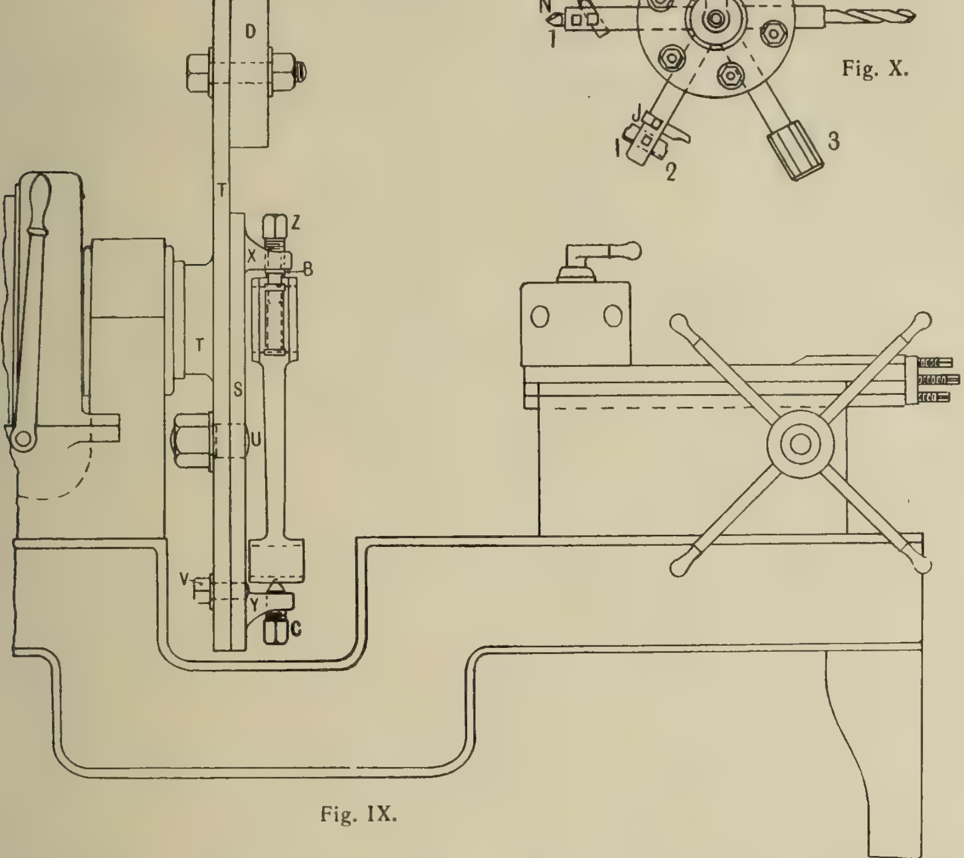


Fig. IX.

packed up, to give the necessary swing, or the head and capstan may be fitted on a gap bed as shown in Fig. IX. D shows a balance weight secured to the face plate to neutralize the weight of the work and fixture, and to give an even turning movement. The lathe should be friction back geared, but it is not absolutely necessary to have an automatic feed to the capstan as the work is of a comparatively short nature, rarely exceeding $2\frac{1}{4}$ or $2\frac{1}{2}$ inches long: therefore a hand feed can be used, the ordinary star with rack and pinion often being employed. Fig. X. shows a plan of the capstan and tools in position in the proper sequence for the various operations involved. 1 shows a stout cutter bar, with single point cutting tool G for opening out the large end, whilst H is a very short stiff centring drill for centring the small end, 2 is a similar bar with a sizing cutter I, and facing and radiusing cutter J, whilst 3 is an adjustable reamer, for finishing the large hole to exact gauge size. 4 is the twist drill for drilling the

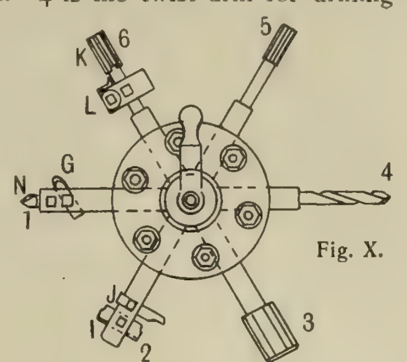


Fig. X.

small end from the solid after being centred by H. 5 is a solid reamer for reaming it out, and 6 an adjustable reamer, for finishing the hole out to size, whilst L is a small facing attachment secured on the shank for facing and putting a very small radius on the small end. The stampings are now taken out of the jig, and when the batch is completed they are turned round in the fixture and faced and radiused by two hardened and ground steel bars, as shown at Fig. XI.; these bars exactly fit the two holes in each end of the rods, and therefore set the work accurately by just dropping them on it until they are secured in the jig. The bars have three flutes cut in the end to collect dirt and small chips, etc., which may accumulate in the bored holes; bar No. 2, Fig. X., can have a pilot end on, steadied by a bush up the lathe hollow spindle to give additional rigidity, and the reamers 3 and 6 should also be made to float.

The time for the complete operations of boring, reaming, facing and radiusing the two ends as described should not exceed 60 minutes.

The rods are now passed on to the inspection department and undergo a rigorous inspection, two ground parallel mandrils, each fitting one end of the bored rods exactly, and not less than, say, 12 inches long, being inserted, and an inside micrometer used to gauge the distance

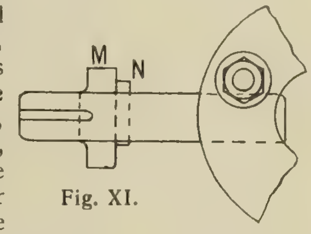


Fig. XI.

between them to see if the holes are exactly parallel and straight with each other. Should they be found to be out of line this can generally be remedied by slight hammering and setting on the surface table. Having come through this test the rods are now passed to the milling department, and the large end is sawn across the centre with a slitting saw, which will take another six or seven minutes each, the time occupied in setting up.

The stampings, gun metal bushes, bolts, caps, etc., are now issued to the assemblers, who fit the rods up complete, the minor operations and processes being to bed the gun metal bushes in the large end and cap, to insert the gun metal bushes in the small end, to chip oil grooves and file them out to a good finish in the large end, put the

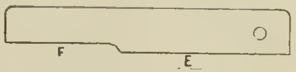


Fig. XII.

two securing bolts in rod and cap, and also put small stop pegs in the bolts to prevent them from turning round, also to file the bushes at the side where they foul the bolts owing to the bolt holes running into the large hole, and assemble the rod complete. The total fitting and assembling time on a 20 h.p. rod of standard type, assuming the man employed is a first-class hand at the job, should take about one hour forty-five minutes, providing the machining of the various component parts is accurate, wherein lies the whole secret of assembling. This gives a total labour time of three hours nineteen minutes per rod complete, and the times allow for first-class workmanship of the highest quality.

THE 15 H.P. STAR CHASSIS.

In the design of the 15 h.p. Star an attempt has been made to pay equal attention to cost of production and efficiency. Though devoid of luxurious mechanical fittings the chassis has practically every necessity, and if the finish is not too high externally it is of a high order of merit where necessary; in fact it is not too much to say that the essentials have been cared for in a thorough manner, and that costs have been kept low by a severe simplicity of individual parts, and the reduction of the number of parts as much as possible. The chassis would probably withstand considerable rough handling, and its strength, using the term in its widest sense, is certainly above the average of cars with similar engine dimensions.

One of the manufacturer's principal aims has been to obtain high power from the engine, which has a bore of $3\frac{1}{2}$ ins. and a stroke of 5 ins. In the endeavour to combine high speed of revolution with high mean explosion pressure some sacrifice of smooth running qualities has had to be made, but the engine is not conspicuously hard running and may be said to be decidedly well balanced at ordinary speeds.

Fig. I. makes clear the simple construction of the engine, and a transverse view is not shown because it would disclose practically nothing of interest beyond the fact that the crankshaft is $\frac{3}{8}$ in. off-set from the cylinders. The latter are cast

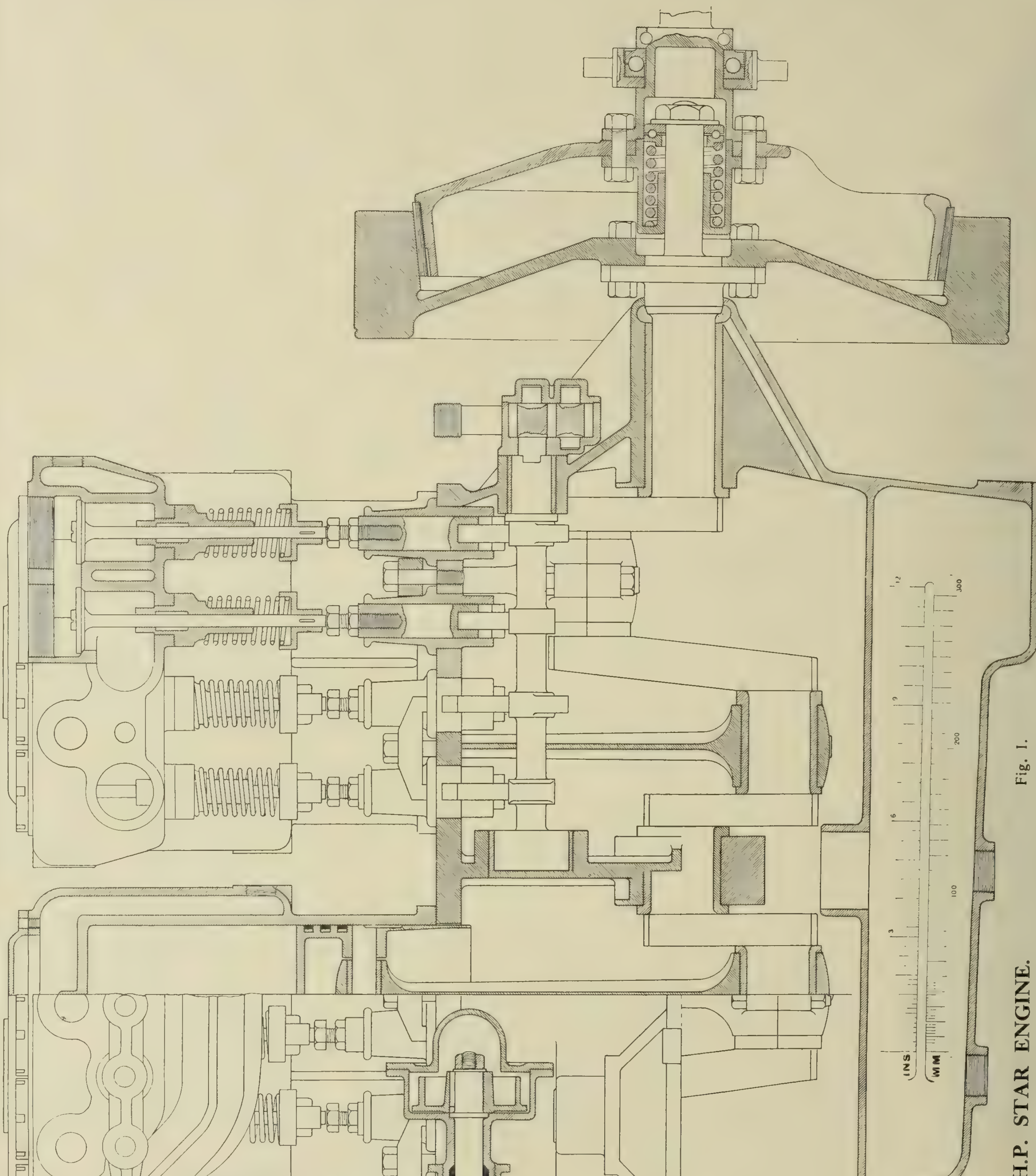


Fig. 1.

THE 15 H.P. STAR ENGINE.

in pairs, the inlet valve ports being connected to a central orifice in each casting in the usual manner. Ample water spaces are given, and the remainder of the cooling system is also above the average of size, being as large as would generally be employed for natural circulation, with the possible exception of the outlet pipes, though a very large centrifugal pump and a round-tube-section honeycomb radiator are used.

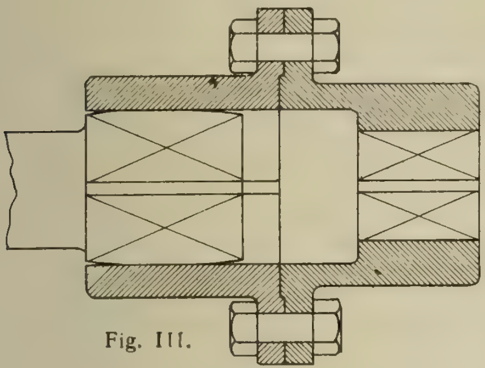


Fig. III.

The valves are of about normal size, the effective diameter being $1\frac{1}{2}$ ins. and the lift $\frac{3}{8}$ ins., though the latter dimension may be increased on occasion. These valves could be improved by the removal of the screw heads and the provision of a larger slot in the main heads, and it would also appear that a small increase in diameter could be obtained without much trouble, still no doubt they are sufficiently free for all ordinary purposes, as is proved by the behaviour of the engine. Tappets of the ordinary roller type with set screw adjustments are used, and are retained in pairs by single-stud dogs, which is an inexpensive and effective method even if its conspicuous advantages from an accessibility point of view are disregarded.

Four long studs are used to secure the exhaust and inlet piping, there being two dogs with a central set screw in each whereby the domed ends of the inlet pipe are held against their seatings. This arrangement has the advantage of simplicity and comparative cheapness, but there is no doubt that it is not so easy to make secure exhaust joints as when there is a separate flange to each pipe end, and though a separate flanged pipe takes much longer to attach, its removal is so infrequent an occurrence that the greater security and neatness of the flange seems worth consideration.

The arrangement of the camshaft and water pumps may be seen by reference to Fig. I., where the size of the timing gears, which are $1\frac{3}{8}$ ins. wide, will be noticed, a similarly large gear being used for driving the magneto, which is situated on the off-side of the cylinders in a position corresponding to that of the water pump. Lubrication is performed by a variety of controlled splash, the pump deriving its supply from the crankcase sump through a cylindrical filter, and forcing jets of oil which spray on the crankshaft. As there is a central division in the bottom half of the crankcase sufficient oil is always retained therein to enable the big ends to dip, though to a quite small extent, and any surplus oil flows down the central overflow seen in Fig. I., so returning to the sump. The engine is also made with a drilled crankshaft and a forced oil supply, but the arrangement just described is the standard one.

There is no feature of the piston that is out of the ordinary, with the exception of the method of attaching the gudgeon pin, which is merely driven in and prevented from turning by a small key. It is claimed that this rather daring attachment has been found quite satisfactory in use, and that the gudgeon pins are not found to work loose. Though many of the gudgeon locking devices in use are altogether too complicated, this system would seem to depend upon the pin being a fairly tight fit in the piston, and as there is no particular trouble in putting a ring over the pin there seems no reason why it should not be done in this case.

As regards materials, the cylinders and pistons are of the customary cast iron, and the crankcase is an ordinary variety of aluminium alloy. Steel with a tensile strength of from 40 tons to 45 tons is employed for the crankshaft, the bearings being white metal, and the timing gears are of mild steel and phosphor bronze. Ample bearings are provided for the crankshaft, which is itself very heavy in order to give sufficient stiffness with only three bearings. The total length of main bearing on the shaft is nine inches, the big ends are each $2\frac{1}{2}$ ins. long, and the diameter of both shaft and pins is $1\frac{5}{8}$ins. : the thickness of the webs should be observed, as this is the most important factor in preventing whip, and it may be added that their width is two inches.

The engine is carried on an under frame by four crankcase arms, and as the shield extends only from the frame side to the under frame, the oil filter is reasonably accessible. The valves are not too easy to get at when all the piping is in position, and it would seem to be possible to arrange the inlet pipe in a somewhat different manner with advantage, but the matter is not of great importance as the whole carburettor and its fittings can be detached easily. It would also be a small improvement were it possible to repack the water pump without taking it to pieces, though for accessibility the gland adjustment could scarcely be better placed.

Concerning the clutch, the only detail which requires more explanation than is given by Fig. I. is the thrust bearings, which are made by the Star Company themselves. These peculiar shaped races are to be found in the back axle also, and it is claimed that their durability is excellent. Simplicity is again the dominant feature of the clutch and gear-shaft coupling and is it possible that it has been carried too far in this particular instance, as the amount of universal motion allowed by a joint of this type is extremely small while, if any motion does take place, the wear is apt to be rapid and knocking is liable to arise as soon as the smallest possible amount of slack develops. On the other hand disconnection is quick and easy if it is desired to remove the clutch from the flywheel.

The provision of a fourth speed in the gearbox is a good example of the maker's desire for efficiency even at the expense of a slightly increased selling price, and the 15 h.p. Star is in a position of decided advantage over most of its immediate competitors by virtue of this extra gear ratio. Fig. IV. shows the general arrangement of the gearbox; the direct third speed and the geared up fourth are controlled by one striker, the first, second and reverse speeds by another, and the means by which the last named gear is obtained without the use of an additional striker is distinctly neat. When the lowest forward speed has been engaged the large wheel on the driven shaft then just clears the rearmost of the two reversing wheels, further movement then simultaneously disengages the forward speed and engages the above-mentioned back half of the duplex reverse pinion. If the movement is continued still further, both the large wheel on the driven shaft and the duplex pinion move together, and the forward end of the latter comes into engagement with the fourth speed layshaft wheel. Thus the part of the gate which gives the reverse speed is merely a continuation of the same slot as that which controls the first and second speeds. This makes a simple gate and a simple striking control,

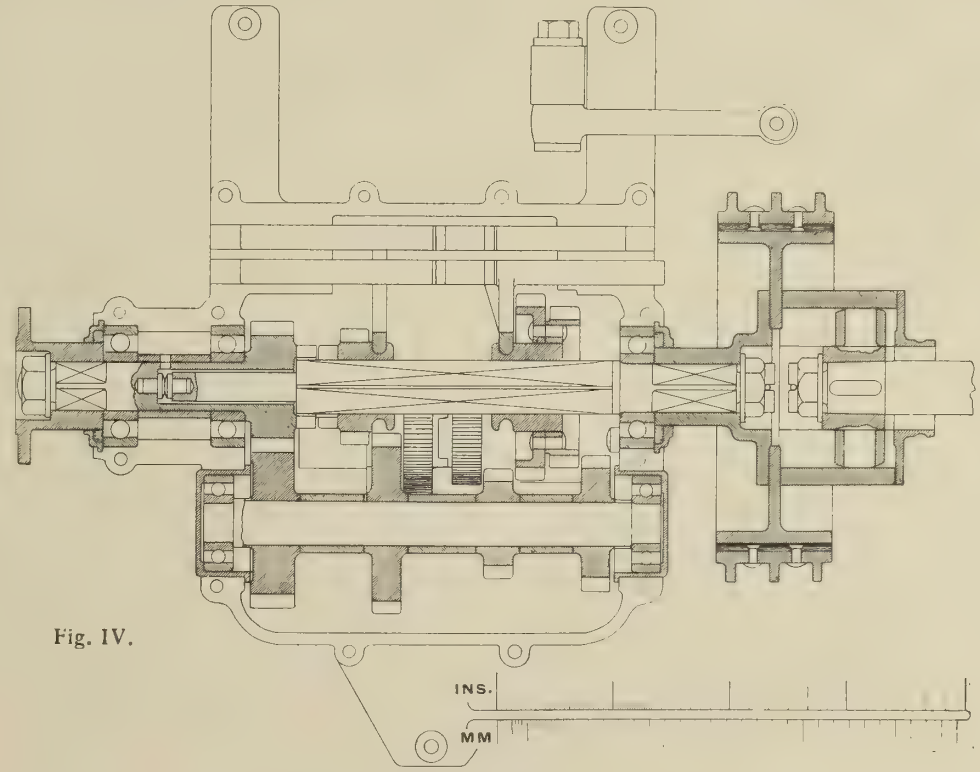


Fig. IV.

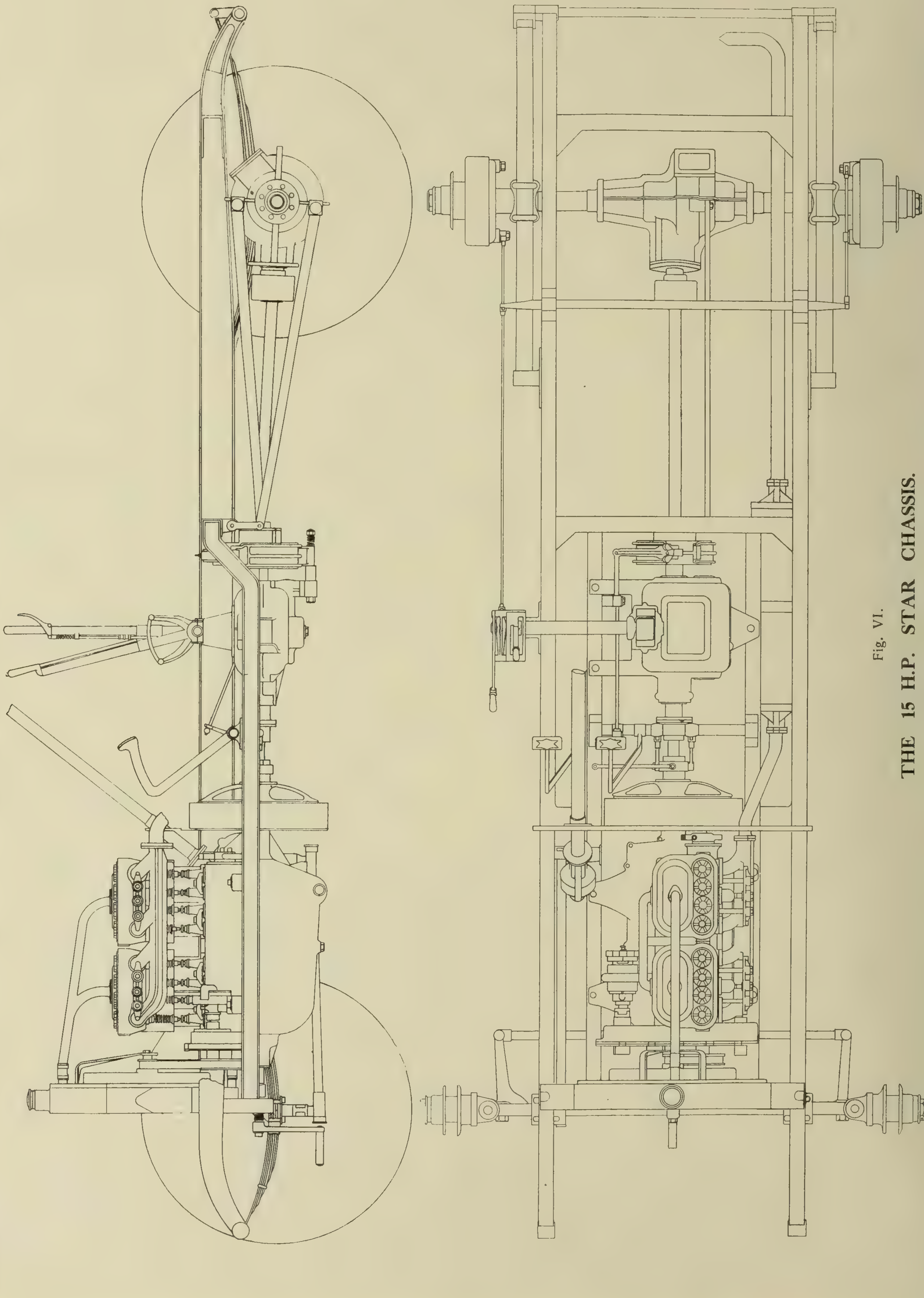


Fig. VI.

THE 15 H.P. STAR CHASSIS.

but it has the disadvantage that the first speed has to be passed through in the act of entering the reverse, still this is not a matter of great importance, and is not a practical inconvenience, or in any way troublesome.

The gears are all made from mild steel stampings which are, of course, casehardened. The layshaft is a stamping and the striking rods and arms are also stamped, the only part of the gear made from the bar being the main shaft, which is of casehardened fifty ton steel, the gears and other shaft being of thirty ton steel.

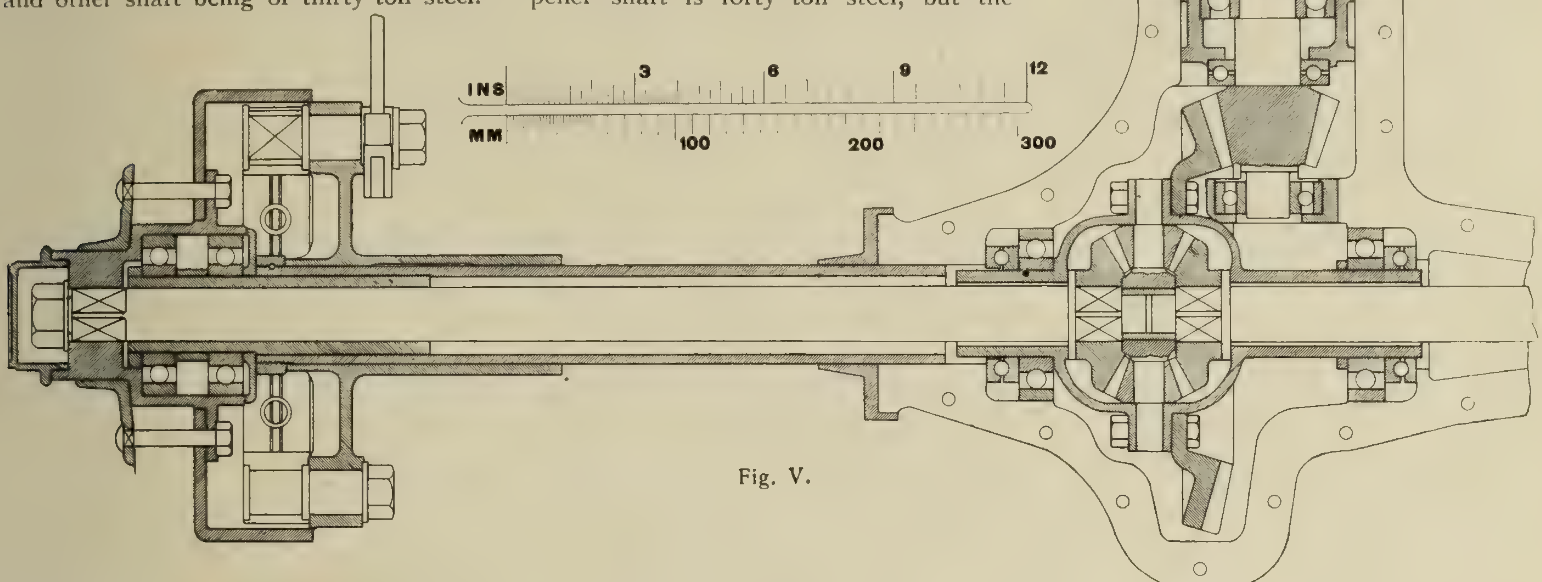


Fig. V.

Particulars of the wheels are as follows: —Layshaft drive 8 pitch, 24 teeth and 32 teeth. First speed 6 pitch, 15 teeth and 27 teeth. Second speed 6 pitch, 19 teeth and 23 teeth. Fourth speed 6 pitch, 26 teeth and 16 teeth. This gives ratios of, first speed 2.4 to 1, second speed 1.6 to 1, and fourth speed .82 to 1, while as the axle reduction is .28 to 1, the total ratios from engine to road wheels are, first speed 8.5 to 1, second speed 5.7 to 1, third speed 3.56 to 1, and fourth speed 3 to 1. The width of the gears is $\frac{3}{4}$ ins. in all cases, except that of the permanent layshaft drive, which is $1\frac{1}{4}$ ins. wide. It would appear, therefore, that an increase of width all-round would probably be beneficial as regards enhancing durability.

The general arrangement of the box is simple though it might be better if the shafts were reduced in length, and if a splined shaft were substituted for the square shaft; still it is far from being a conspicuously noisy gear, and the shafts are both large in diameter and well supported. A special grease inlet to the spigot bearing is supplied at the front end and should be observed, as it removes the disadvantage of the otherwise commendable length of bearing at this point. Every part is a simple machine job, there being no intricate pieces, and it is likely that it would be difficult to design a four-speed gear that could be made more cheaply with due regard to durability and effectiveness. The foot brake and the universal joint are recent improvements, as the latter has been increased in size and the former was hitherto of the internal expanding pattern with a pressed steel drum. As, however, this drum exhibited a tendency to open and become bell-mouthed the present design was adopted, the drum still being a pressing. Actuation of the contracting shoes is performed by means of the usual locomotive joints and requires no further description, the shoes being supported from the bottom of the gear-box, and the control being by the customary pattern of toggle.

Both the front and rear end universal joints are of the same pattern, and so have the advantage that the same parts can be used for either, which small machine shop economy would seem to have been overlooked by the majority of designers, for it is most unusual to find both joints of the same style. The propeller shaft is forty ton steel, but the joint ends are stamped and are fitted on tapers. Most of the telescopic motion on the shaft is supposed to take place at the rear end as there is a spring inside tending to keep the shaft as far forward as possible.

The axle is a conspicuously strong one, though study of the drawings would lead to the expectation that it is somewhat heavy; still, even if this is so it is an error on the right side. Malleable iron is used for the central casing, while the sleeves are weldless steel tubes, and the driving shafts are fifty ton steel.

Both the driving bevels and the differential pinions share the massiveness which characterises the whole of the axle, and they might be expected to wear unusually well; the driving bevels are 5 pitch and have 14 teeth and 50 teeth respectively. Particularly firm support is given to the bevel pinion by the large spigot bearing, and the other bearings are of good dimensions, though it would seem that if the thrust race behind the crown wheel is sufficiently large then the other must be needlessly so. Still, the fact of their being similar would enable them to be interchanged when wear of the more heavily loaded of the two did eventually take place, which is a distinct if small advantage. It might possibly be an improvement if the hub bearings were spaced slightly further apart at the expense of a small increase in the size of the hub shell; still, some support is given by the end of the driving shaft, and the latter would resist most of the lateral stresses on the wheel. The internal expanding brake follows common practice, and it need only be added that the drum is a pressing, while the shoes are cast iron. Free spring pads are employed, allowing movement of the sleeves within them, and the motion of the axle is controlled by a triangulated torque rod. One small, but useful, fitting, which is a noticeable absentee on this, as well as several other, chassis, is a drain plug in the bottom of the differential casing, though there is a large filling lid, and while a drain is not often required, its absence when it is needed may prove a matter of considerable annoyance.

Fig. VI., showing the plan and elevation of the chassis, gives a good idea of the compactness of the whole design, and also shows the springing and steering arrangements. The steering gear is a worm working with a complete wheel, the latter being made solid with its shaft, so to take up wear by turning the wheel the position of the steering arm is altered. There is also an eccentric adjustment to the wheel bearings which might be very useful with a complete wheel type of gear, as when the wheel had been turned and wear distributed all round it it would be possible to use an eccentric adjustment without causing any binding on the locks. Plain bearings are used in the worm box. The rest of the steering follows standard practice, as do the H section front axle and the front hubs.

Taken all together, while the chassis is not the very highest expression of automobile refinement, it is none the less a worthy representative of its type, and is of a nature to give long and satisfactory service without unusually careful handling. The great use of stampings, and the care with which each part has been considered from a workshop point of view, is such that it forms an excellent object lesson to those who are in the habit of using special tools for small operations, and do not consider cost piece by piece. Nor has the car suffered by this form of carefulness to more than a quite insignificant extent, as the simplification of individual parts has not taken the form of splitting them into a greater number of lesser parts, thereby resulting in troublesome and expensive erection. Both the details and the whole have obviously received well-balanced consideration.

FACTORS THAT HAVE CONTRIBUTED TO THE ADVANCE OF AUTOMOBILE ENGINEERING,

And which Control the Development of the Self-propelled Vehicle. Being an extract from Mr. F. W. Lanchester's Presidential Address to the Institution of Automobile Engineers.

WE are apt to imagine that until the advent of the modern period of steel construction and the invention of the heat engine, the march of improvement in matters relating to locomotion had ceased, and that both on land and sea a state of stagnation had been reached; such, however, is far from having been the case. The horse-drawn carriage was nearly as great a novelty in England about the year A.D. 1600 as the motor car in 1900, and during the three centuries that have intervened, its development and improvement have been nearly continuous, the art of coach-building made steady and uninterrupted advance; wheel building, springs, axles and bodywork, all shared in taxing the skill of the craftsmen employed, and all bear eloquent witness to the progress made.

Making a slight digression, it may be remarked that it is on record that the introduction of the horse-drawn carriage was met by a similar and quite as great a howl of opposition in its day as greeted the modern motor car a few years ago, and, moreover, the grounds of complaint were almost similar. Pedestrians and horsemen complained of the danger and the dust, and others there were who feared that the demand for a lighter type of horse would ruin the breed of charger necessary to the safety and defence of the country. In truth it is probable that the dust nuisance caused by horse vehicles driven fast on such roads as then existed, may have been far worse than that due to motor cars on the roads of to-day (*c*).

The above digression brings into prominence a matter of great weight, *the importance of road building*. There is no doubt that the attentions of the early carriage builder must have been directed to vain, or at the best, partially successful efforts to obtain good results in spite of the virtual absence of roads as we today understand the term. The engineer of early last century solved (or shelved) the difficulty by building his own roads of his own materials.

The importance of the road question is so great, both as to the past and the future development of the self-propelled vehicle, that I shall have to return to it later; it may be almost said that it is *the* question of the hour.

Returning to glance at the early development of locomotion on water we again find that under the old régime of wood construction and propulsion by sails, there had been, right up to the end, continuous improvement and development, punctuated by inventions of no mean order. The art of sailing to windward by tacking was in itself as great and important a discovery from the nautical point of view as the steam engine and screw propeller put together, and the various types of rig by which it is accomplished constitute as many important inventions originated to meet the varied conditons encountered, and all this discovery and development took place in the comparatively short space of some ten centuries or thereabouts.

It is of interest to note that whereas the progress of invention and improvement in carriage building and locomotion on land has been almost entirely due to the requirements of peace, namely, commercial expansion and the need for better means of transport, both in the past and in modern times, the development of the marine vessel has at all times been closely connected with the needs of warfare. Thus as far back as historical record carries us boats and ships of one kind and another have played a prominent part in warlike operations, and the attempts to obtain the highest possible speed either to pursue or to evade pursuit, have unquestionably had (and still have) an enormous influence on the march of progress. One has only to reflect on the educational value of the naval constructional work of to-day not only in the matter of craftsmanship, but also in materials and methods, to appreciate its importance and influence; it is difficult to even guess what the present position of shipbuilding would have been if, during the last few centuries, no need for naval construction had existed.

At the present time the question of naval and military necessity again promises to be of service as a powerful incentive to progress in that most recent and fascinating development of automobile engineering—the aeronautical machine. We are in this latest development privileged to witness the beginnings of a new mode of locomotion, and to some degree perhaps we have been able to experience in watching the earliest flight something of the feelings of primitive man who first saw a wheel employed as an aid to transport, or first used a canoe as a means of locomotion.

At present the particular commercial or practical uses that will be made of the flying machine are in the main a matter of conjecture, but we do know almost as a certainty that it will before long be looked upon as a necessity in the conduct of military operations, and will be extensively utilised as an auxiliary both in the army and navy. It is probable that the experience gained by its employment in the services will, in the future, be one of the most important factors in its development, and will enable invaluable data to be obtained as to its applicability to other uses. It is well to bear in mind the fact that up till the present the flying machine, although familiar to us by reason of the many competitions that have been held, has so far not one single act of commercial utility to its credit.

The importance of the military outlook for aeronautical engineering is one of its most promising features. Every new development or enterprise requires heavily financing in its initial stages of development; and it is impossible to create or establish a new industry unless there is ample prospect of a sufficient margin of profit to pay for experiments and failures. The modern automobile movement has so far been singularly fortunate in this respect; not only does the motor car effect, mile for mile, an enormous saving to those who use it when compared to the horse traction that it has replaced, but the other advantages, namely, its higher speed, greater distance capacity, and convenience, are so overwhelming, that it is without a competitor, and it is able to regulate its own price. In spite of these facts, until recent years the automobile industry in this country, except in a few isolated cases, has not proved conspicuously remunerative.

Up to the present the aeronautical movement has been financed by private generosity in the form of monetary prizes, and by the losses made by the promoters of aviation meetings, but this cannot go on indefinitely. These meetings and competitions have resulted in some improvement in the machines, and considerable progress in the *art* of flight, and so far the risks have been to some extent justified, but there is a great deal of useless repetition of flying, and many incitements to fly in unsuitable weather and under dangerous conditions, and many lives have been wantonly sacrificed* The modern development of the aviation meeting tends to nothing more (from a sporting or scientific standpoint) than a big boy offering a small boy sixpence to venture on thin ice and endeavouring to collect ninepence from some of his friends for the privilege of looking on. Fortunately, experience shows that the ninepence is difficult to collect.

When the present phase of flying is over there will doubtless remain a certain number of men who are prepared to fly as a genuine matter of sport, and competitions in the form of long-distance flights will probably continue to be held to decide the possession of some trophy or the relative merits of different types of machine. Under these conditions with machines of sufficient speed and power, and with the provision of properly laid-out alighting grounds, the risks would be reduced to reasonable proportions. Beyond this we may be sanguine enough to believe that certain useful applications of flight will be discovered in connection with postal and similar service. All told, however, it is very doubtful whether for some time to come there will be enough real uses for aerial flight, *apart from its military applications*, to put it on a sound industrial footing, and it is certainly in this latter direction that we must look for the principal incentive to progress; already a

(*c*) The following quotation from Sheridan's "Rivals" points to the fact that a horse vehicle, even as late as the year 1775, could be quite as offensive as any modern motor car in the matter of dust-raising:—

"*Bob Acres*.— . . . Warm work on the roads, Jack! Odds whips and wheels! I've travelled like a comet, with a *tail of dust all the way as long as the Mall.*"

*The death of C. S. Rolls occurs as an instance.

great part of the interest and support given to aeronautics in European countries is due to the obvious potential future of mechanical flight from a military point of view. Thus we again hope to see the instincts of fighting and destruction turned to serve the purposes of civilisation and constructional advancement just as they did in the most primitive times in the history of the race.

Mechanical Development.

I will now pass on to the discussion of the development of the automobile vehicle from a less general standpoint, and deal with the mechanical side of the question.

The modern automobile or motor car as we know it to-day owes its existence to a number of independent factors; these may be summed up in the two words *materials* and *inventions*, and it is due to the enormous and rapid advance in our knowledge of the properties and handling of materials, some of them unknown a few decades back, that the development and perfecting of the motor car has taken place with such rapidity.

It is difficult to find even so much as one discovery either in material or invention that is definitely essential to the construction of a successful vehicle, yet we know how much we are indebted to such materials as vulcanised rubber; aluminium and its alloys; nickel, chrome-nickel, and other high-class steels; petroleum, both as fuel and lubricant; and discoveries and inventions such as pneumatic tyre, Otto cycle motor, induction coil, either as used in battery or magneto ignition, ball bearings, and a multitude of less important items. Also in the factory it is difficult to over-rate the importance of modern gear cutting and generating machines, grinders of precision, automatic and semi-automatic machine tools, high-speed cutting steel, etc., etc., all necessities of a modern automobile factory. To all intents and purposes, 60 years ago *none* of the above were available, and even 30 years back, that is to say, but 15 years before the inception of the industry in this country, nearly all the main essentials, such as the pneumatic tyre, the petrol motor (*d*), ball bearings, etc., were still unknown, mineral oil for lubrication was quite a novelty, and petroleum spirit was to be obtained only in homœopathic quantities at the apothecary's shop (*e*).

It is a remarkable fact that the whole of the more essential component discoveries that contribute to the satisfactory and harmonious *ensemble* constituting the modern motor car should thus have come through to time like parts of a preconcerted scheme. In some cases we can doubtless trace intention; the magneto ignition apparatus, for instance, is (in its present commercial form) definitely the outcome of the automobile, but the Otto cycle motor and the pneumatic tyre were developed quite independently, with entirely different objects in view, and the pioneers of the aluminium industry (aluminium became an article of commerce in the early nineties) can have had no idea that they were preparing to produce a metal for the express use of the automobile engineer! When these three factors entered into alliance in the interests of the automobile, they had never seen each other before.

It has already been pointed out how the pleasure car, which at present constitutes the bulk of the automobile output, is free from external competition, due to the overwhelming advantages possessed by motor propulsion. There is competition enough within the trade, but otherwise than under exceptional conditions the horse is quite out of court (*f*).

One result of this is that the prices of materials may rise and fall within wide limits without the manufacturer being made to suffer. When, however, we consider the case of the commercial vehicle, the position is different.

I have, on more than one occasion, been asked to advise carrying companies and such like, on the question of a wholesale substitution of motor vehicles for horse traction; at first I went into the question with enthusiasm, but I soon discovered what a difficult proposition confronted us, and (this is some years ago) I had reluctantly to admit that no money could be saved, and so advised my client to continue with horses. To-day, even, it is only in certain cases that the change from horse to motor traction can be made with advantage.

Thus the future of the commercial vehicle is largely bound up in the question of cost of materials, *especially* fuel. A few examples of the fluctuations of the last few decades may be of interest:—

Aluminium shows a fairly steady descent—

	s.	d.	
1892	4	0	lb.
1894	2	0	lb.
1896	1	5	lb.
1900	1	4	lb.
1904	1	2	lb.
1908	0	9	lb.
To-day ...	0	7	lb.

Caoutchouc (Para) has shown accelerating ascent—

	s.	d.	
1900	4	2	lb.
1902	4	2	lb.
1904	5	2	lb.
1906	5	3	lb.
1908	5	3	lb.
1909	9	0	lb.
To-day ...	12	5	lb.

Petrol, rapid and irregular fluctuation—

1896	No regular market.
1900	10½d. gall. (wholesale by contract).
1902	8d. gall.
1904	6d. to 11d. gall.
1906	11d. to 14d. gall.
1908	9d. to 12d. gall.
To-day ...	9d. gall. excluding tax.

Lubricating Oil.—During the period 1900 to 1910 the variation that has taken place in the cost of materials is the equivalent of approximately 30 per cent. to 40 per cent. fluctuation on the finished product.

Of the above the future of the commercial vehicle is affected more especially by the cost of petrol. Unfortunately, this commodity has fallen into the clutches of the Chancellor of the Exchequer, a fact that is not only a direct set back, but also is like a sword of Damocles to the manufacturer of the heavy vehicle, the present rate of taxation may only prove the thin end of the wedge. There is no reason why heavier petroleum should not be utilised for motor lorries and goods tractors, and a very necessary saving could be effected.

The best results at present achieved by the commercial vehicle, as represented by the goods lorry, work out at approximately 60 ton miles per gallon. The Renard road train has recently made a showing of 45.5 ton miles per gall. on ordinary roads. The current figure according to the best modern practice for the motor omnibus (fully loaded) is, roughly, from 40 to 50 ton miles per gall. It is not to be anticipated that any great advance will be made on these figures by existing methods.

In brief, the future of the commercial vehicle depends upon the saving of money and on nothing else, economy of first cost and economy of operation and upkeep, and after everything has been done that the designer and constructor can accomplish, the rest will *depend upon the road on which the machine has to operate.* It is no exaggeration to say that the best motor lorry ever built could be made a losing proposition by being used over some of our main roads of a few years ago, and that many a machine that has been deemed a complete failure could have given a good account of itself if scientifically made roads had been available, and in addition the roads themselves would have proved more economical. If the above be true under existing conditions, once let the designer *know* that he will have good roads on which to operate, and the conditions will be still more favourable.

I do not think that sufficient attention has been called to the need for *uniformity* in the character of the road surface. If a vehicle has to be constructed to operate for 20 per cent. of its time on badly-built roads or roads in a bad state of repair, it has to be designed suitably, and it is unable to take due advantage of a really well-made road when the opportunity occurs. In a sense this statement requires explanation; I will give an illustration. It is notorious that the main roads in France are some of the best in the world, straight, and of good surface. At the same time the roads in the villages and smaller towns are abominable, badly laid and worn-out *pavé* being plentiful. A French expert driver, reporting on three pleasure cars, which we will call A, B, and C, made comments as tabulated, the *spring periods* in each case being as given in column 1. Now we know that the criterion of the hardness of a suspension is the *spring period* (*g*), and the accuracy of the observations may be gauged by the complete agreement shown; these observations also, however, show how impossible it is to suit both kinds of road with

(*d*) The "Otto" gas engine was then in its infancy.

(*e*) In 1896 one had to obtain *benzoline* from a drug stores or oilman; frequently I have had to visit four or five shops to obtain enough to "fill up," and that with a tank of but five gallons capacity.

(*f*) We always find in every great industry on a sound industrial footing that a saving of £ *s. d.* is at the bottom of things. Thus to-day any moderate-powered car—say, not exceeding 38 R.A.C.—*can* be run at less than 6d. per mile, and has a seating capacity of four or five, or even more, persons. Against this the cost of running the barest equipment in the way of horse vehicle —say, a dogcart—will cost a suburban doctor from 10d. to 1s. per mile. It may be objected that the cost of running a car with chauffeur and garage charges, washing, etc., comes to more than 6d. per mile. This is true; but, on the other hand, keeping a pair-horse brougham often costs 2s. 6d. to 5s. per mile, and in some cases twice this amount.

(*g*) Proc. Inst. A.E., Vol. II. p. 192.

one and the same car, and hence the importance of uniformity in the roads of the country. In the case of the commercial vehicle the matter largely resolves itself into the question of carrying an unnecessary weight of spring and axle steel and a low *load/tare* efficiency over the whole of a journey for the sake of one bad stretch of road—the resulting want of economy emphasises the point.

Type.	Period. Vib. per minute.	Remarks.
A	72	Springs rather too easy on main roads, but very good in towns.
B	88	Far too "hard" on town *pavé*, but very good on main roads.
C	82	A good compromise, but still somewhat "hard."

I may add that further experiments showed a periodicity of 78 vibrations per minute to afford the best compromise.

The Future of Heavy Cars.

Questions relating to the future of the heavy vehicle have been discussed at some length owing to the fact that it is to be anticipated that the next big expansion of the automobile industry will be in that direction; up till recently the manufacturers who had taken up the commercial vehicle have been able to make but little progress, and those who tackled the subject ten years ago have on the whole lost heavily. Looking backward, we can see the reason, mechanical difficulties quite as great as involved in the production of the pleasure vehicle, and the irksome condition of £ s. d. always at hand; no money to play with and no margin to pay for failures, added to that roads that in many cases rendered the proposition impossible.

The men of means who in the early days purchased and ran cars when the expenses of so doing were not only heavy, but an entirely unknown quantity, have done the country a service that has never been adequately acknowledged; they have provided the sinews of war whereby the position of the industry has been established, and its future on a still broader basis is assured. Each one of them have been worth to the country at least a couple of dozen of the average philanthropist.

The 15 years of automobile development that have elapsed since the first vehicle of the modern period made its appearance in this country has resulted in the definite establishment of one general plan of construction that to-day is adopted by practically every maker of repute; this is the type founded on experience mainly of the type of machine—the passenger or pleasure vehicle. Different manufacturers adopt, so to speak, their own individual interpretations, and wide divergence is to be found in the matter of detail; still the main underlying scheme is, we may say, standardised. There is a sprung body containing the motor at its foremost portion with the so-called radiator in immediate proximity; there is the gear-box immediately abaft the motor with intervening clutch; the axes of rotation of both motor and gear shafts being parallel to the direction of motion; there is a universally jointed shaft passing to the underframe or axle frame; there is the right angle drive, worm or bevel; and finally there is the differential gear and axles on the right and left hand driving the respective road wheels. The steering gear, too, is part for part functionally identical on almost every make of vehicle.

At present we do not know whether this standard type is as suitable for other purposes as for that for which it has been developed. The tendency at the moment is to assume that it is so suited, and so long as the components, *i.e.*, motor, gear-box, etc., remain as at present parts of the whole combination, the assumption seems fair; if, however, any radical change is found to be desirable in the *mechanical system* it would not be astonishing if an equally radical change were found to be required in the general scheme of arrangement.

When we turn to discuss the relative merits of the established mechanical system and the various possible alternatives, we must not lose sight of one important fact—the rough and ready nature of public opinion in engineering matters. It is conceivable that, from a strictly utilitarian point of view, several different types of vehicle would be better suited to the varied conditions to be met, than one stereotyped article, and yet the public might insist on buying the one type with which they had become familiar. From broad considerations of policy it is quite likely that the public are right; it may within limits pay to secure uniformity of practice at the expense of the slight advantages to be gained by specialisation. We have this distinctly conservative influence to take account of when any question of change is under discussion.

From the purely technical point of view the permanence and extension of the present practice in construction must rest on the intrinsic merits of the various components of the existing mechanical combination. Let us examine it part by part.

The petrol motor has made a very great advance since its first introduction—in fact, the modern four and six-cylinder engines now fitted to high-class cars have reached a degree of perfection that would never have been deemed possible ten or twelve years ago. The weight per h.p. has come down from about 30 lbs. to in some cases less than 10 lbs. We may take it that the best practice of to-day ranges from 9 to 12 lbs. per b.h.p. (*h*) (sustained load). The vibration is trivial compared to what it used to be in the days of the two-cylinder engine; in this respect the six-cylinder has some advantage, but in certain cases there is still trouble from the synchronous vibrations to which this type is liable. Another remarkable advance that has been made is in the direction of silence; it is true that much of the silence of a modern petrol motor is due to the elimination of the vibration due either to inherent want of balance or to want of stiffness, still the advance that has been made is not entirely explainable on these grounds, and the superiority of the sliding (sleeve) valve is in this respect too well known to require comment.

On the whole we have no reason to have the least misgivings as to the permanence of the petrol motor, much as we already know it as an element of the combination. We may anticipate further reduction in weight (the aeronautical motor leads the way at present at between 3 and 4 lbs. per b.h.p.), but it is not capable of long-sustained effort at this rating. We may anticipate other improvements in further reduction of vibration, and possibly the fuller development of refinements, such as self-starting, but no revolutionary change is in sight.

In the rear axle department, the right-angle transmission and balance gear, etc., there seems but little room for further improvement; the worm will almost inevitably oust the bevel on the score of silence; the efficiencies are approximately equal, but again there appears to be no demand for any drastic change. Perhaps on powerful vehicles of the heavier type there is an opening for separate driving units to abolish the differential; this is already being done in one at least notable instance. However, this involves a duplication rather than essentially a departure.

The change gear-box is evidently the weakest point in the chain of mechanism. In the gear-box (of the sliding gear type) we have a piece of mechanism that we should most of us like to see abolished; it is notoriously a makeshift appliance, and its survival to-day is in main to be accounted for by the fact that the h.p. commonly used in pleasure vehicles has been increased to such an extent that the direct drive is in operation for 99 per cent. of the distance run, and for the remainder we put up with the makeshift and make the best of it. After all, when one has become accustomed to sliding gears they are not such an abomination as would be thought by the uninitiated. It may occur to many that a gear box represents many parts and much money for a mechanism in use for so short an aggregate time, but when its services are wanted they are wanted badly. Moreover, such a standard of criticism will not bear very close investigation. For instance, in the course of a recent conversation with a well-known authority on the manufacture of fire-arms, I learnt that the aggregate working life of a rifle—the actual total time it was undergoing wear and tear—amounted to no more than four seconds.

If there is any portion of the accepted scheme that is likely to upset the present disposition of the functional components of the modern car, either as affecting the pleasure car, or the type of commercial vehicle of the future, that portion is without doubt the change gear-box. In other words, if a new type is to be evolved superior to that which now reigns supreme, the key to the position is in the abolition or remodelling of the change-speed gear.

Let us look around for possible alternatives. Firstly, we have the epicyclic type of gear, such as adopted in the Lanchester car; this is mechanically superior to the sliding gears, but it is costly to manufacture, and therefore will only appeal to a limited market. In any case, its adoption does not lead to any disturbance of the *ensemble*, but merely the substitution of one type of gear-box for another.

The most promising alternatives to mechanical change gear up till the present proposed or utilised are: (a) hydraulic transmission, (b) electrical transmission, and (c) electrical auxiliary. When either of the first two are employed they may either be arranged as direct substitutes for the change gear-box, or they may be made to operate direct from motor to road wheels, so doing away with the right-angle drive (bevel or worm), and even the differential. Both arrangements possess advantages and corresponding disadvantages. When the complete transmission is

(*h*) Without fly-wheel. This is the best basis of comparison; the amount of fly-wheel the designer may fit is so largely a matter of taste.

effected by a single step, either hydraulic or electric, there is the saving of two to four mechanical transmissions (depending upon whether the direct drive or a gear would have been in use), to set off against the hydraulic or electrical losses. On the other hand, when the gear-box alone is replaced, the loss of the bevel or worm gear still remains (amounting to, roughly, 10 per cent. of the power transmitted), but it is possible to utilise a form of hydraulic transmission in which the difference of torque between the motor and arbor shafts alone is taxed, such gear being strictly the analogue of an epicyclic gear of variable ratio. Such a gear has been proposed many times, and has been used experimentally more than once. The Hall hydraulic gear is a typical example, and was, I believe, the first of its kind. More recently it has been proposed to use an electrical transmission designed on the same principle, and though experiments in this direction have been carried out, I have not heard of any successful results.

On the question of efficiency some divergence of opinion exists. The best results obtainable with a double hydraulic transmission, *i.e.*, overall efficiency pump and motor, are in most cases under rather than over 70 per cent., though recently it has been *reported* that tests of a rotary pump and motor of novel form have shown as much as 80 per cent. On the basis of direct transmission from motor to road wheels this latter would give the hydraulic transmission as about 10 per cent. inferior to the direct drive, but about the same as the present efficiency when driving through the gears; such a result if confirmed by later tests would be most hopeful. On the 10 per cent. basis the inferiority of the hydraulic gear would vary from 10 per cent. to 20 per cent., according to whether compared to the gear drive or direct.

We could scarcely hope to obtain such good efficiency from an electrical transmission, for owing to considerations of weight the results obtainable in stationary work do not apply. It is improbable that it is "commercial" to use dynamo and motor of more than 55 per cent. or 60 per cent. overall efficiency, hence it is improbable that this type of gear can ever come into use unless in exceptional cases. Beyond this it is in the majority of cases not possible to drive direct on to the road wheel axles by electrical means, so that one gear transmission must be retained, and the efficiency will be reduced accordingly.

The epicyclic analogue possesses certain attractions. Again, the electric transmission will, compared to the hydraulic, be at a disadvantage. It is possible that some reconsideration of this form of hydraulic drive may result in something of real commercial utility. On the equivalent of the direct drive the hydraulic loss is zero, and (at 80 per cent. for the double transmission) the efficiency would be superior to the ordinary gear-box transmission over a comparatively wide range.

We will now pass to the consideration of the third alternative—*electric auxiliary*. The use of electrical power as an auxiliary means of propulsion opens up a new field of possibility. We have no longer to deal merely with a substitute for change gearing, but with a combination possessing certain valuable properties of its own that are clearly destined to play an important part in the future development of the automobile.

In the simplest and most straightforward application of the petrol electric combination a direct current dynamo is mounted on the engine bed plate, and the engine and dynamo shafts are keyed or bolted together, so that they revolve as one. The dynamo is of the shunt wound type, and acts alternatively as dynamo or motor; it has become customary to apply to it the name *dynamotor*, a term which it is convenient to adopt.

The petrol-electric installation so constituted is arranged to operate through the medium of a magnetic clutch either on the usual differential, or may be fitted in duplicate and arranged to act by duplicate worm or bevel drives direct on to the road wheels. A storage battery is provided usually of about 5 per cent. of the total weight of the vehicle, which takes up any excess power when going down hill or on a level, and which gives up the energy so stored when required to ascend a gradient. The alternative function of the dynamotor is determined automatically by the speed variation: when the petrol engine is slowed down by excessive resistance the voltage in the armature falls, and the battery begins to assist to drive the vehicle; when on the contrary the engine attempts to race away, the voltage rises as a consequence, and current flows in the opposite direction, and spends itself in the charging of the battery. Thus an approximately uniform speed is maintained over the whole route travelled; the vehicle takes but little notice of the ups and downs encountered. Likewise, when stopping part of the energy is thrown into the battery, and is drawn out again when getting under way.

It will be appreciated from the detailed discussion of the subject that the future of the petrol-electric auxiliary system more widely than at present in contemplation must rest largely with the question of accumulator weight, not the weight per h.p. hour, that is the energy content, but rather the h.p. output, *i.e.*, the *rate of doing work*; at present the best results commercially attainable for brief discharges of but a few minutes duration is the equivalent of one h.p per 40 lbs. weight. This is reasonably satisfactory for vehicles of moderate speed—less than 20 m.p.h.—but it is not good enough for the pleasure vehicle. If the battery weight could be reduced to 20 lbs. per h.p. the latter as a proposition would be worthy of serious consideration.

In conclusion, we may say that as in the past we have seen a gradual unification of type in the construction and design of automobile vehicles, even though destined for widely different purposes, so in the future we may anticipate a gradual differentiation of types for specific purposes, each embodying modifications adapted to its own particular needs. Any change in this direction must come about slowly. The buying public are a conservative body, and accept every new development with caution. Up till now this caution has been of great benefit in establishing *one* really satisfactory type of vehicle, with such variations as exist confined strictly within certain well understood limits, in place of a wide variety of types of indifferent performance. This stage of achievement is now, however, completed, and the public have in the process become sufficiently educated to appreciate a departure, even if of radical import, at its true value, and so in prospect there is to-day a wider field opening to progress and development than has existed in the past history of the automobile movement.

Editorial Note.

At the end of the address Mr. Lanchester explained various points in connection with the latest embodiment of the petrol electric system with which his name is now identified, and also exhibited a diagram showing how it was that the rate of doing work by the accumulators, rather than their h.p. hour output, was the main factor to be reckoned with, as far as the accumulators were concerned. As various formalities had to be complied with in connection with the introduction of the new president, Mr. F. W. Lanchester, and the retirement of his predecessor, Dr. Hele Shaw, there was practically no time left for serious discussion at the end of this extremely able and interesting presidential address.

Perhaps the most broadly momentous suggestion offered in this address was that the point may have been reached when the industry, after having elaborated along general lines, might find it desirable to consider more particularly specific requirements. From many points of view one cannot but be grateful for this hinted advocacy (for as such, rightly or wrongly, the writer interprets it) of more individualistic treatment of motor design. Coming, as it does, from such a source, this hint appears to us particularly opportune at the present juncture, for there is danger in delaying such a policy. The cars of the present day are constructed along such generally similar lines that, though the public at large may by now be sufficiently educated to appreciate roughly the advantages of an unusual mechanical combination, if lucidly explained to them, such appreciation is liable to be obscured by their preconceived notions of conventional car design. The more conventional car design becomes, and the longer it remains conventional, the more difficult will it be to cope with this conservative tendency in the public character, which was fully recognised by Mr. Lanchester. Further, granted that a radical alteration, that is really intrinsically good, is in theory bound to be adopted on its own merits, the task of educating the public up to it is likely to become increasingly difficult and expensive, the more the public become accustomed to one particular type of mechanism.

We do not suggest that startling innovations merely in themselves are desirable as a means of educating the public, or that any radical improvement involving sweeping changes in mechanism arrangement are imminent—though we shall probably see these in the fullness of time—but we do maintain that the time has come to consider whether certain specific requirements, as, for example, those of the car for town work, are best served by the usual standard arrangement of mechanism, and if not, to make some attempts to accustom the public to less usual but more specially suitable designs.

Mr. Lanchester also brought forward another factor when he suggested the probable influence that the Road Board might exercise on the roads of this country in the future, but it is unlikely that the operations of this body will exert so much influence as the presidential address appears to suggest, at any rate for many years to come.

Though, as was pointed out, the Road Board at present have nearly £1,000,000 at their backs, the expenditure at present sunk in the roads of this country approximates to some £200,000,000, and bearing this fact in mind, the considerable sums available in the present and near future are not likely to make so much impression on the road surface as at first sight appears probable. In another very interesting interpolation added by Mr. Lanchester—and, consequently, not printed in his address—it was pointed out that in the days of the turnpike it was possible to know the earning capacity, and so the actual relative wear and tear demands made on each road or section of road. Since the abolition of the turnpike system, however, there had never been any systematised method of estimating the earning capacity of any given road, and consequently of estimating the expenditure that was justified on such a road, but Mr. Lanchester hinted that the advent of the Road Board might remedy such a state of affairs. If things are to be so, the present writer would like to submit that the probable tendency of such a system—at any rate, for many years to come—while leading towards an improvement of the main traffic arteries, would leave little more than enough to keep the by-roads up to their present standard of condition, and as by-roads, as well as town traffic, have to be reckoned with, road requirements for some time to come are not likely to narrow down sufficiently to allow of much specialised design.

Still, it must be admitted, or, at any rate, hoped, that the tendency towards more or less road uniformity has begun, and is a factor to be carefully studied in the future. For the present, as far as road surface requirements only are concerned, some may think that designers and inventors may tend towards providing adjustability for various road requirements. For instance, Mr. Lanchester quoted the case of a car tested over different road surfaces with springs of varying periodicity, showing that with ordinary systems of spring suspension at best only a compromise can be obtained. Now the production of a spring system that could be rapidly adjusted to passing road requirements is by no means an impossibility—in fact, something very like it was brought forward about a year ago, and it is conceivable that this system of adjustability to temporary needs might be extended to other parts of the car structure. Already we have the engine control levers and the gear box as examples embodying this principle.

Besides the requirements of special work, road conditions, and public taste, the other really important factor external to the car is that of economic requirements—in other words, the influences that govern costs, both of production and working, and in this connection, though Mr. Lanchester touched on the subject of petrol taxation, one could have wished the time at his disposal had allowed him to deal further with the fuel question. True, the possibilities of petroleum were mentioned, but we should like to have seen some reference to other fuels as well; notably to producer gas, which is attracting a good deal of attention now in other engineering quarters, and which, in the writer's opinion, offers the cheapest and least easily-taxed source of power at present available. Hitherto but little has been done in the way of using producer gas on road vehicles, and considerable difficulties still remain to be overcome, but the more or less experimental work in this direction that has been carried out at Glasgow and in France have given quite sufficient promise for the future.

Turning now from the general considerations to the particular details with which Mr. Lanchester dealt, all who heard or read his address will agree with the author in what he had to say about the petrol engine (except perhaps on the highly controversial valve question), but here, too, many will join with us in wishing that more had been said, especially on the relation of stroke to bore, and the elaboration of a simple and comprehensive horse-power formula, now that the R.A.C. formula has outlived its usefulness.

Where Mr. Lanchester, however, states that there seems but little room for further improvement in the rear axle, the writer ventures to differ. One has only to study the imperfect differential action of the back axles on cabs to notice the defects of the present type of balance mechanism. How often does one see one road wheel slipping round furiously when the clutch is put in, or brake applied. What is wanted is a balance gear that will act by distance (or its consequence—speed) rather than by resistance. Such a mechanism may be impossible, but the word impossible is one for which the present-day engineer has no great liking, and in which he has but limited belief.

CORRESPONDENCE.

No responsibility is accepted by the Editor for the opinions of correspondents, and the right is reserved to omit any portion of a letter if it is considered desirable to do so. Letters to be inserted with a nom de plume must be accompanied by the name and address of the writer. If publication in the next issue is desired letters should reach the Editor in no case later than the 20th of the preceding month.

ARRANGEMENT OF HUB BEARINGS.

Sir,—The article of Mr. Pugh's in last month's issue on the above forms very interesting reading, and should direct attention to a weakness in motor car designs.

I do not know, however, why the writer thinks that the cup and cone type of bearing may some day become popular. I should have thought that he would have condemned it because it was not what might be called scientific in action owing to the spinning motion imparted to the balls, apart from any other consideration.

I believe the annular type of bearing was originally designed to overcome this, and may be roughly taken to do so, so long as the pressures to which they are subjected are acting in the same plane or parallel to that of the revolution of the bearings, but immediately a load is applied to the race due to the thrust this force tries to wedge the balls between the curved faces which are coming together under its influence. Also when the bearings are under the influence of thrust a spinning motion is imparted to the balls because the curves of the races coming together make up a form of bearing similar to the cup and cone type. This motion must be a source of rapid wear. It will be agreed that when two such curves as the usual annular bearings possess approach each other under the thrust loads, there is not only considerable bursting pressure, but the balls are subjected to a corresponding crushing load. Take, therefore, the case of an inside bearing of the outside wheel when a car is rounding a corner, it might be satisfactorily concluded that about one-fifth of the balls are subjected to the following loads:—

1. That due to the proportion of the weight of car it has to carry.
2. The enormous loads due to the moment about the tyre occasioned by the thrust.
3. The thrust pressures and consequent crushing and bursting stresses.

It may be worth while pointing out that the balls have a spinning motion under these conditions.

Mr. Pugh's article has been very instructive, and it taught me that the present form of annular ball bearing by itself does not meet the needs of an automobile hub. At any rate, the opinion I have formed is that a thrust bearing is desirable to prevent the balls from being wedged or having a spinning motion given them whilst under such heavy loads.

GEO. H. RODWAY.

Sir,—Will you allow me to draw attention to what I consider to be an error in the latter part of Mr. Pugh's excellent article on the "Arrangement of Hub Bearings" in the October "Automobile Engineer." I am referring to his advocacy of a single journal bearing in back wheel hubs. With this arrangement the function of holding the back wheel vertical, against the tendency of side thrusts to deflect it from the vertical, devolves upon the driving shaft. This shaft is therefore subjected to a bending moment, the magnitude of which is the load on the bearing, near the centre of the axle, multiplied by the distance of this bearing, from the one in the wheel hub.

Taking the top row of figures, given in the table referring to Fig. VI., this bending moment is:—

$$1{,}312 \times \frac{480}{25 \cdot 4} \text{(inch lbs.)} = 24\,800 \text{ (inch lbs.)}$$

If the shaft be $1\frac{1}{2}$in. diameter, the stress due to this bending moment alone is:—

$$\frac{32 \times 24{,}800}{\pi \times (1\frac{1}{2})^3 \times 2240} \text{ tons per sq. in.} \left[\text{from the formula } f = \frac{32\ M}{\pi\ d^3}\right]$$

This gives a stress of 33.5 tons per square inch. With such a stress the shaft would bend immediately. Of course, the forces assumed to be acting are excessive, and will be met with only in exceptional circumstances, but it seems to me that it is extremely desirable to protect the driving shafts from forces due to side pressure on the wheels, the magnitude of such side pressure often being quite indeterminate, as when a wheel runs against a curb or other obstacle.

I know of no better way of resisting these side pressures than to fit in the hub two large journal bearings, set well apart and mounted on the axle casing.

J. H. SINCLAIR.

Sir,—It is to be regretted that Mr. J. V. Pugh in his most interesting article did not arrive at some more definite conclusions. He commences with an expression of doubt, and leaves us with that feeling. Mr. Pugh's strongly pronounced leaning towards the cup and cone adjustable bicycle type bearing is possibly on account of its adjustability to take up wear, and the simple way in which all looseness could be removed. I must, however, beg leave to contradict him when he says: "and so the cup and cone bearing failed because no proper provision for the exclusion of water was made."

The sole cause of failure was because the general dimensions and the point contact of the balls were too small to carry the loads. Water had nothing whatever to do with the failure (although we all know its serious effects cannot be over-

rated where it can obtain entrance). Certain cars had these cup and cone bearings in their bevel pinion and differential back axles break up, which on examination showed ample lubrication, and entire absence of any evidence of water (because no water could possibly enter this portion). The front and rear wheel hub bearings also broke up; and here it might be said that water could enter, but I never found traces of it, or any evidence of its influence. The radial load carrying capacity of the cup and cone bearing is much inferior to the present non-adjustable journal bearing, consequently, its dimensions would have to be increased beyond the present non-adjustable bearing, and many of these are even now too small to do their duty. These almost impossible modifications would still only leave us with the points of three balls to carry our load, and to resist thrust wear, and it is upon the question arising out of these three load carrying contact points to which much of our consideration should have been directed.

Hereunder I therefore propose to discuss :—

(A) The adjustable cup and cone bearing with flat surfaces.

(B) The same bearing with curved surfaces.

(C) The present non-adjustable ball bearing of the annular type.

(D) An adjustable taper roller bearing.

There is ample evidence to show that in consequence of the bicycle ball bearing having given satisfactory results on the cycle, and possessing such a simple method of adjustment, against side thrust, looseness, and wear, was hastily adopted by many continental and other automobile manufacturers, without sufficient preliminary tests or serious calculations. Those who had experience with these bearing failures, soon discovered that the dimensions of the cones were only just big enough to admit the axles; ball dimensions were sometimes increased from bicycle sizes of ¼ in. or 5/16 in., to ⅜ in., but the general external dimensions of the bearing were primarily controlled by the diameter of the axle, the idea being to keep everything as small as possible. I do not think that this bearing is ever likely to be reintroduced, because it is inferior for sustaining radial loads to the present non-adjustable ball bearing (see load calculation No. 3), and, therefore, all dimensions would have to be materially increased beyond the present type, to make the cup and cone bearing of equal wearing value, and even then, the inadequate wearing contact of the points of three balls still remain.

The failure of the cup and cone bearing for automobile work was undoubtedly the cause of

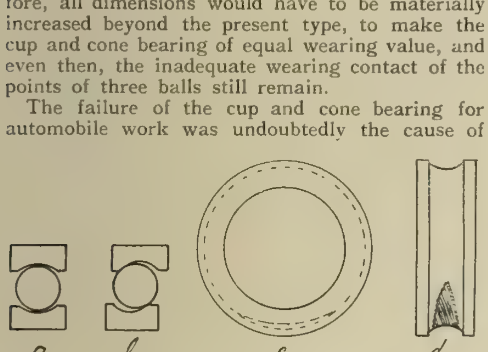


Fig. I.

the general adoption of the non-adjustable journal ball bearing, originally developed for other engineering work. I have made a collection (from different parts of the country) of samples of this type of non-adjustable ball bearing, of various makes, taken from cars of British and foreign manufacture, and in most cases I find that the bearings have not been replaced by new ones because the old ones have (broadly speaking) worn out, but because they have worn loose, and as no adjustment is provided that will take up this looseness, comparatively new bearings have had to be scrapped.

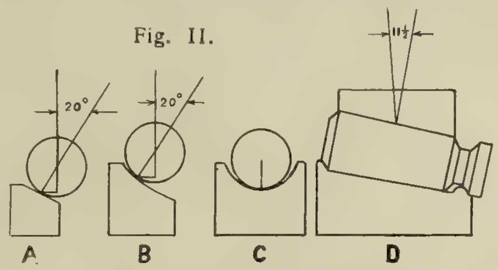
In the majority of these cases, it is difficult to imagine how the car owners could have tolerated the excessive noise and rattle that must have developed through this looseness.

Fig. I.a shows a bearing in its new condition, when it is (so to speak) in perfect adjustment. The radius of the bearing surface is struck from a centre taken at 7/10 of the ball diameter, this being common practice. After the bearings have worn loose, they are in the condition shown in Fig. I.b, and in all bearings taken out of back axles, or any place where the outer ball race ring is fixed, the inner, or central ball race, being the live or revolving portion, appreciable signs of wear are more rare. There is only a general looseness, as explained in sketch B, permitting the central ball race to be freely moved about in the directions indicated. But in front wheel bearings, where the reverse condition applies, viz., the outer ball race revolving, and the central ball race being fixed upon the axle, wear is very appreciable in one portion only, namely, on the under side of the stationary ball race, as indicated in Fig. I.c.

This wear commences, and leaves off, within about 25% of the total circumference, three-fourths of the circumference showing no signs of wear or reduction of diameters; the under side, in nearly all cases being worn until the balls convert the form of the wearing surface from a point contact into a burnished, semicircular groove, closely approximating to the circumference of the ball. The axial or side thrust load is indicated on one side of the ball race upon the remainder of the circumference.

The capacity of this type of bearing for sustaining side thrust loads being so small explains why looseness quickly develops, and when once set up in the road wheel bearings, must go a long way towards explaining the development of noise and rattle, as these bearings constitute the rotating links between the pneumatic tyres and

Fig. II.



the total load, and the least looseness in here must give rise to mechanical shocks between the loose opposing parts.

This weakness under side thrust loads raises the important question of accidental stresses, which are almost entirely confined to axial or side thrust strains, and here the individual driver is more or less responsible in several ways; he may run his front or rear wheels against the kerb at an angle. This is no remote contingency, neither is it an accident that seldom occurs, there are few drivers who are not forced by accident or circumstances to rub tyres with the kerb and scarcely notice the occurrence. Side stresses thrown upon the bearings by 4,000 lbs. in motion —however slight the movement—by a wedge-like action, through the angle of contact between the tyre and kerbstone, are sufficient to permanently destroy the very small amount of resisting power offered by the present non-adjustable bearings, the cutting of a corner by glancing off the kerb, or banking, and the result of slowing up a side slip is also productive of unknown side load pressures. Any of these accidental forces may be realized within a few days of the car going into service, and surely some provision against such common occurrences of this character should be made, by providing a bearing capable of resisting such severe side thrust loads. You may increase dimensions, but you cannot increase the number of points of bearing, and whether it is a ball that is carrying the load, or whether it is a roller, only three of them can be regarded as load carrying factors. We, therefore, reach the limit of a ball bearing when we arrive at its point to point contact load carrying capacity. That this limit has been exceeded is quite clear from the proofs I have gathered from different parts of the country, as indicated in my sketch, wherein the balls, having worn away their original point to point contact surface by creating a reflex of their own shape, bring about the rapid wear of themselves as well as the bearing surfaces upon which they revolve. Obviously, the enormous advantage in the side thrust and radial load capacities of the taper roller is what we are looking for, and we find that while the non-adjustable ball bearing will sustain 217 lbs. side thrust load, the taper adjustable roller bearing is capable of sustaining 8,050 lbs. (See table 6.)

The question of adjustability is also a vital one; manufacturers go to great trouble and expense in making provision for taking up wear in almost every other part of the car; therefore, it is most remarkable that in a position like the road wheel bearings, where great compound wear and the severest conditions have to be met, no adjustability is provided. In fact, the question of adjustability seems to be almost ignored, and a bearing, which is freely admitted as incapable of resisting side thurst loads, is often adopted. The following comparative calculations have been taken out on the basis of journal loads, and simple axial loads, and in this condition can be easily understood. I am indebted to Mr. Alden for these calculations, and to several motor repairers for specimen bearings.

In Fig. II. the following designations are adopted :—

a. The bicycle type of adjustable ball bearing with flat cone surfaces.

b. The bicycle type of adjustable ball bearing with curved surfaces, with radius of curvature .700 of ball diameter.

c. The annular type non-adjustable ball bearing where the radius of curvature also equals .700 of ball diameter.

d. The adjustable taper roller with length equal to twice the mean diameter.

Let it be assumed that the diameter of the ball or roller is one inch, and that the diameter of the cone is five inches. The strength of the bearing, from the point of view of the load it can sustain, must depend upon the area of contact between the loaded surfaces. In cases a, b and c, this area (normal to the line of pressure) will be an elipse, while in case d, it will be a rectangle. The areas are as follows :—

TABLE No. 1.

	Area.	Per cent.
(a)	.00634 ...	100
(b)	.01108 ...	175
(c)	.01108 ...	175
(d)	.17000 ...	2,680

and provide a rough measure of the carrying capacity of the different bearings. However, since the line of contact is only coincident with the line of radial pressure in case c, it is obvious that the radial load capacities will bear somewhat different proportions to the figures given above. If it is assumed that the load carrying capacity of bearing a, with an area of contact of .00634 square inches is represented by 100, then the vertical loads possible are :—

TABLE No. 2.

	Load lbs.	Per cent.
(a)	94 ...	100
(b)	164.5 ...	175
(c)	175 ...	181
(d)	2,630 ...	2,800

If it be also assumed that the bearings contain fifteen balls or rollers, as it is well known that only a fifth of these are effective against radial load, then the radial load capacity of the bearings would be three times that given above, or :—

TABLE No. 3.

	lbs.
(a)	282
(b)	493.5
(c)	535
(d)	7,890

As regards thrust load it may be assumed that the line of average pressure will be midway between the edges of the path of contact and the vertical, though it is probable that it is about one-third of the way from the vertical, and not one-half. On this assumption the same normal pressure that was assumed in the previous tables will be produced by the following axial loads :—

TABLE No. 4.

	Load lbs.	Per cent.
(a)	34.2 ...	100
(b)	59.8 ...	175
(c)	14.5 ...	42.4
(d)	536 ...	1,570

To obtain the total thrust capacity of these bearings it is necessary to multiply these figures by fifteen, giving :—

TABLE No. 5.

	Load lbs.	Per cent.
(a)	513 ...	100
(b)	897 ...	175
(c)	217 ...	42.4
(d)	8,050 ...	1,570

Thus the relative values of the different types for both radial and axial loads compare as follows :—

TABLE No. 6.

	Radial load	Thrust load.	Thrust load in terms of radial load per cent.
(a) ...	282 ...	513 ...	176
(b) ...	493.5 ...	897 ...	182
(c) ...	525 ...	217 ...	41.4
(d) ...	7,890 ...	8,050 ...	102

These figures disclose some interesting points. One of the principal being that the old-fashioned ball bearing (a) and (b) is better adapted for end thrust than for radial load; and that the annular type ball bearing (c) is only about 40 per cent. as good. But we find that the adjustable taper roller bearing (d) is capable of withstanding approximately the same load in both directions.

R. F. HALL.

AUTOMATIC ENGINE STARTING.

Sir,—I should like to assure Mr. Schofield that I in no way misunderstood his remarks. I think his criticism in every way most pertinent to the particular system of starting it refers to, and in my reply I endeavoured to show that to have fully elaborated on the question of the various methods of field magnet winding and permanent magnet would have necessitated, at least, an article dealing with this particular system of starting separately instead of an article dealing with starters generally.

In Mr. Schofield's letter in your last issue, he implies that he means his statements to be taken in their strictly literal sense, and if this is so I am afraid I must differ with him so far, at any rate, as the precision of his statement is concerned. He says, "a dynamo for accumulator charging is always shunt wound," and immediately after he admits that it may be series wound in addition. But, if in addition to the shunt winding a machine has a series winding, it is a compound wound machine, and it would be as descriptively incorrect to call such a machine a shunt wound one as to describe grey as white, or black. Again, a permanent magnet machine may be used for accumulator charging. It is not shunt wound.

The points raised in the last paragraph of Mr. Schofield's letter would promote an interesting discussion at a meeting of electricians. As regards the bulk of a permanent magnet machine. It does not follow that this need be greater than that of an ordinary magneto, because it does not follow that its output need be great enough to turn the engine. There are other ways of starting an engine besides turning it, and a small electric motor might be used to bring about starting conditions.

The "natural wastage" of modern permanent magnets is very much less than that of magnets made a few years ago.

The effect of armature reaction on the life of the magnets *is* a very interesting study, and in the main it may be taken that the demagnetising effect of the armature is largely counteracted by the magnetising effect of the eddy currents set up. As an example of this effect—and at the same time a valuable tip to those concerned with magnetos—take a magnet which has lost its magnetism to some extent, apply it to a mass of iron and for a sufficiently long period—five or ten minutes—move it towards and away from this iron. It will be found that the magnet recovers its magnetism.

I would not for a moment suggest that in such systems of train lighting as the Stone, or others, the permanent magnet type of dynamo should be employed, but I am open to conviction that, for the specific purpose of charging batteries used for motor car lighting or ignition, the self-excited type has an advantages over the simple permanent magnet machine.

J. DALRYMPLE BELL.

MOTOR-CYCLES.

Sir,—I was very pleased to see "D.W.G.'s" suggestion in this month's *Automobile Engineer re* including motor-cycle construction in your excellent publication. There is no doubt but that the motor-cycle is capable of great development; therefore, I hope this suggestion will be carried out. I (also many others, I trust) would like to see some space devoted to the design and construction of aeroplanes, including helicoptenes, etc. In conclusion, I can speak from experience as regards some parts of your article, "Specialised Education for Automobile Engineers." I refer to apprentices working from early morning to night in the works, and then attending technical classes in the evening. I did this for some years. Result—Nervous breakdown. Technical knowledge is a very useful thing, but it can be got at too high a price.—Wishing you all success.

ALPHA.

ON TESTING CARS WITH AN ACCELEROMETER.

By H. E. Wimperis, M.A.

The paper read at the Sheffield meeting of the British Association in September last, "On the Use of an Accelerometer in the Measurement of Road Resistance and Horse-power," gave details of the instrument, and of some of the tests made with it, but did not describe its use for ascertaining the condition of cars, old or new. It is upon this side of its use that, at the Editor's request, the present contribution will chiefly dwell.

For a full account of the theory of the accelerometer used, reference should be made to the B.A. paper, but it may be mentioned that it consists of a lop-sided copper disc, mounted on a vertical axis, controlled in its rotation by a coiled spring, and "damped" by being placed in a strong magnetic field. Any acceleration causes the heavier side of the disc to lag behind, and so partially wind up the spring; the degree to which the spring is wound up measures the acceleration. In addition there is a "compensating balance," which enables the instrument to record correctly, even when travelling around sharp curves on the road, or when the road is heavily cambered to one side or the other. The instrument is provided with a fine pointer, which enables resistance in "pounds per ton" to be read with considerable accuracy.

centre of figure. On the pivot of the disc is fastened a spur wheel which gears in with another equal spur wheel mounted on a parallel axis and carrying the pointer. This pointer moves over the scale shown in Fig. III. It will be seen that there is a small permanent magnet

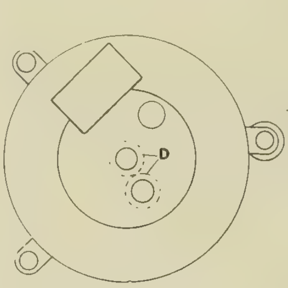


Fig. I.

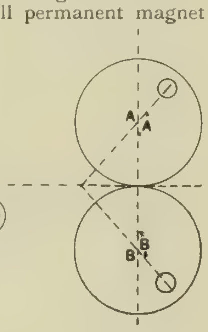


Fig. II.

placed so as to "damp" the motions of the copper disc without having any of the "sticking" qualities which accompany frictional damping. The spring which is coiled up by the rotation

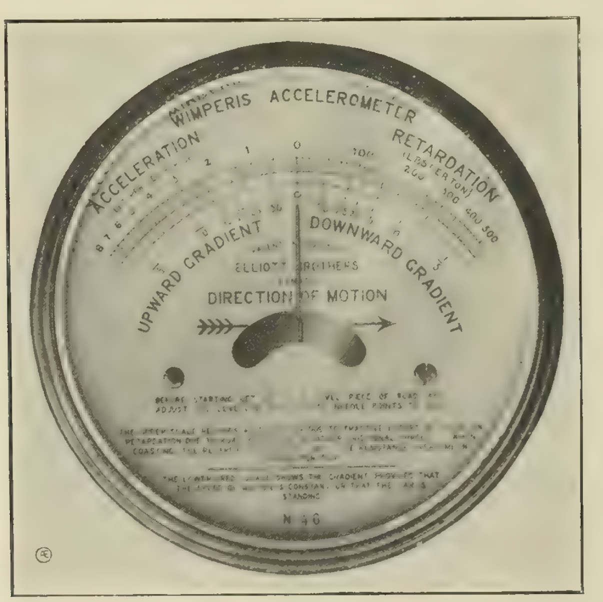


Fig. III.

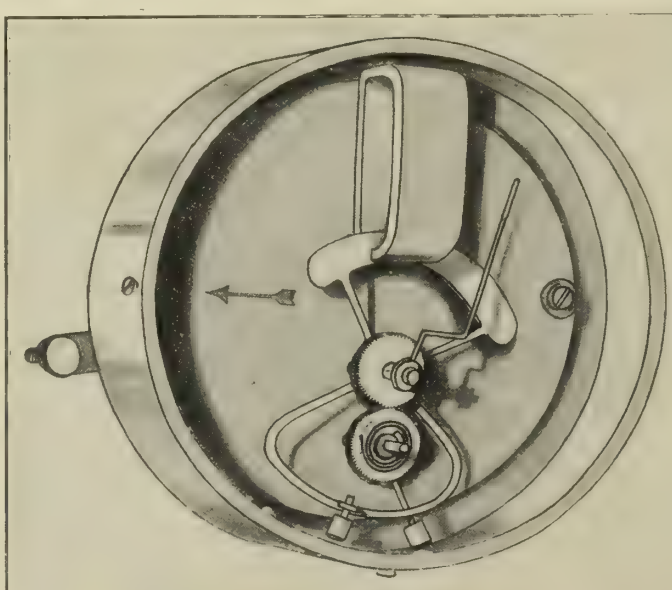


Fig. IV.

A general outside view of the instrument is shown in Fig. I., the dimension across the dial being about four inches. In Fig. I. is given a simple diagram illustrating the relationship of the various parts. The copper disc is shown at D; it has a hole cut in it near the circumference, to throw the centre of gravity slightly out of the

of the disc is not shown, but it lies in the horizontal plane just above the disc. In later forms of the instrument the arrangement described above has been modified, in order to facilitate adjustment, and an internal view is shown in Fig. IV. The general principle of working is in no sense affected by this alteration.

The forces acting at the centre of gravity of the disc are directly proportional to the impressed acceleration, but as the disc rotates through angles of 30 degs., 45 degs., 60 degs., etc., the "arm" gets less, and the couple twisting the spring does not rise so rapidly as the acceleration producing it. The scale has to be graduated in accordance with this fact, and the divisions close up when the angles get considerable. The law which governs this is that of the change of the ratio $\frac{\theta}{\cos\theta}$

It is now necessary to describe the mode of action of the compensating device. If, whilst the disc is deflected by an acceleration, a second acceleration (or slope) should be acting at right angles to the first, there will be an additional couple tending to wind (or unwind) the spring. The needle must then give false indications of the acceleration which it is desired to measure. This would be a very serious fault, since such transverse accelerations are very common in practice, and often of considerable amount, sometimes as much as 10 ft. per second. (This type of error is noticeable in the pendulum forms of instrument, where the complicating vertical acceleration often far exceeds that of gravity itself.) To get over this difficulty the two gear wheels, already mentioned, were added, and it was so arranged that the moments of mass about the two respective axes should be equal. This equality of mass moments makes the system equivalent to two equal copper discs geared together at their circumferences, and each having their centres of gravity eccentric to the same extent. The effect of this is shown in Fig. II. Inspection of this diagram will show that forces in the direction of the centre line will cause the two discs to roll together, whilst forces in a transverse direction cause no rotation whatever. Forces perpendicular to the paper can, of course, produce no rotation of the disc, so that the instrument records the acceleration in one of the three directions of space only, and is not affected by whatever may be happening in the other two. Thus the dial may even be tilted until it is vertical, so that the whole force of gravity acts

across it without the readings being affected at all. The author believes that this is the first accelerometer to read in one direction only.

It is evident that when the instrument is tilted in the direction of motion the needle will move through an angle corresponding to the angle of tilt, and this angle can be read from the (red) graduations on the dial. The instrument thus becomes a "gradometer," and it has the property that

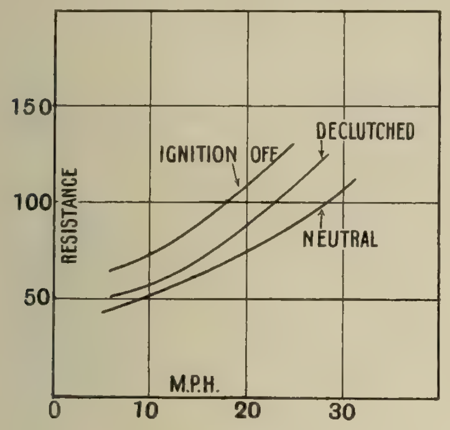


Fig. V.

it can be used just as well for this purpose on a car which is climbing a slope as on one which is standing on the slope, always provided that the speed of the car is uniform.

The model most generally useful for motor car work is graduated up to 5 ft. per sec. per sec. of acceleration, and up to 350 pounds per ton of resistances overcome, besides the red figures showing the gradient. Owing to the way in which an accelerometer works, the following rules* give the interpretation of the readings :—

(1) The reading at any moment, whether the road be level or not, shows the acceleration or retardation due to the algebraic sum of engine effort, road resistance, and braking (if the brakes are on).

(2) If the car be "coasting," and the brakes be off, the reading is the road resistance (including any "air resistance.")

(3) If the car be standing, or travelling on a slope at a uniform speed, the needle will indicate the gradient.

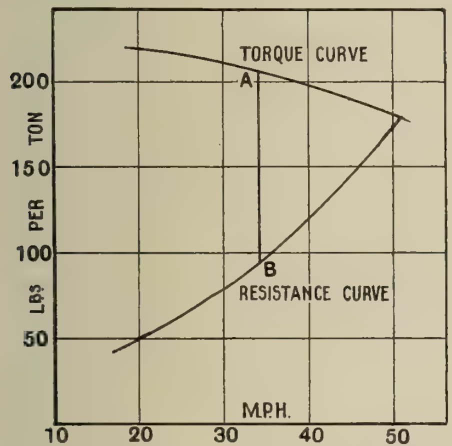


Fig. VI.

(4) If the car is suddenly pulled up by the brakes on a level road or otherwise, the needle will show the total retardation in pounds per ton ; it will also point to the gradient upon which that amount of braking effort just holds the car in.

Bearing these facts in mind there will be little difficulty in following the actual mode of operation when testing cars. The procedure here described is substantially that in force among the engineers under whose supervision the testing of large numbers of touring cars and motor wagons for Government use is carried out.

The four routine tests in which the instrument is used are :—

(1) Measuring gradients of roads encountered.
(2) Testing the brakes of the car.
(3) Measuring the average road resistance as a check on the "miles per gallon" measurement.
(4) Measuring the amount of engine friction, and the mechanical efficiency of the engine.

*Note.—As mentioned in the B.A. paper, a small addition to the "pounds per ton" records has sometimes to be made to allow for changes in rotational momentum when such occur. This normally is of negligible amount.

As regards (1) the measurement of road gradients, this is often essential, as it may be specified that a car must be shown to be capable of ascending a certain slope. Sometimes this is intended as a check on engine endurance, or on radiator temperatures, in which case it is necessary to find a slope of some length. It is a great convenience when searching for hills having the exact slopes and lengths required, to be able to measure gradients without stopping the car.

Test No. 2 is of very real service, as it sometimes happens (far more often than would be anticipated) that cars are sent out for test with the brakes in such poor adjustment that considerable danger is liable to be encountered on steep hills—and for test runs a hilly road is naturally chosen as affording the severer test. At the beginning of a test run it is therefore essential to see that either brake acting alone will suffice to hold the car on the steepest hill to be encountered. This test can be made quite simply on any piece of road, whether level or not, by declutching the engine and putting on one of the brakes to its full extent. If this brings the accelerometer needle over to a gradient reading on the dial of, say, 1 in 7, then that brake will be capable of holding the car on that grade. This test will show that a powerful foot-brake puts a far heavier stress on the transmission mechanism than the most powerful drive of which the engine is capable.

The third test, of average road resistance, is hardly less important, as it enables a proper interpretation to be given to the fuel consumption figures, "miles per gallon," or "gross ton miles per gallon," as the case may be. If, for instance a given car is required to show a fuel economy of not less than 40 gross ton miles per gallon of fuel, it would be very unfair to reject the car because on a day when the roads were exceedingly heavy the performance only showed, say, 35 gross ton miles. The heaviness of the roads should be taken into account; such a day, too, will very possibly be a wet one, and a hood may be in use, so leading to greater wind resistance. But by making measurements of the average road resistance at the time, it is possible by a simple proportion to get the equivalent gross ton miles, and so give the car a fair trial.

It may be mentioned, as a matter of interest, that a knowledge of the average road resistance and of the fuel consumption enables the average brake thermal efficiency of the engine to be worked out. Thus, if during a test run the average road resistance (measured by "coasting" with the clutch out) were 84 lbs. per ton, and the fuel consumption was at the rate of 30 gross ton miles per gallon, it would follow that the total brake work done by the engine per gallon of fuel would be $30 \times 84 \times 5{,}280 = 13{,}300{,}000$ ft. lbs. And if the specific gravity of the petrol were 0.74, and the thermal energy in each pound were 15,000,000 ft. lb., the average brake thermal efficiency would be $\dfrac{13{,}300{,}000}{7.4 \times 15{,}000{,}000}$ or just 12 per cent.

The curves given in Fig. IV., which are taken from actual tests, will show how rapidly the resistance increases with velocity, and why it is, therefore, that fast cars always show fewer "miles per gallon." The following table also shows how greatly road surface affects the resistance. The nature of the tyre (solid or pneumatic) seems to be of very much less importance. Each of the readings was taken on a motor-wagon with solid rubber tyres.

Nature and Condition of Road.	Resistance (at Clutch) in Pounds per Ton.
Tar macadam, dry and hard	70
Ditto, very muddy and sticky	95
Partly rolled road metal..	120

The fourth and last of these routine tests is the measurement of engine friction. This is measured by getting the road resistance when declutched, and then repeating the measurement with clutch in and ignition off. Both measurements should, of course, be made at the same car speed. The difference in the two readings gives the effort necessary to rotate the engine, and measures the engine friction. (The ratio of the two will be easily seen to be the mechanical efficiency of the engine at that speed.) With a new engine which has been run about 100 miles or so, the engine friction will usually be about 15 lbs. per ton. The three curves shown in Fig. V. are taken from an actual test. The lowest was taken when the car was "coasting" in neutral gear, the middle one when declutched, and the upper one when the ignition was cut off. The difference in height of the upper pair is, of course, the engine

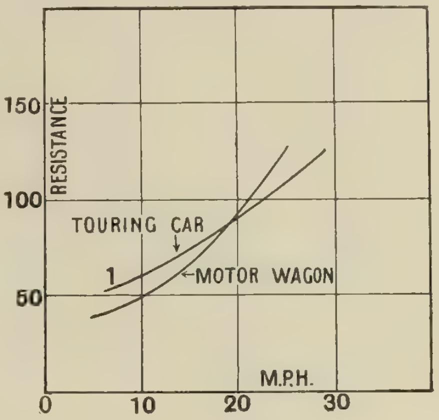


Fig. VIIa.

friction above mentioned. The difference between the lower pair gives the effort needed to rotate the inner part of the clutch and the gear wheels in the gear box—it will be noticed that this difference increases rapidly with the speed, probably because of the churning up of the lubricant in the gear box. It will be realised that the routine tests here mentioned are only a small fraction of those which are possible with the accelerometer. Records can be taken on the various gears, and the transmission losses traced out. By noting the acceleration readings when the car is in motion it is possible to obtain the engine torque at various speeds, and thus reproduce the curve got from the bench test of the engine, with the advantage that it is taken under actual working conditions and at a time when the ability of the carburettor to "pick-up" can be noted. If in Fig. VI. the upper curve be the torque curve and the lower

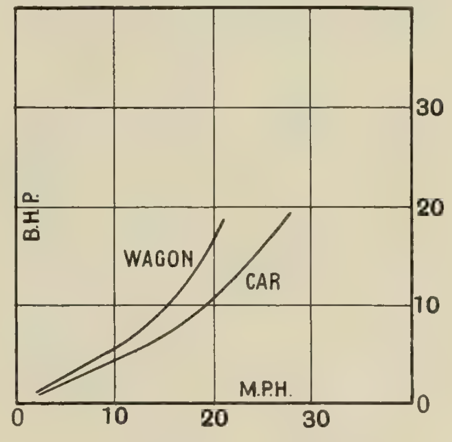


Fig. VIIb.

the road resistance at the clutch, the accelerometer reading at any moment when the throttle is wide open is given by the vertical height AB between these two curves. The point of intersection of the two curves gives the steady speed of the car on a level road with the throttle wide open; the curves shown were taken from an actual car. The set of curves given in Fig. VII. shows the b.h.p. necessary to propel a touring car and a motor wagon at various speeds.

It may be stated in conclusion that the formula for the resistance (in neutral gear) of the motor wagon was found to be

$$R = 38 + 13\left(\frac{V}{10}\right)^2$$

where R is in lbs. per ton and V in m.p.h.

For the touring car of Fig. V. it was

$$R = 47 + 11.5\left(\frac{V}{10}\right)^2$$

Whilst for the touring car of Fig. VI. it was

$$R = 45 + 11\left(\frac{V - 13}{10}\right)^2$$

but in this case the resistance was measured in the declutched position, and there was a following wind of about 13 m.p.h.

THE INSTITUTE OF METALS.

A brief abstract of the proceedings at the recent autumn meeting in Glasgow.

THE Institute recently held its autumn meeting in Glasgow, and among the papers read were several of great interest, both to the maker of alloys for automobile work and to the user.

Dr. Cecil Desch read a very valuable paper dealing with common defects in alloys, and spoke of the very backward condition of the study of what might be called the diseases of alloys. The manufacturer called upon to produce a new alloy, or to modify the properties of a familiar mixture to meet special requirements, has generally to grope in the dark, and it is hoped that in this connection the museum illustrating the "diseases" of metals, to be formed by the Institute, will be of great service. The principal defects dealt with by Dr. Desch were the following :—

Sponginess, caused by gases dissolved in the molten metal being released at the moment of solidification. This may, of course, be corrected by re-melting and pouring at the proper temperature. Blisters in rolled sheets are generally due to the same cause. Often the gas pores are so small as to escape detection, except under the microscope, but the casting is nevertheless likely to prove faulty.

The presence of oxides is generally due to allowing dross to enter the moulds, or to overheating. They cause brittleness, of course, but can be largely guarded against by the use of a de-oxidiser.

Inequality of composition is a common cause of a faulty casting, and in the case of metals which do not mix readily rapid cooling is essential, to prevent the constituents settling down according to their densities. The liability of aluminium to become enclosed in a film of oxide as soon as it is molten is a difficulty often met with in the casting of an alloy of this metal.

Excessively coarse structure naturally implies weakness as the strength of an alloy depends upon the interlocking of neighbouring crystal grains as well as their size. Casting at too high a temperature is the general cause of this fault.

Wrong thermal treatment is a fruitful cause of trouble. Quenching from a high temperature often increases strength, but invariably makes the alloy brittle and unable to resist shock. Heating at too high a temperature, or for too long a time during annealing, is apt to produce a dangerously coarse structure, while unequal thermal treatment of a casting or forging produces, of course, differences of structure and of size of grain.

Molecular or allotropic changes give much trouble in some metals. Tin, for instance, will change into powder at low temperatures, and in cold countries, such as Russia, this "tin plague" is the cause of great inconvenience and loss. Some of the aluminium alloys are liable to spontaneous disintegration, and aluminium containing 20 per cent. of tin will, in time, break up into coarse crystals. The presence of impurities has an influence on some of these changes; copper and manganese, for instance, being liable to spontaneous collapse unless the materials are absolutely pure.

Shrinkage cracks may be due to the unsuitable arrangement of the mould, to a wrong temperature of casting, or to great brittleness of the alloy at a temperature just below that of solidification. The latter condition is the cause of the tendency of aluminium-zinc castings to crack during casting.

"Season cracks" in brass, which show after rolling, are due to the strained condition of the outer surface, consequent on the formation of an unstable mixture. "Fire cracks" during annealing are due to much the same molecular changes, and are generally produced by the use of insufficiently heavy "pinches" in breaking down. It is somewhat remarkable that the state of strain producing these cracks may be removed by shock, and their development may be arrested by striking the object with a wooden mallet, or, in the case of rods, by "springing" them. Severely worked articles, such as brass spinnings, are apt to crystalise, especially when subject to vibration. In this connection the remarkable fact has been recently demonstrated that crystalisation in a strained spinning may be produced by "inoculation" with particles of crystaline metal; the fault, like the "tin plague," being therefore contagious.

Chilling cracks are due to too rapid cooling, and cracking during hot working is the result of treatment at unsuitable temperatures. Most alloys show increased brittleness at some characteristic temperature, and many of the bronzes can only be worked safely over a very small range of temperature.

A supplementary paper was read by Mr. H. S. Primrose, metallurgist to Messrs. Weir, of Cathcart, who, in 1904, built the British Darracqs for the Gordon Bennett of that year. Mr. Primrose made out an excellent case for metallography as an aid for the brassfounder, and showed that the camera and microscope were most valuable aids in the detection of faulty casting, and, while quite easily employed, were of far greater value to the foundryman than the more common method of judging quality by fracture.

The usual standard to which most foundries have to work is governed by rigid specification of the tensile strength and elongation of test bar from the castings produced. While this method affords a valuable check upon the acceptance of faulty workmanship, failure generally results in throwing the onus upon the metal, but most cases of microscopic examination reveal the fact that structural deficiencies are to blame. The ultimate strength and degree of elongation which a metal possesses depends entirely on the nature of the crystalline arrangement, and this in turn depends upon the rate of solidification, and also to a less extent upon the rate of cooling after solidification. For each metal or alloy there is a certain size and arrangement of grain which gives maximum strength and durability. If this be departed from, then there is not sufficient interlocking of the crystals, and thus large cleavage planes are formed, but if the crystal grains are too small the interlocking may be so minute as to be almost valueless.

The results of an investigation into the heat treatment of brass were given by Messrs. Bengough and Hudson, and their paper contains a mass of figures likely to prove of great service to manufacturers and users. It contains much which may be of value to the automobile engineer, but as space precludes the giving of any of the tables here we must refer those interested to the journal of the Institute, which will contain the full papers and the discussions thereon. Nobody who attended this meeting could fail to be impressed with the scanty information we possess relating to the non-ferrous alloys as compared with our knowledge of iron and steel. To the automobile engineer who has to deal with a very highly stressed construction, in which the necessity for weight reduction makes a reasonable factor of safety only possible by the use of the highest quality materials treated in the most efficient manner, the Institute of Metals should make a distinct appeal.

INSTITUTION OF AUTOMOBILE ENGINEERS.

Graduates' Section Proceedings.

ON September 28th there was a meeting of the Graduates' Section of the Institution, and a paper was given by Mr. L. H. Baskerville Cosway on "Long *versus* Short Stroke Engines." Mr. Cosway discussed the mechanical disadvantages of the long stroke engine, laying particular stress on the difficulties caused by the connecting rod fouling the sides of the cylinder. He also criticised the long stroke engines, with which he had made experiment, on the grounds that they were harsh, and generally less controllable than engines of more nearly "square" proportions. At the commencement he gave a definition of long stroke as being any stroke exceeding 1.25 times the bore. In the discussion which followed most of the speakers endorsed the author's remarks, and it was also suggested that the long stroke engine was heavier, power for power, than an engine with a short stroke, the argument being that the area of the cross-section of the engine increased roughly with the square of the stroke, while the power increased only at a much smaller rate. However, the considerable inaccuracies of this assumption were pointed out by subsequent speakers. It was obvious that the majority of those present were very strongly in favour of short strokes.

The attendance was extremely poor, even considering the comparatively small number of London graduates and the fact that the paper was the first of the season. It is surprising that the younger members of the profession should be so little alive to the advantages of meeting their fellows as appears to be the case. Of course, the papers given by graduates for graduates are seldom of as great importance as the papers read before the Institution as a whole, but there is no doubt that the discussion of points which arise from the least important paper may have great educational value. Although there are fewer manufactures in London than in Coventry and Birmingham, it seems that there must be something lacking when it appears that the largest city in the world does not produce a dozen students sufficiently keen on their occupation to give the small amount of time necessary to attend one of these meetings.

The second paper on "Petrol Consumption," read by Mr. Burchell, was somewhat better attended. The author had handled his subject very well, and discussed it from almost every possible point of view. He gave considerable attention to the defects of various carburettor systems, and was especially hard on the automatic air valve when controlled by an adjustable spring, for the reason that it seemed to be the habit of drivers to tighten up the spring the first time they were troubled with starting in cold weather, and that when once tightened the adjustment seemed never to be slacked off again. The discussion showed that all present had followed the paper very closely, and the graduates' section may certainly congratulate themselves on the fact that the first paper to be read in the new premises of the Institution was a distinct success.

The following is the programme of meetings for the present session :—

Nov. 5.—Visit of London, Coventry and Birmingham Graduates to Olympia Show.
Dec. 8.—Paper on "Agricultural Motors," by R. K. Hubbard.
1911.
Jan. 19.—Prize Essay on Olympia Show.
Feb. 16.—Paper on "Steam Motor Vehicles," by E. D. Suggate.
Mar. 16.—Paper on "Motor-Cycles," by T. E. B. White.
April 20.—Paper on "Steering," by C. E. G. House.
May 18.—Paper on "Magneto Ignition," by B. W. Ainsworth.
Visits. —Aero and Motor Show, Commercial Motor Show, official visits to both.

A prize will be awarded for the best essay on the Olympia Motor Car Show in November : open to all graduates of the Institution.

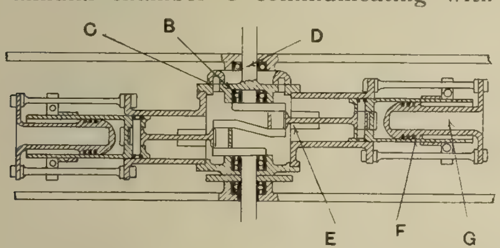
A prize will also be awarded by the Council to the graduate who writes the best paper during the year ending with the annual meeting in March, 1911.

RECENT AUTOMOBILE PATENTS.

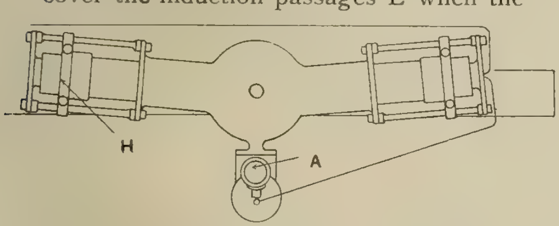
By Eric W. Walford, F.C.I.P.A.

A Two-stroke Revolving Cylinder Engine.

The two cylinders are set slightly out of line to compensate for the weight of the magneto arranged at A. It will be understood that the cylinders revolve around the fixed crankshaft, or the latter revolves as well as the cylinders. The crank chamber is formed with a flat face B, and against this rests an open sided annular chamber C communicating with



the induction pipe D, and also with ports, allowing gas to enter the crank chamber. The gas finds its way along the recesses E into each cylinder above the piston when the latter is, of course, at or near the bottom of its stroke. The piston is provided with a tubular extension F, which surrounds the fixed cylinder head G, and the tubular extension F is formed with an exhaust port. When in operation, as the cylinders rotate the outward movement of the piston draws gas into the crank chamber. This is compressed by the movement of the piston towards the centre of the shaft until the pistons uncover the induction passages E when the

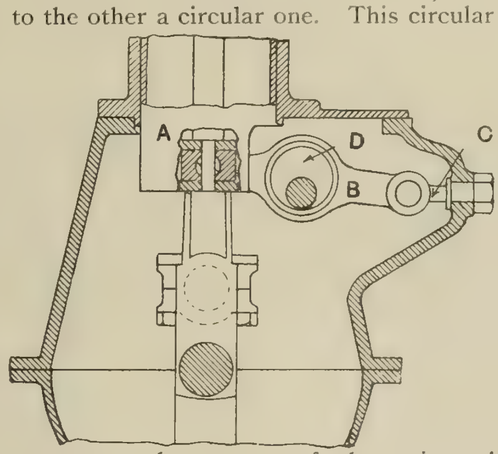


gas enters the cylinder above the pistons. The outward movement of the pistons compresses the gas, and at the extreme position this is fired, driving the pistons inwards until the exhaust ports H come below the fixed head, allowing exhaust to take place. It will be noticed that exhaust takes place radially. So also does the admission of fresh gas by the passages E. The filling and exhaust of the cylinders is therefore assisted by the centrifugal action due to the rotation of the engine.

F. Umpleby. No. 19,969/09.

A Sleeve Valve Operating System.

This sleeve valve is of the type which is reciprocated and also moved angularly. The sleeve is shown at A, and to it is universally jointed one end of a lever B, the other end of which is free to slide only on a guide C. The centre portion engages an eccentric D on the half-time shaft. The eccentric imparts to one end of the lever a rectilinear movement, and to the other a circular one. This circular

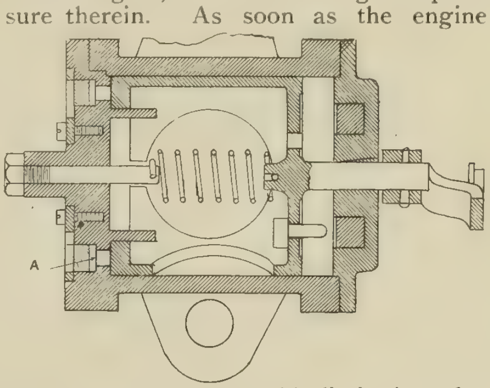


movement, by means of the universal joint, imparts the required vertical reciprocation and angular oscillation in a simple manner.

F. E. Ripley. No. 19,778/09.

A Throttle Valve Improvement.

This particular construction has been evolved to overcome a very present trouble with existing forms of valve. When the throttle is suddenly shut, when running at high speeds, considerable suction is set up on the suction stroke. This draws the oil past the piston rings into the cylinder, with subsequent well-known disadvantages. The present construction comprises a rotating barrel throttle of substantially the usual type, but the feature of it is that it is free to move against the spring illustrated, so uncovering air holes at A. Thus when the throttle is suddenly shut the very sharp suction referred to moves the throttle barrel bodily against the spring, allowing air to pass to the engine, and maintaining the pressure therein. As soon as the engine

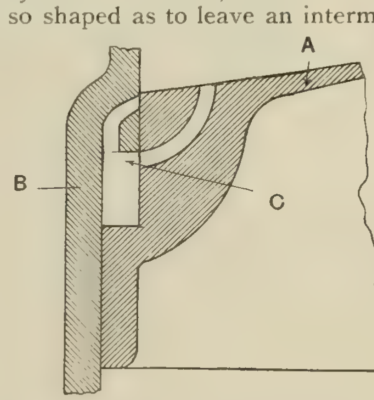


speed falls to a reasonable limit the valve re-seats and operates in the ordinary manner. The valve spring is adjustable.

E. Brandt. No. 22,544/1909.

A Simple Igniting System.

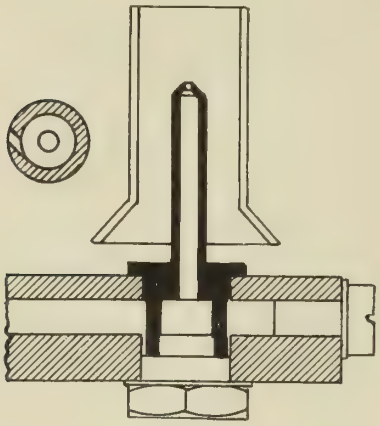
In this system ignition is effected by super-compression of a small portion of the charge. In the small detail drawing part of the piston is represented at A and the cylinder wall at B, and the two parts are so shaped as to leave an intermediate



groove C, in which part of the gas is compressed very highly before the end of the compression stroke is reached. This super-compressed gas ignites the charge in the combustion chamber through the curved ducts, which come into line at or near the end of the compression stroke. A number of other constructions are illustrated, whereby the super-compression is obtained indirectly from the piston and also by the movement of the connecting rod. H. Royer. No. 21,191/09.

A Carburettor Jet Improvement.

The inventor points out that the quantity of air passing through the carburettor is roughly proportional to the square root of the suction pressure, whilst in the case of the petrol this is not strictly the case,



but increases at a more rapid rate as the suction decreases, owing to the friction, viscosity, etc., being approximately the same at all times. The idea is to absorb some of the energy of the moving petrol, and for this purpose it is caused to enter the jet nozzle, which is illustrated herewith in section, by means of an inclined passage, which causes a whirling motion to be set up in the jet, or chamber at the foot of it, so that a slight drag is imposed upon the petrol passing to the jet at high speeds.

A. G. Lea. No. 22,156/09.

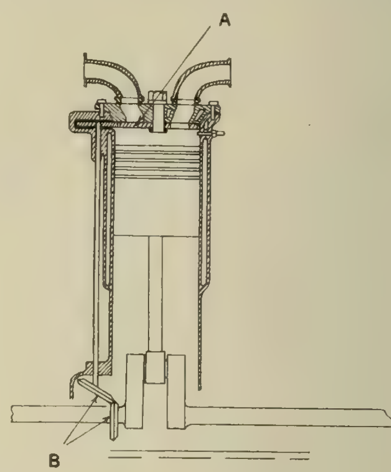
A Novel Radiator.

This comprises a tank, without any special radiating surfaces, beneath which is arranged a fan, delivering air by a passage to a nozzle which projects above the water level. The water outlet from the cylinders terminates in an adjustable nozzle, the idea being that the blast of air delivered by the fan will cause an ejector effect, which will suck the water from the water pipe, mixing with it and cooling it. The mixed water and air impinge on the baffle plate, bringing the two into intimate contact, the air passing away through holes in the baffle.

F. H. Smith. No. 19,618/09.

Valve Gear.

The valve used is of the well-known disc type, being shown at A. It is provided with a single port coming opposite the induction and exhaust passages at each rotation. It will be gathered that it is advantageous to rotate the valve quickly at certain points, and cause it to dwell during the opening of the ports. The gearing used gives this speed fluctuation by means of oblique eccentric bevel wheels B. In another construction eccen-



tric spur gears are described and illustrated; it is probable, though, the inventor refers to elliptic spur wheels.

D. Marshall. No. 22,849/09.

REINECKER SPUR GEAR PLANING MACHINE.

It has already been decided beyond all possible doubt that to obtain silent running spur gears it is necessary that the tooth form should be as accurate as possible, and that the greater the accuracy before hardening the less is the inaccuracy afterwards likely to become. Just as planing has proved to be a satisfactory method of manufacture for bevel gears, so is it coming into favour for spur gear finishing, and the Reinecker machine has been designed on the lines of the same maker's bevel gear planer: that is, to produce accurate teeth automatically. Although the machine is capable of cutting gears from the blank, it would scarcely be economical to use it for such a purpose, owing to its naturally rather slow operation. Thus it is to be regarded as a means for accurate finishing rather than as a complete gear cutter, although, as the blanks only require to be roughed out previous to finishing, the first process can be performed more rapidly, and less carefully, than it would need to be were it the only operation.

The blank requires to be accurate in one respect only, namely, as regards its outside diameter, for it is to this that the tool is set at the commencement of operations. As is shown in the illustration, the blank is fixed on a mandrel, which is in turn mounted on a carriage, and, by means of a system of change gears, the blank is both rotated by the mandrel and traversed by the carriage simultaneously. Meanwhile the tool reciprocates and is gradually fed to the depth of the tooth. It will be seen that the cutting edge of the tool therefore needs no complicated grinding; it has merely to be straight, as it operates on only one tooth-face at a time. The normal tool has its edge at an angle of fifteen degrees to the vertical, and this is suitable for almost all gears, though, as will be seen later, it may sometimes be advantageous to use a tool with a different angle for special work.

It will at once be obvious that with such a method of cutting the face of the tooth, it is possible to get almost any tooth form by altering the relation between the speed of rotation and of traverse of the mandrel, and this is made use of in varying the height or thickness of teeth, so as to avoid under-cutting of the pinions, in a pair of gears where one is much larger than the other, the difference in tooth form obtainable being shown in Fig. I. in an exaggerated scale. Needless to say the automatic dividing, which takes place at the completion of each face, is capable of being set for any pitch, just as the tool can be set to give any involute form to the teeth that may be desired.

The manufacturers make an interesting recommendation for gears which are required to run at high speeds, and that is that the tops of the teeth should be rounded off, and there is an attachment by which this can be accomplished

Unfortunately, there are at present no figures available which serve to show at all accurately the extra time which would have to be allowed for this method of gear finishing, but the following examples are quoted as showing the time required to cut from the solid. This is distinctly slow cuttting, but, of course, the times for finishing would be but a quite small fraction of the amount quoted:—

Cast iron gear, 5.5 diametrical pitch, 96 teeth, 60 mm. face, in five hours.

Cast iron gear, 2.5 diametrical pitch, 59 teeth, 120 mm. face, in fifteen hours.

Bessemer steel gear, 4.75 diametrical pitch, 18 teeth, 75 mm. face, in three hours.

We should be interested to hear from any readers who have been using one of these machines, both as to the value of the accuracy of their work, and as to the extra cost of the additional operation

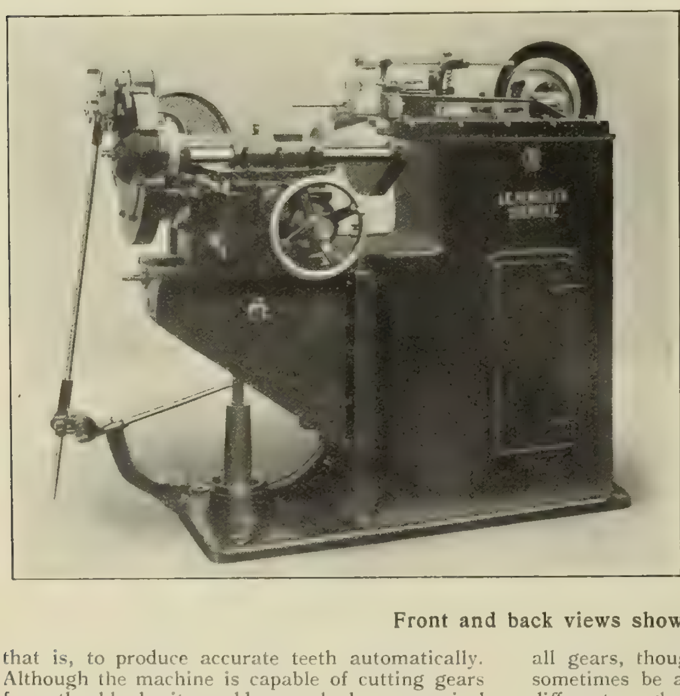
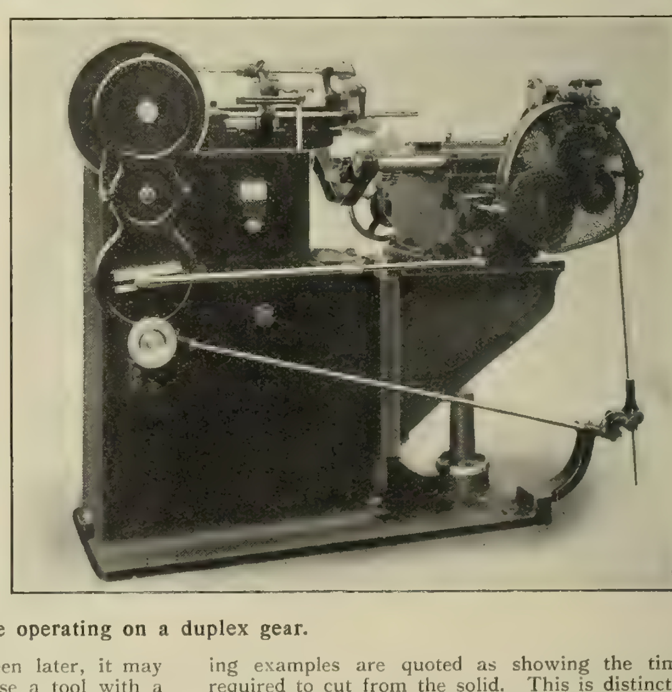


Fig. I.



Front and back views showing the machine operating on a duplex gear.

THE

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Correspondence on interesting subjects, which come within the scope of this journal is cordially invited. All letters must contain the name and address of the writer, but not necessarily for publication. If a reply is desired by post enquiries must be accompanied by a stamped addressed envelope.

CONTENTS.

PRESENT-DAY STANDARD PRACTICE.

ALTHOUGH it is not proposed to do more than touch upon the Olympia Exhibition here, as this event will be dealt with very fully in *Automobile Engineering*, which is principally devoted to it, it is nevertheless worth while to consider the general development made clear by the cars exhibited, and to analyse the trend of design shown thereby on broad and general lines. Firstly, the most striking feature of the exhibition was the evidence of the all-round improvement of small details. The car of to-day is undoubtedly more convenient in many ways than its predecessor of only twelve months ago, particularly in that it is usually provided with a more or less automatic lubrication system; its control levers are better arranged, and petty details of both chassis and body exhibit evidence of the care and thought which has been given to the elimination of every inconveniently placed lubricator or adjustment, and of parts which are likely to produce noises of the smallest character.

Of course, although so much has been done, there is much that still remains which is easily capable of greater improvement, and in some respects where advances were perhaps to be anticipated none are in evidence, but speaking quite broadly, it is not too much to say that the march of progress has been as rapid during the past year as it has ever been in the history of the industry. It is only after a prolonged inspection of new cars that this fact becomes obvious, and the impression which would have been given by a rapid tour of Olympia would have been quite different, for far the most striking thing to a casual observer was the extraordinary resemblance between one chassis and another, especially in the large class of cars with engines having a nominal horse-power of from fourteen to twenty, with a bore of from 75 mm. to 85 mm., and a stroke varying from 100 mm. to 140 mm. (the largest number of engines with the same bore and stroke are 90×120 mm.). Here the majority of both British and Continental makers seem to have trodden the same path, and the result forms an interesting study, for in it may be found impressive evidences of the effect of popular fashions and the effect of the law of the survival of the fittest. The two causes are, to a great extent, intermingled, and are almost impossible to separate with complete certainty. For example, splash lubrication of the uncontrolled type has almost disappeared, its place being taken either by the trough system or by forced or semi-forced systems. This is, of course, an instance of the triumph of the better method, better from the point of bearing durability, of oil economy, and of exhaust cleanliness, yet it is safe to say that many manufacturers have adopted improved lubrication more because they thought the public would demand it than because it made their car a better article. Also, if this matter of lubrication is examined in greater detail, it is easy to see where fashion alone has had its say apart altogether from questions of convenience or efficiency. Take the position of the oil pump and filters; in certain notable instances both these parts are easily accessible, can be removed without emptying the whole crank case, are light, and are driven by short shafts, but in most cases exactly the converse of the above is to be found. Because some of the earlier forced lubrication systems were inconvenient, just as soon as it became necessary to attend to any part of them, there is no reason why the mistake should be perpetuated, and fashion or lack of thought can alone account for carelessness of this nature.

The high average excellency of the chassis is sufficient to show that it is scarcely so likely to be the latter fault as the former influence.

However, to return to the consideration of fourteen to twenty horse-power standard practice: commencing with the engine, this has usually a single piece four-cylinder casting with the valves arranged in pockets on the near side, an internal passage for the intake leading from one or two ports on the off side of the cylinders, a loose single exhaust pipe, enclosed valves with a quick detachment device for the cover plates, and adjustable tappets, usually made loose, and held in place by single stud dogs, and sometimes made as part and parcel of the cylinder casting. Average features are also a four-point suspension aluminium crankcase, with three white metal bearings for the crankshaft and white metalled big ends, a gear pattern oil pump carried by the lower half of the crank case, and driven by a skew gear situated between the cams of the two rear cylinders, either forcing oil to the main bearings and to troughs, to troughs alone, or to all bearings except the small ends through a drilled crankshaft. The timing gears are usually one of metal and the other of some fibrous material, are situated in front of the cylinders, and drive a magneto by means of an extra pinion, the machine being situated on the off side of the engine with its axis parallel to that of the crankshaft.

Quite a large number of engines in the class, of course, differ considerably from this general specification. Thus a good many are provided with a cross-shaft for the magneto drive, this often being adopted to remove the necessity for

another spur gear with its inherent tendency to noise. Where pump circulation is used—and it has very many advantages over its now popular rival—there is much more advantage in the cross-shaft, as it enables a single skew gear to replace two spur pinions, and also makes for accessibility as a rule, particularly as regards the pump gland and as regards the magneto contact breaker also, if the machine is not mounted too far to the side so as to come very close to the front mudguard when the bonnet is lifted. Again, a fair number of engines are supported at three points only, and several are bolted up to the gearbox, the whole unit then almost always being carried from three points only. With four-point suspension the arms of the crankcases are usually connected by tray-form webs of aluminium, on which small parts are mounted, and which also act as an under-screen by joining up the frame sides to the crankcase.

It is probable that rather too much has been sacrificed in the endeavour to obtain a smooth exterior, but only the makers themselves and their founders are ever likely to know how much trouble and how many disappointments have been encountered before the right methods for obtaining reasonably good castings were hit upon, even if they have yet been discovered by all who are now using one-piece castings with elaborate internal passages. In passing it may be remarked that a good instance of the way in which the thickness of thin internal walls is liable to vary could be examined on the stand of one of the exhibitors who had a part-sectional engine. This is not the time to reopen the discussion as to the comparative merits of different casting systems, but it is not out of place to state that there is, as yet, far from sufficient evidence to show that the one-piece four cylinder is in any way superior to the pair-piece casting, though the latter is probably cheaper, and more convenient generally, in workshop and in use, than is the single-cylinder arrangement. Probably many of the valve cover plates have appeared owing to the mistaken notion that they would deaden the slight noise made by the tappets completely. Their effect in this direction is very slight, but the lubrication of enclosed valves is certainly more easy to maintain in a state of efficiency, and also it is probably well worth while to protect the valve stems from dust and dirt.

Figures as to the degree of compression used are very difficult to obtain, but it is probable that compressions are mostly slightly higher than they were, and in several instances where stroke has been increased by some five millimetres or so the cylinders have been unaltered. Sometimes this has resulted in harsh running, which is often ascribed to the longer stroke, whereas it ought to be laid to the account of the higher compression pressure and the consequent more violent explosion.

From the standpoint of accessibility, as regards engines only, there is no very striking improvement, though the tendency to place the float chamber of the carburettor and the magneto both very low down, which was observed last year, seems to be less marked. Valves are the easier to get at by reason of the general removal of the carburettor and the inlet pipe to the off side, and by the use of the common exhaust pipe (not in itself too desirable a feature). The accessibility of the parts of the lubrication system has already been mentioned, but it might be added that the filtering arrangements are frequently inadequate both in respect to the filters in the filling cups to the crankcase, and those through which the pump draws—or ought to draw—its supply from the sump. For the forced or the trough system of lubrication to be at its best it is essential that every solid particle should be removed from the oil at each circuit, and this can only be assured by several successive filtrations through layers of gauze. In any case, the filter ought to be very easy to detach for cleaning, and much improvement is possible in this respect. In fact, the troublesome nature of the task of cleaning out either the cylinders or the crankcase is the chief fault of the vast majority of present-day engines.

Carburettors as fitted to the fifteen horse-power class are so varied in character that it would be misleading to pick upon any one type as being representative of all. Many makers are still using a single jet, which supplies a saturated mixture to a chamber, where it is diluted by an air valve, and others use two jet types, with or without additional air control. Very many chassis were quite innocent of inlet piping, the carburettor being attached directly to a flange on the cylinder casting communicating with branched passages inside. A very few carry the mixture from the carburettor on the off side to the inlet valves by means of a long external piping system, and in several cars the carburettor is still on the near side, with a two-branch inlet pipe, though this form is more common where the cylinders are cast in pairs than when the one-piece arrangement is used. Carburettor control is most frequently by throttle only, the air admission being automatic, but a good number of chassis have both a hand lever on the wheel and a pedal, others having the pedal only, with a dashboard setting to fix the cut-off of the pedal, while a still smaller number have hand control alone.

Ignition is fixed in a regrettable number of instances, though this practice does not appear to be increasing much. Presumably it is being realised that the great additional efficiency which is obtainable by ignition point control is well worth the very small extra cost.

Large and Small Engines.

So far attention has been given only to engines of the class specified at the beginning, and the remarks made apply to an extremely limited extent to the larger and smaller engines. Taking the latter first the numbers exhibited at Olympia were so small that it would be unfair to make any deduction from them, and those which did appear seem to have undergone but little alteration in the past twelve months. Save for only one or two, the very small car is always built to sell at a price, and though this applies throughout the whole range of cars, it is much more pronounced below a chassis value of, say, £200. Everything points to the tendency for the small car to become more and more closely allied to the motor-cycle, and there is a conspicuous gap between this style of design and that of the small four cylinder chassis rated at even so little as ten horse-power. Two cylinder cars, and really well-made cars with single cylinders, are few and far between.

Large engines, in fact, large chassis throughout, have received very little attention for several years, especially in France, if Olympia may be taken as evidence. While the fifteen horse-power design has been the chief labour of automobile engineers, the older, larger, and reasonably satisfactory vehicles have, perforce, had to be left alone, and this gives the impression that the lessons learnt in constructing the smaller cars have yet to be applied to their larger brethren. There is no doubt that as the small car improves, the market for large cars will decrease; but it is certain that the average of large British cars is, at the present moment, above that of any other nationality. In only a few cases is the one-piece cylinder used for engines with a cubic capacity of over three thousand to three thousand five hundred cubic centimetres, pair castings being the commonest practice for both four cylinders or six, though the practice of making six cylinders by the use of two blocks of three is on the increase.

Considering the fact that the principal fault of the six-cylinder engine is its liability to violent vibration at certain speeds, owing to deformation of the crankshaft, it is surprising that there are still so many designs in which only three bearings are used, and this is still more extraordinary when it is noticed that the three-bearing arrangement for four-cylinder cars of fair size is not gaining any ground.

Of course, in considering the lack of development of large four-cylinder engines, it must not be forgotten that the small six-cylinder is becoming very popular, and that it seems likely to become the standard type for cars of from twenty to thirty nominal horse-power, and cars of this class usually bear a very strong detail resemblance to the fifteen horse-power type, their lubrication systems and the disposition of the parts being often precisely the same as in a lower-powered car of the same make. Amongst very large engines the four and six-cylinder position remains much the same as it has been for two or three years, and the only notable changes are the almost complete disappearance of the low tension form of ignition, and the provision of a good number of forced lubrication systems.

No attempt to deal with the present-day types of car engines, however brief, would be complete without some consideration of new valve systems. The Knight engine has, of course, gained a good deal of ground in France and Germany, through its partial adoption by the well-known manufacturers, who have obtained licenses, and there is no doubt that it has been improved in detail since it first appeared upon the market, while its effect may be traced by the chain camshaft drives to be found on several engines with poppet valves. Perhaps contrary to expectations, there were no other new types of engine at Olympia actually fitted to cars, although several British makers have made more or less satisfactory experiments with rotary and other patterns of valves. One or two new varieties which were exhibited separately are dealt with elsewhere in this issue, but they do not appear to possess sufficiently outstanding advantages to make them likely to come into very general use in a short space of time.

Clutches and Clutch Couplings.

Clutches have obviously not been altered by most makers, and the leather-faced cone still seems the most popular, the multiple plate being a close second, with all other types so far behind as to be negligable in a general summary such as this. There is a most noticeable increase in the number of clutch brakes, and there are now only a few clutches which put perpetual end thrust on the crankshaft. The excellent device of fitting a cover plate to keep oil from reaching the leather is to be found on a few cars, and as a rule the clutch is so fitted that it can be removed without much difficulty. Four bolts are most frequently used to hold the pedal, or striking, shaft, and a form of semi-universal coupling almost invariably connects the clutch to the gearshaft, anything from one to half-a-dozen bolts having to be taken out in order to give sufficient longitudinal freedom to the clutch to allow it to be slid off the spigot end of the crankshaft. It is noticeable, perhaps naturally, that cars with leather clutches usually have a fan behind the radiator, and that it is amongst the plate-clutch fitted cars that most combined fans and flywheels are to be found, though this type is not so common as its merits seem to make reasonable.

Clutch and gearbox couplings are occasionally found which would permit the crankshaft and the gearshaft to revolve freely when out of alignment both as regards parallel as well as angular displacement, but the commonest type is undoubtedly the one in which there is a " square " block fixed to each of the shafts and a longitudinally split box containing the two. This coupling is not at all durable if called upon to work otherwise than in exact alignment, it is very difficult to make so that it is neither too tight or too loose, and the slightest shake in it at once produces a knock of the most objectionable character. The forked or ring type is much preferable, but is useless for parallel displacement, unless used in duplicate at each end of a short separate shaft, so that probably the best and least costly connection is a double De Dion type, or a single ring type combined with a De Dion joint, this having the advantage that, being telescopic, it easily gives room for the sliding of the inner portion of the clutch. The size of clutches, that is, the diameter of the leather pattern, and the contact area of the plate pattern, has increased slightly, and might in many cases be made still greater with advantage to the durability.

Gearboxes.

Of course, the principal gearbox development is the increasing number of fourth speeds which are provided, and the action of the manufacturers who have thus altered their practice cannot be commended too highly. There is no doubt that an experienced user would be influenced very greatly in the choice of a car by the number of its gear ratios, as the extra comfort, extra economy and extra engine durability which is given by a comparatively high top speed is sufficient to be of very great importance. The geared up fourth speed is not so common as the direct fourth, and it is largely a matter of the exact work which the car will be called upon to do which should be the deciding factor in making a choice between the two arrangements. For an open car to be used principally upon country roads the direct fourth is to be preferred, but for a heavy covered car, or a car to be used chiefly in towns, a direct third is better, as it is the ratio likely to be most in use; also a direct fourth on such a car is a temptation to bad driving, as its quietness cannot help causing it to be used as much as possible, and the engine will be allowed to " hang on " on top speed at times when a change down would be of the greatest benefit to the whole chassis.

Gears themselves are decreasing in width and the fullest advantage is thus being taken of extremely tough steels. It is more than doubtful whether it will not be found that a good many of the fifteen horse power class have been cut down too much on gear width, in the endeavour to obtain very short stiff shafts, for the tooth load must be very great indeed in many cases. It would seem to be better to use improved material in such a way as to increase durability than in the way we have mentioned, for the extra stiffness obtained by cutting down the length of the shafts by ten per cent. or so has very small effect on silence, and this also tends to decrease as tooth pressures rise. As regards the pitches in use exact details are but rarely disclosed, but the average for the smaller types of car would be about seven (inch diametral), the permanant layshaft drive not infrequently being of a finer pitch. On the score of noise there is but little to choose between coarse or fine pitches, though the pitch of the note may be altered, and the general tendency is to use pitches which are as small as possible consistent with ease in changing speed. More care is being taken to prevent leakage of oil from the box, and the split box seems to be slowly disappearing, owing to the greater manufacturing convenience of the one-piece pattern. Where striking rods project through the walls of the box they are very frequently capped, and caps or stuffing boxes are usual to all the shaft bearings. Needless to add, ball bearings are almost universal for gearbox work, and where two are used at the same end of a shaft they are spaced rather further apart than formerly. The squared shaft for the sliding gears is disappearing rapidly, and it is a disappearance which is in every way desirable. The hexagon shaft has a large number of admirers, if the evidence of Olympia may be so translated, but the splined, or castellated, shaft is used on the majority of first-class cars and there is little doubt that it is the best type, because it is not difficult to make it a most accurate fit with the gear sleeves, while its moment of inertia is the same on any diameter.

There is evidence of small general improvement in the strength of the striking mechanism, and the provision of some locking device, which renders the engagement of two gears at once quite impossible, is now standard practice, the old notched bar with a ball plunger having been found unreliable as soon as some wear has taken place on the change lever and the dividing bar of the gate. The single sleeve, or straight through, type of gear is now only to be found on a quite small number of cars. Concerning the gates, the most common form is that in which the change lever is fixed to a tube which slides outside the hand brake lever shaft, but as this arrangement needs considerable attention, if it is not to be allowed to become stiff, there is a tendency to provide a freer connection by means of a change lever hinged at the bottom and a shaft quite distinct from the brake mechanism. It is only in comparatively few cases that the reverse pinions are mounted so as to revolve only when in use, and it is probably an unnecessary refinement, while it has the disadvantage that it compels the use of extra striking gear. Still there are some cases where a normally stationary reverse pinion has been designed with very little extra complication.

In many cases gearbox bearings are protected as much as possible by enclosing washers, and it will be instructive to see whether these have any direct effect upon the durability of the ball races.

Opinions as to the best method of support for the gearbox are considerably divided, and it would be hard to say whether any one method is gaining ground at the expense of other methods. A very common design is to suspend the box by three or four bolts from a pair of cross members, and another common arrangement is to mount it on a sub-frame, which is in turn hung from cross members. Yet another is to rest it on two dropped cross members of the main frame and in a few rare instances the box has cast arms resting on the main side members of the main frame. In addition to all these different methods there is, of course, the so-called unit system, in which the crankcase and gearbox are bolted together solid, or even made in one piece. Probably this is the best design of all, provided that the casting or castings are sufficiently rigid, which is not easy to guarantee, and there would seem to be many reasons in favour of placing the flywheel at the front end of the crankshaft, though when this is done the shaft needs to be perceptibly strengthened. A conspicuous fault of unit systems is that the clutch is often very difficult to obtain access to, although there is no need for this to be so.

Propeller Shaft Arrangements.

Casual observation would lead to the belief that the standard type of propeller shaft is that in which there is a universal joint at the gearbox end only, the shaft being inside a torque tube, but the double universal, with a tubular or stamped torque stay, is almost, if not quite, as common. The relative advantages of the two broad types have already been discussed in these columns and the conclusion arrived at was that the double jointed shaft was the better. Still, the torque tube design is cheaper and usually neater in appearance, so it seems likely that it will become standard practice. Of methods of anchoring the forward end of the tube the favourite is the forked piece slung from a cross member and free to swivel on the torque tube. Sometimes it is spring-hung like the ball end of an old pattern torque stay, but this is entirely purposeless, as it merely results in transferring all driving shocks to the universal joint, and so to the gearbox bearings. Of course a few forked ends are fixed to the tubes and the spring hanging made to take the place of a swivel, but this must obviously throw such twisting stresses on the tube that it cannot be regarded seriously. If an enclosed propeller shaft is to be used there is little doubt that the hollow spherical end, fitting in a corresponding socket on the gearbox, is the best arrangement, as this combines an entire cover for the universal

joint while it gives complete swivelling freedom to the tube. It is growing in favour and will probably be found in greater numbers next year.

Of universal joints themselves there is little to be said; where telescopic motion is necessary the De Dion type is far the most common, though it is usually made much too small for its work. The next in order of numbers is the ring joint, this promising to displace the older pattern of forked joint, although a fair quantity of cars still have the latter. Encasement, even when not combined with a torque tube, is usually satisfactory, the hemispherical pressed sheet cover being the commonest. This is a step in the right direction, but universal joints are still one of the weakest parts of a chassis by reason of their inadequate dimensions, the difficulty of lubricating them and the absence of easy means of removing the effects of wear.

Under this heading the small general improvement in methods of staying the back axle may be mentioned. Here the use of torque tubes which prevent any telescopic motion has had its effect, because it has produced several cars with radius rods, it being found that if the rear springs are shackled at both ends the axle will roll if its movements are guided by the torque tube only. Torque and radius members combined into links which give a parallel motion to the axle do not seem to have gained much ground, and this is to be regretted, as they have a distinctly beneficial effect upon the steadiness of the car.

Axles, Springing and Steering.

Singularly little alteration is observable concerning rear axles, beyond the increase in the number of worm drives. The bevel differential is far more usual than the spur pinion type, and ball bearings are almost invariably used. The size of the journals and the thrust bearings seems to be sufficient in most cases, and, taken broadly, the rear axle is now one of the most reliable and soundly constructed chassis parts. Pressed steel cases are becoming more usual, while cast steel or cast iron is often employed for axles of the semi-floating or Renault type, these usually being split in a vertical plane, transverse to the axle. It would be difficult to estimate the most common form, but it is probably that in which there is a malleable iron or aluminium differential and driving gear case with flanged conical steel tubes bolted to it, the road wheels being mounted on the outside of each sleeve. There is no noticeable increase in the number of axles arranged so that the differential can be withdrawn without disturbing the road wheels, except that the increase in the use of pressed steel means a corresponding increase of the type. Filling and extracting orifices for lubricant are still capable of a good deal of improvement. Front axles are practically all H section forgings, the jaw part of the steering swivel being most frequently on the stub axle.

Springs have increased a little in length, and the rear spring pads are commonly free to turn on the axle, while three-quarter springs are very popular indeed. The transverse spring has almost disappeared, and will leave few mourners, if any. Great attention has obviously been given to shackle design, and there are now but few cars on which the complete lubrication of every joint is not quite easy.

In road wheel hubs the ball bearings are frequently inadequate, either by reason of smallness of dimensions or by the absence of a thrust bearing. The appearance of the wheel is certainly the limiting factor, and there is evidence of efforts to obtain greater stability by placing the bearings wider apart in the hubs. Many makers are using double row ball journals on the inside of each hub and roller bearings have made some progress. The unnecessary, and even undesirable, splaying of the front wheels still continues, but we are glad to observe that the splayed rear wheel has no new adherents.

Steering gears have changed very little indeed, which is equivalent to saying that most of the parts are too small to be reasonably durable. Ball joints are generally better designed and better protected, while their greater durability has led to their use on the tie rod instead of pin joints, on one or two cars. Many more makers are raking the steering pivots so that their axes more or less intersect the points of tyre contact with the road, and this, of course, necessitates the use of a ball jointed tie rod.

Frames and Brakes.

In this section there is little to be said. Frames tend to increase in section in both directions, become yet narrower in front, to give a wide steering lock, and are very frequently raised over the rear axle. The pressed steel cross-member is becoming universal, as the tubular form is more expensive in every way, and no better. Whether this endeavour to make frames behave like inflexible bed plates is a wise one is not certain, but the body builder at present assumes that there is no whipping and, though a flexible frame could easily be made, it is less simple to see how a body could be otherwise than a rigid structure.

A reduction is noticeable in the number of attachments to the frame, the tendency being to group everything on the crankcase or gearbox castings, so reducing the task of setting one part in alignment with another.

The internal expanding brake with a pressed steel drum and cast shoes, and hand lever actuation, is almost universal on rear wheels. Methods of encasement have improved a little, and there are many devices to ensure equal pressure on each shoe as well as on each wheel. Front wheel brakes cannot be said to have become popular, for though their advantages are undoubted they have not got the stopping power of the propeller shaft brake. Quite a large number of makers are using expanding brakes on the gearshaft with the open side forward, so that the shoes are extremely difficult to reach, but the commonest type is the pair shoe external with either cam or toggle actuation. The best form—the locomotive type with balanced shoes—is proving its merits slowly and stands a little in advance of its position last year.

THE CUTTING OF BEVEL GEARS.

By Henri Perrot and Maurice Jerôme.

A THEORY of spur gearing founded on the principle of there being a centre in space from which all objects are seen, will still be the same as when considered in the usual manner, but a pinion which, when viewed from this centre, appears as a spur wheel, will be in reality a bevel gear. All that is necessary, therefore, for the cutting of bevel gear teeth, is to realise a machine in which the parallel lines of straight tooth gears are replaced by lines converging to a common centre, and where the straight rack of infinite length is replaced by a rack in the form of a great circle, the faces of which will be portions of great circles. This is the main idea, from which have sprung most of the different bevel gear-cutting machines actually in use, which produce by mechanical means the rotation of the spindle which carries the pinion to be cut, and also the travel of the cutters, in such a manner that their relative movements take place on the principle already mentioned. On such machines, therefore, we shall have one or two cutting tools with straight sides moving backwards and forwards in a straight line, and their cutting edges will engender a plane passing through the machine's centre. As all the lines generating the form of the tooth converge to the same point, and as the point of the cutting tool must generate the bottom of the tooth, it is necessary:

(1) That the extended line of travel of the tool's point pass through the centre of the machine.

(2) That the line of travel of the tool is parallel with the bottom of the pinion's teeth

(3) That the apex of the cone formed by the pinion passes through the centre of the machine.

(4) That the axis of the spherical rack, characterised by the cutting tools, and the pinion to be cut, have the same movement of rotation relative to one another, just as though they geared together.

This last condition can be fulfilled:—

(1) By revolving the two axes of pinion and rack without altering their position.

(2) By rolling the pinion on a rack which is considered as being stationary.

A typical example of the first type is the Gleason machine, whilst the Bilgram machine illustrates the second.

The Gleason machine (Fig. I.) has two cutting tools, provided, with a reciprocating movement in a straight line. These two tools work on opposite sides of the teeth, one tool coming forward, whilst the other one is receding, and the length and position of travel of the tools is adjustable. The cutting speed is likewise adjustable, and is obtained by a combination of gears. The pinion to be cut is attached rigidly to a spindle passing through a sleeve, and this sleeve carries the dividing mechanism, which acts on the pinion spindle. Thus the dividing gear being in its locked position, the

whole contrivance—spindle and sleeve—can rock together, whilst, when the indexing is taking place, the pinion and

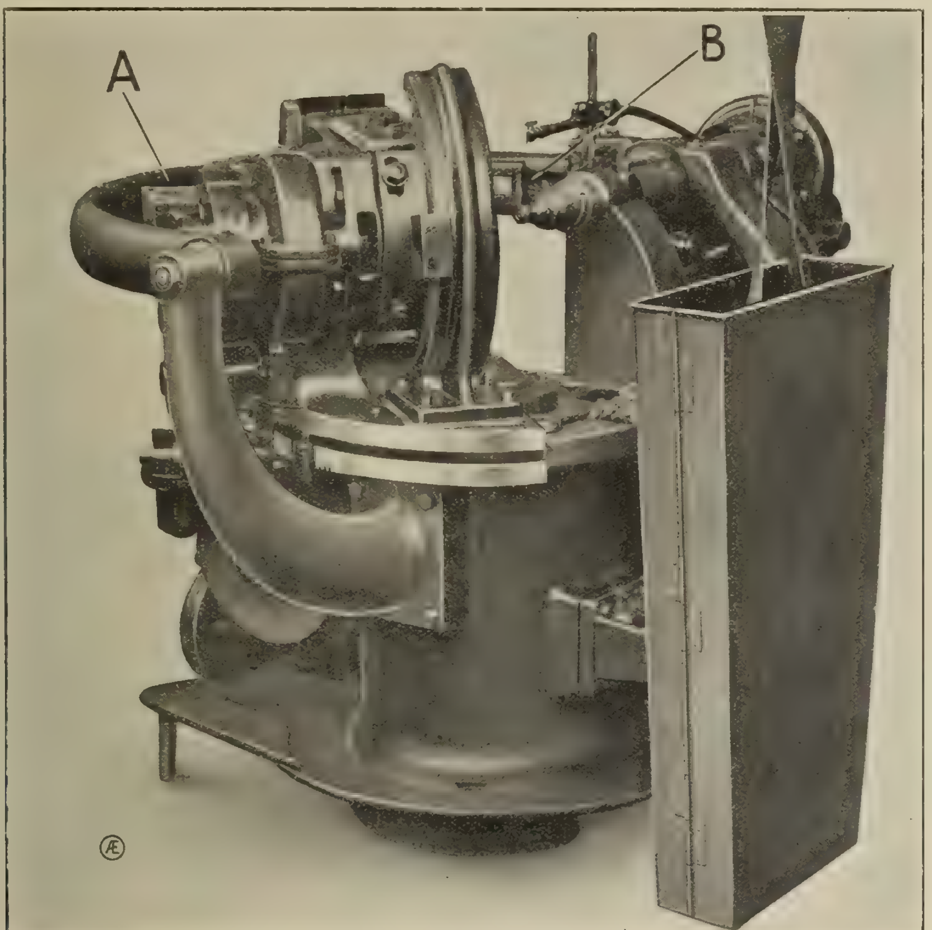
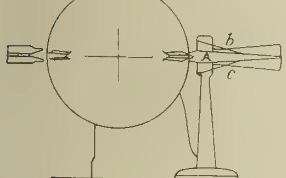


Fig. I.

spindle can turn while the sleeve is stationary. The sleeve is free to slide in a hole provided in the end of a semi-circular arm A, which is carried by a bearing at the other end, and the axis of this bearing is a diameter of the sphere, of which the centre is the apex of the pinion to be cut, which is also the centre of the machine. On this arm at B a segment may be fixed in a certain determined position for realising the amplification of the pinion to be cut. This segment meshes with a segment of a crown wheel having a plane face fixed rigidly to the tool slides, being an amplification of the crown wheel.



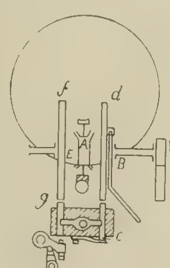


Fig. II. Fig. III.

In this manner the semi-circular arm, in turning round the axis of the pinion, rocks the pinion being cut and the tools as if they geared together. The rocking movement of the semi-circular arm is obtained by means of the cam C acting on a lever D, which in turn operates a vertical rod E, fixed at one end to the semi-circular arm A. The speed of the semi-circular arm may be varied by means of change speed gears, driving the shaft, on which the cam operating the semi-circular arm is placed, by worm gearing. On this same shaft there is also another cam, which operates the backward travel of the tool holder and the entry of the tools at the bottom of the teeth, by means of a system of levers with adjustable travel.

In the old type of machines the cycle of operations used to be as follows :—

(1) Entry of the cutting tools at the bottom of the teeth, the semi-circular arm remaining horizontal.

(2) Rise and fall of the semi-circular arm producing the shape of the tooth.

(3) Disengagement of the tools from the worm whilst the indexing took place.

The disadvantage of this method was that the tool, starting to cut at the full depth of tooth, before having begun to generate, took a cut the whole depth, unless a roughing cut had previously been taken tangent to the gears' finished profile, in which case the tool would start generating at once. Usually, however, this was not the case, and the wear on the machine was very heavy. The cycle of operations was therefore modified as follows :—(1) The semi-circular arm being down, entry of the cutting tools nearly at the bottom of the tooth; (2) Rise of the semi-circular arm to the top of the tooth; (3) Fall of the semi-circular arm, the tool meanwhile having descended to the full depth of the tooth by means of the second cam placed on the same shaft as the cam D, thus making a tooth with a highly finished face. This cycle of operations has a slight disadvantage, which is that it is absolutely necessary to have a regulating mechanism (see Fig. II.).

This apparatus consists of a sensitive gauge, carried on a base plate, the axis of articulation A of which is in the same horizontal plane as the pinion, two springs b and c pulling against one another, and maintaining the needle opposite the zero. The other end of the needle is cut out to receive the tooth between its faces, and also fits in between two teeth, so as to be used for pinions with either an even or odd number of teeth, and the dividing mechanism is then set with the needle at zero.

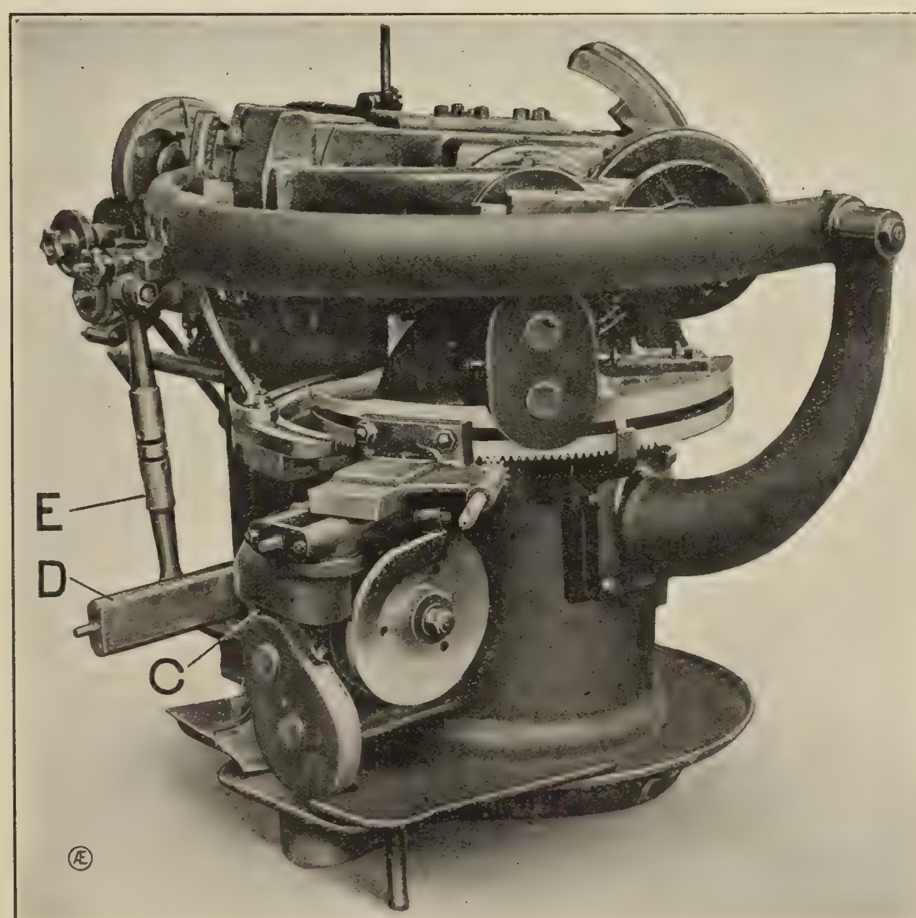


Fig. Ia.

The dividing mechanism (Fig. III.) consists of a differential gear A, one of

the two shafts B being held stationary by a finger c, which engages in a disc d, fixed rigidly to the shaft B, whilst the other shaft E, runs free. When the indexing takes place the disc f, which was previously idle on its shaft, becomes engaged by a ratchet, whilst the stop finger c, disengaged for an instant from its notch, allows the disc to rotate one revolution. At the instant when the finger re-engages with its notch, the ratchet g frees the other shaft. The solitary revolution of the disc d is transmitted to the spindle carrying the pinion to be cut, by means of change gears P and worm gearing.

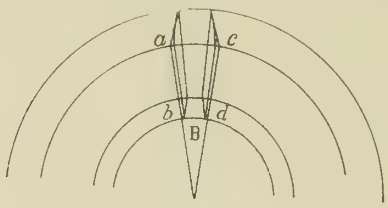


Fig. IV.

As regards setting the cutting tools, let us consider a tooth seen in perspective (Fig. IV.). Then the traces of the two planes tangent to the tooth's flanks, representing the profile of the spherical rack, will be ab and cd, forming between them the angle β. This angle β is therefore the angle which the two slides must form together, and must be calculated, it being necessary to provide on the machine a mechanism which assures the extended line of the tools' travel passing through the machine centre. This angle, of which the theoretical exact calculation is rather complicated, is usually obtained approximately, and with only a very slight error, by supposing the surface of the bottom of the tooth of the spherical rack to be flat, and the sides to consist of little faces, corresponding to the teeth. If the height of the tooth above the pitch line is made equal to the module, and the depth below the pitch line equal to the module + 1/10th of the thickness of the tooth on the pitch line, then the following formula gives the angle for inclining the slides :—

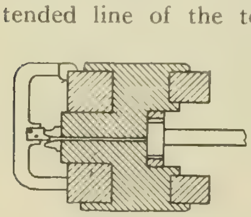


Fig. V.

$\tan \beta = \frac{m}{G}\left(0.7854 + 1.187 \tan \alpha\right)$ where m = module, β = angle of slides inclination, G = length of pitch cone side α = angle of pressure.

If, however, it is necessary to cut gears which do not conform to these rules, a sketch carefully drawn to several times full size will give the required angle quickly and sufficiently exactly for all practical purposes.

The tool slides receive the reciprocating motion which is imparted to the tools by means of two racks gearing into a pinion, each rack being connected to one of the tool slides, and the angle of the two racks, and consequently of the tool slides, may be varied, for obtaining different angles of tooth, by pivotting the racks around the axis of the pinion, a graduated scale and vernier gauge being used for setting correctly. For verifying the position of the tools' points a special device is used. This appliance is placed on one of the faces of the slide corresponding to the tool, and its extremity must just touch the point of the tool, at the same time as its cutting edge, as shown on Fig. V. For facilitating the setting of the tools, the tool holder is made in two parts, each having the shape of taper packing strips, sliding on one another, so as to diminish the distance of the cutting edge from the abutment face of the tool holders.

We have already seen that one of the essential conditions of the machine was that the imaginary apex of the pinion to be cut should be in the centre of the machine. Therefore the face of the machine carrying the tool slides, being set at an angle equal to the outside diameter angle of the pinion to be cut, must be parallel to the lateral face of the turned blank, and at a distance equal to the distance from the centre of the machine to the face which carries the tool slides. This can be verified on the machine by means of a special appliance (Fig. VI.), which is provided with an enlarging device, and placed on the top slide. At the same time the angle of the pinion to be cut may be checked. It is easy to conceive that by this method an error in the outside diameter of the pinion to be cut might pass unnoticed. However, such an error would not affect the quality of the gear cutting, but instead of having the prearranged module, a larger or smaller one would result, according as to whether the diameter was over or under size. This question we shall go into further on. The Gleason machine realises in an absolutely theoretical manner the principle which we have expounded in the beginning, with, however, the following restrictions :—For theoretically correct cutting it is necessary to have a segment corresponding exactly with the pitch angle of the pinion to be cut. These segments can easily be made for special requirements when once the diameter of the sphere corresponding to them is known. If a segment is used, intended for a smaller pitch diameter than that of the pinion to be cut, then, when the semicircular arm is at the top or bottom of its travel, the contact between the spherical rack and the pinion does not take place in the same manner as if the movements between the pinion and tools were amplified exactly. The curve of the segment being smaller than it should be, the ratio of the speeds between the plane crown wheel and the segment is altered, and a tooth is obtained too square at its extremity, while the contrary occurs if too big a segment is used, the error resulting from this often being considerable.

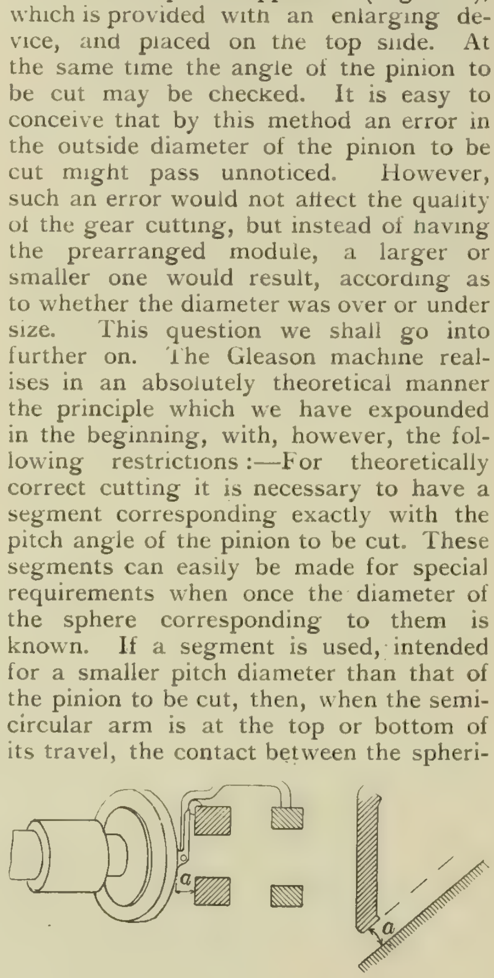


Fig. VI.

For correction, the following method is sometimes used :—Without changing the segment, a cut is taken, as if for a smaller root of tooth than the theoretical size; then a cut is taken as if for a root of tooth larger than the correct size. In this manner the extremity of the tooth can be thinned at will, and the root narrowed until the bevel wheel and pinion mesh together perfectly.

It should also be noted that an error in the root of a crown wheel's tooth alters its profile but very slightly, whereas for a pinion the alteration is considerable, especially as in rear axles for automobiles, where the ratio is sometimes as much as five to one.

It is therefore preferable in such cases to cut the crown wheel first, and make the pinion to suit. This method is, of course, only a workshop makeshift; it is vastly preferable, both for precision and saving of time, to go to the expense of making segments to the correct sizes, and thus turn out work regularly. It is also preferable to employ the Gleason machines for finishing only, so as not to overwork them and so as to obtain the maximum finishing output.

The roughing out of the blanks may be performed more profitably on special gear milling machines of a lower first cost, which also have a greater output for this class of work than the Gleason machine.

Care should be taken not to take too much cut for finishing, as the tool has to finish the whole gear, and it is always difficult to set the same tool twice in exactly the same manner. To facilitate the changing of the tool without stopping the machine, all that is necessary is to have an apparatus situated on the side of the machine for adjusting the tool holders beforehand.

It sometimes happens that gears have to be cut with a shape similar to that shown on Fig. VII. The gear milling machines, which always start cutting from the small end, will not answer the purpose at all, as the milling tool would not clear itself, but would cut into the spindle of the pinion. This roughing out may be done on the Gleason machine in the following manner :—Fix on the lower tool holder a gashing tool, suitably backed off, and set the slide to zero, the axis of gashing being directed towards the centre of the machine. Set a tool in the other tool holder with its two flanks unequally inclined, so as to cut off the angles left by the gashing tool. In this manner a good roughing out is obtained, and these tools stand up better than the tools ordinarily employed for roughing.

Apart from its theoretical qualities, such a machine must possess certain practical advantages. The whole mechanism must be as rigid as possible, this condition being rather difficult to obtain in view of the number of moving parts. It is also important that the machine should lend itself readily to the fastening on of heavy fixtures for the different classes of work which may have to be cut.

The Gleason machine, as actually made, allows a sufficient margin for the draughtsman to design work in a proper mechanical way. The spindle driving the pinion is hollow, and bored taper, so that a large diameter mandril may be posted through, which will easily stand the tortional strains it is subjected to. If necessary, especially for crown wheels having a large diameter hole, the gear can be centred on the mandril, and held between two plates bolted together, of which one is screwed on to the work arbor sleeve of the machine. In this manner the tortional strains are taken on the outside diameter of the sleeve, instead of only on the spindle. The draughtsman must always provide sufficient space at the end of the gear for the tool to clear itself properly.

Conditions for Working Economically and Well.

We have seen in Fig. VI. that the method for adjusting the pinion in position is by measuring the distance a. If we have a series of pinions or wheels to cut we fix them against one of the rear faces, therefore the distance from this face to the angle forming the largest diameter must be exactly the same for all the series of gears which we have to cut. This condition being fulfilled, there will be absolutely no adjustments to make whilst cutting the whole set of pinions or wheels.

To obtain bevel gears which mesh correctly together, they must be assembled in exactly the same position as when they were cut, that is to say, not only must their axes concur to the same centre, but also the apex of their pitch cones must coincide with the same centre of their axes. The only concession which may be granted is that their axes be not quite at right angles. In other words, if we cut a pair of gears whose axes should be at 90°, and if the gears once assembled should be at 91° or 92° or more, as long as the axes of the two gears meet together with the apex of the pitch cones of the two gears, the gearing of the teeth will be excellent, the only result produced by the error of their angles being a slight play between the teeth.

This results from what we explained in our previous article on "Spur Gearing," in the September issue, viz., that it was possible with such gears with teeth cut to an involute curve to increase the gear centres without interfering with the correct meshing of the teeth. We

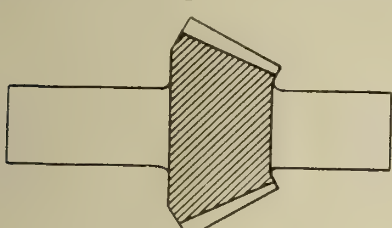
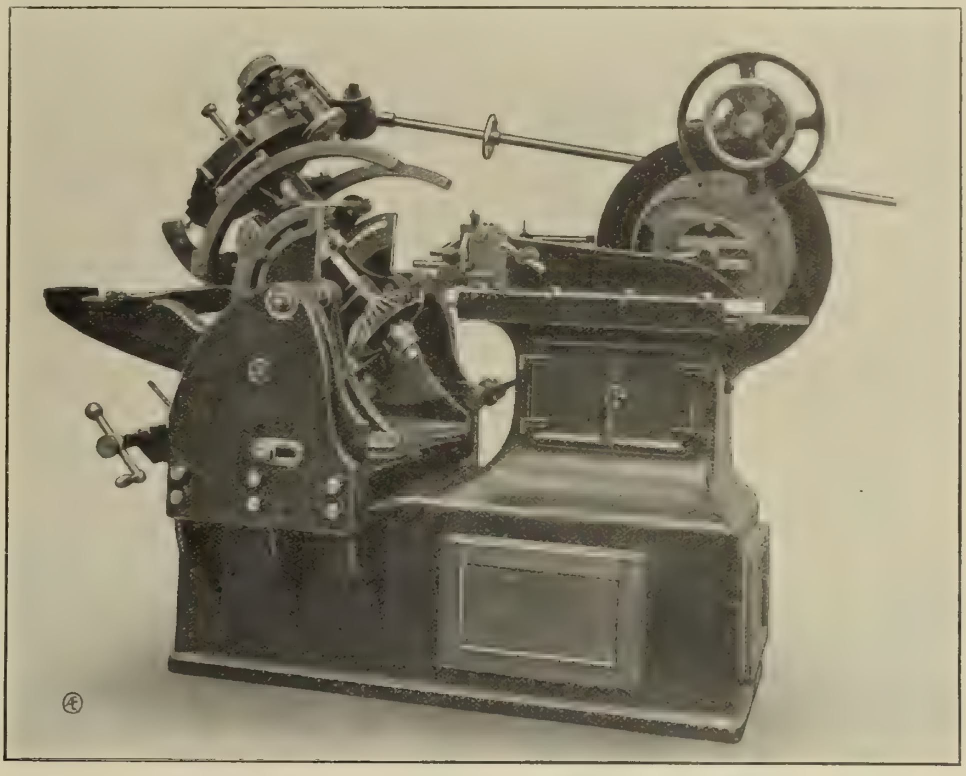


Fig. VII.

now see that, in virtue of the similarity which we have set forth at the beginning of the present article, between spur and bevel gear, the angle of the axes may be varied in involute curve bevel gears without interfering with the correct meshing of the teeth, if the conditions previously stated are fulfilled. Therefore, when it is desired to engage bevel gears deeper into mesh, to take up slack between the teeth, the two gears should be pivoted round their axial centre, and not slid bodily along their axes, as is usually done. As in most cars actually made the rear axle casing is not designed for varying the angle of the axes, it is important that the gears should be assembled exactly as they are cut. The gears should therefore be assembled on the same faces as when cut, and accurate gauges should be made for checking the similarity of position.

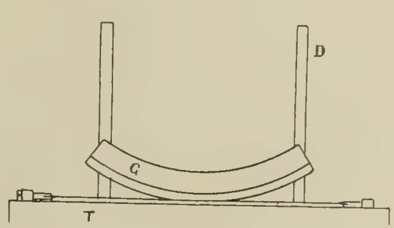
In spite of the large number of moving parts, the Gleason machine is very strong, and cuts gears with great precision; it has a range of change gears for altering the cutting speed, independent of the change speed for the feed, so that it is possible to cut hard steels as well as cast iron or bronze; it is entirely automatic, one operator being able to attend several machines, thus making it very economical, especially for repetition work. Its output is, of course, considerably lowered when cutting piece after piece of different patterns.

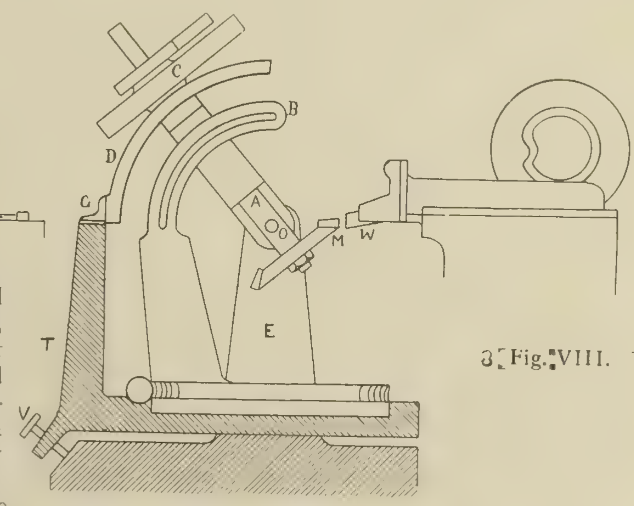


The Bilgram Full Automatic Machine.

The Bilgram Machine.

This machine, which is made in Germany, is based on the same kinematic principle as the Gleason, the difference being in the mechanical means employed for realising the movement. The crown gear, which is characterised by the tool, is stationary so far as angular position is concerned, whilst the pinion to be cut is rolled upon it; the machine has a single tool and, instead of completing one side of a tooth and then going on to the next, indexes each tooth in succession for every stroke of the tool. The tool is reciprocated by a slotted crank, adjustable for varying the length of the stroke, and provided with a cam for lifting up the tool on the return stroke so as to enable the pinion which is being cut to revolve the distance of one tooth.





3 Fig. VIII.

The pinion to be cut is fixed on the arbor A, Fig. VIII., which is supported by a bracket pivoted around the axis O, and locked in position on the graduated quadrant B by means of a nut. The top part C carrying the dividing head has two flats, by which it is held between the segments D. The whole of the mechanism is supported by a bracket E, free to pivot in a vertical direction, and the pivot centre of the support bracket E, the axis of the work arbor, and the extended line of travel of the tool, all pass by the centre O, which is the centre of the machine. The support bracket is pivoted by means of worm gearing fixed on the frame F, the centre of gravity of the mechanism being so situated as to always throw some weight on the screw V. There is also a bolt for locking E and F in position when they are in their correct places, while there is a table T carried by the frame F in the same horizontal plane as the tool.

The rolling of the pinion to be cut on the crown gear is obtained in the following manner:—On the two segments D, Fig. VIII., is fixed a third segment G, having the shape of a portion of the cone. Opposite to the pitch cone of the pinion to be cut, two steel tapes are fastened at one of their extremities to the two ends of the segment G, and at their other extremities to the ends of the table T. By suitably tightening these two tapes the correct rolling of the cone opposite to the pitch cone of the pinion to be cut is obtained, by means of the segment G rolling on the table T, which represents the crown wheel. The rolling of the tapes occurring in a circular path, radial with the centre of the machine, entails a certain variation in their tightness. This,

however, does not affect the working of the machine materially. The movement of rotation is obtained by means of worm gearing, which rotates the support bracket E, around its vertical axis O, and consequently takes the axis of the pinion to be cut with it, as also the segment G, which therefore rolls on the table T.

The remark we made with regard to the Gleason machine is equally applicable to the Bilgram. It is only possible to cut gears theoretically correct by using on the table a segment corresponding exactly to the pinion to be cut. A set of these segments is made and delivered with these machines, so as to be able to cut gears with a near enough approximation in the majority of cases.

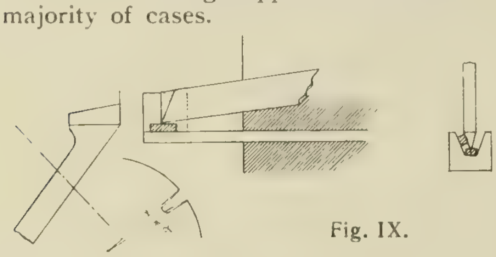
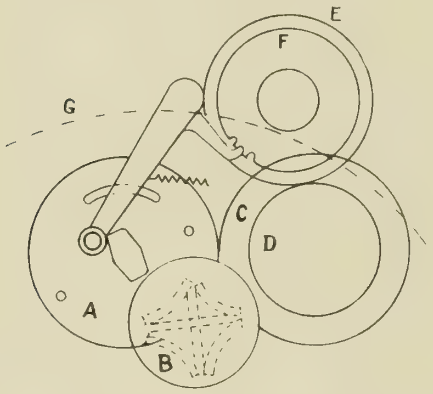


Fig. IX.

The pinion to be cut being fixed on the arbor A, the side O M of the base of the tooth pitch cones is then calculated and measured off with the help of a divided scale, the slide of which is parallel with the line of travel of the tool. When the length O M is read off on the divided scale, the pinion is correctly placed. Another method consists in measuring the height of the tooth above the line of travel of the tool's point by means of a depth gauge. The two methods may be employed simultaneously, and enable the



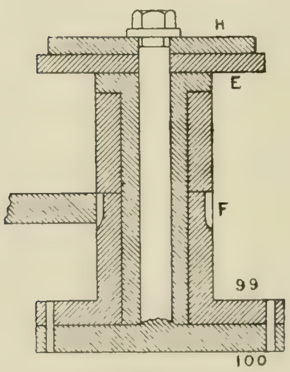


Fig. X.

size of the blank to be verified. The front end of the divided scale has a V-shaped head, very carefully made, to serve for setting the tool. For this purpose two packing pieces are provided, one for the depth (which is 8 m/m., and does not vary), the other for the sides, which varies according to the angle of pressure of the teeth. In all cases the point W of the tool must always travel along the line of the bottom of the tooth. When all the teeth are completely finished on one side, a plate in the dividing head must be turned round to such an extent that the pinion to be cut revolves through an angle α (Fig. IX.), and the tool for cutting the other side is then set in the manner just described. This angle α varies with the number of teeth N of the pinion to be cut, and the angle of pressure β, and the calculation for α leads to the following formula :—

$$A = \frac{600}{N}\left(0.5 - 0.738 \tan \beta\right)$$

Where A is the number of 1/100th of the circumference, which must be made with the revolving plate of the dividing head (Fig. X.). This dividing head is of a quite special kind. A continuous rotating motion is imparted to a plate A, by means of worm gearing, and this plate has two rollers on one of its faces, diametrically opposed, and two segments. Another plate B is arranged, with circular grooves struck off the centre of the plate A, to the same radius and width as the two segments, and standing out from the face of the plate B, so that they may engage with one another. In this manner, even when the plate A is revolving, and one of its segments is engaged in one of the grooves in the plate B, then the latter is kept stationary for a time. The plate B has also two straight ribs at right angles, standing out deeper than the others, and in these the rollers of the plate A engage directly the plate B is freed from engagement with the segments of the plate A. These rollers then take the plate B round, loosing it directly one of these segments on A comes into one of the curved slots on the plate B, the latter then becoming stationary again. In this way the plate B has a discontinued movement, which movement is transmitted by a train of gears C.D.E., to a pinion F, which gears with a large wheel G, fixed on to the work spindle. On the same axis as the pinion F there is a division plate H, so that a finger lever operated by a cam fixed on to the plate A engages with a tooth of the plate H, at the commencement of each cutting stroke of the tool, and disengages on each return. If N is the number of teeth to be cut in the pinion, and the gear ratio of the pinion F to wheel G is 1 : 6, then the dividing plate H will make $\frac{6}{N}$ turns for $\frac{1}{N}$ of the large wheel. If we call n the number of teeth in the plate H, then for the division to be correct $\frac{6}{N}$ must be a multiple of $\frac{1}{n}$ or 6 N must be a multiple of n. The small pinion F (Fig. X.) carries a plate which has 99 holes equally spaced round a circle on its face, and 99 divisions, round its periphery, corresponding to them. This plate is fastened against another plate, with a reduced diameter extension, which is in turn fastened to the dividing gear E, and to the ratchet H. This plate is divided into a hundred holes, arranged on the same diameter as the 99 holes of the plate carried by the pinion F. Therefore, one plate may be rotated in relation to the other 1/100th turn, and a fraction equal to a certain number of 1/9,900. Example—

Suppose we cut a wheel with 65 teeth, with an angle pressure of 15°. From our formula we find $A = \frac{600}{65}\left(0.5 - 0.738 \times 0.268\right) = 2{,}F9$, but A being in $\frac{1}{100}$th parts of the circumference this expression becomes :

$$\frac{2}{100} + \frac{79}{10000} = \frac{2}{100} + \frac{78}{9900}$$

We take out the stop pin which fixes the two plates, we move two divisions, and count to the 78th hole, where we fix the stop pin.

This machine is very suitable for pinions having a large hole, and the manner in which the tool operates is very advantageous. It is, however, important that there should be no play between the tool and the tool holder, if a high efficiency is aimed at. It is useful for very hard steel gears, as it is capable of taking a powerful cut at a low cutting speed. From the sketch of the machine it can be noticed that the pinion takes its abutment whilst being cut on its inside face. This face should therefore be very flat, and for repetition work should always be the same distance from the outside edge. Again, if these pinions once cut are intended to bear against the outside face, then the distance from the outside edge to the outside face must be absolutely alike for all.

Method of Cutting Pinions which have a Small Number of Teeth.

It often happens in automobiles that it is necessary to cut bevel pinions with a small number of teeth. If the ordinary methods are employed for cutting them, taking an angle of pressure of $14\frac{1}{2}$°, then the tooth is narrower at the root than on the pitch line, or in other words, there is "undercut." This not only weakens the tooth, but is also detrimental to the smooth meshing of the gears.

Experiments have shown that a correction is necessary for pinions having a smaller number of teeth than shown on the following table :—

Ratio	2 : 3	1 : 2	2 : 5	1 : 3	1 : 4	1 : 5	1 : 6
No. of teeth	20	24	26	27	28	29	30

Two methods may be employed for avoiding undercut :

(1) By increasing the angle of pressure up to 20°, and even more in special cases.

(2) By diminishing the addendum or height of the tooth above the pitch line in the crown wheel, at the same time increasing that of the pinion a corresponding amount, so as to retain the same total depth of tooth, and pressure angle.

The amount which it is necessary to diminish the height above the pitch line of the crown wheel must be obtained from a drawing made several times full size, and the diameters taken from it.

The two methods give equally good results, but the first has the disadvantage that it increases the pressure on the bearings slightly. On the other hand it has an advantage in that it is possible to use the special Brown and Sharp tooth caliper (which gives the height of tooth above the pitch line for each module, at the same time as the correct thickness of the tooth on the pitch line), thus making it easy to verify all sorts of pinions.

For teeth cut according to the second method, it is necessary to make a special gauge giving the addendum and thickness of tooth on the pitch line for each kind of pinion and crown wheel.

OXY-ACETYLENE WELDING AND CUTTING.

A consideration of the value of the process for automobile work.

By A. J. McDonald.

ONE of the greatest advances in the industrial world in recent years is undoubtedly the oxy-acetylene and oxy-hydrogen process of welding and cutting metals.

In 1904 a cutting blowpipe was patented and to-day, because of its perfect simplicity, its facile management, compactness and portability, an oxy-acetylene outfit has become an absolute necessity to the metal world generally and an ideal apparatus for the automobile engineer in particular. Properly constructed, the apparatus is perfectly safe to use, even for the inexperienced, but it is also essential that the properties of the gases and the conditions under which they are utilized are fully understood. For in the first case unless the gases are under complete control, which is not the case with a badly designed blowpipe, they become dangerous to use, and in the second case without a knowledge of what is actually taking place, good welding is not by any means likely to result. An apparatus generating so intense a heat is not an appliance to be carelessly handled.

Made scientifically, such as the better qualites are, they are almost automatic in their work and can be not cnly safely used, but the work can be done more quickly, better, more easily, and more economically than by any other method. It is intended in this article to give a good working knowledge of the process, what to avoid, how to select a blowpipe capable of satisfactory work and proof against back-fire and other dangers, the best methods of generating and using gas, and the various methods of treating different metals.

This knowledge is not at all difficult to acquire, and, once the working conditions are understood, failures should be impossible. The success of the process is due to the recently discovered methods of passing the two gases in certain proportions through a metal torch or blowpipe, which would *not* allow the flame to flash back into the gas retorts. Various designers have succeeded in fulfilling the necessary requirements satisfactorily in different forms—such as automatic stop-valves, close-mesh fine-wire screens, porous earth, fine tube coils, a packing of many small tubes in a larger one, and many other systems.

Acetylene contains 92.3 per cent. of carbon and the balance is hydrogen, a proportion of which may be phosphorated and sulphuretted, and there are small percentages of other impurities. This is the reason of the highly combustible and explosive nature of the gas.

The only difference between combustion and explosion is the speed of propagation of the flame. High speed means explosion, low speed combustion. The presence of hydrogen retards combustion of carbon in oxygen, and therefore the speed of the propagation is slower than if the hydrogen were not there.

This is the reason why pure oxygen is mixed with acetylene to generate the hottest flame for welding heats and is the greatest heat that has as yet been produced by combustion, being within a few degrees equal to that of the electric arc.

The metal torch or blowpipe must have several special features complied with in its construction. It must have a mixing chamber, where the two gases can be admitted in the right proportions, it must be light, very strong, and be able to withstand great heat. Also the construction must permit of the two gases being mixed as near as possible to the flame, and the quantities of gas used must be supplied in the smallest possible volume. The blowpipe must be constructed and connected up to its gas supply in such a manner that perfect freedom of movement is obtained, so that every part of a job can be easily reached. The hose conveying the gas must be able to withstand a pressure of 400 lbs. per square inch, for a period of not less than five consecutive minutes, so it must therefore be made of not less than three ply, and protected by a strong, flexible re-inforcing outer cover. The fittings must all be of the best, and the hose must be clamped to the fittings in such a manner that they will stand a pressure at least twice that obtained through the regulating valves. Stop-valves must be placed in perfectly accessible positions on both gas supplies.

Different Systems.

Roughly, there are three systems of blowpipes—low pressure, medium pressure, and high pressure. In the low-pressure torch the acetylene is drawn into the tube at the point of combustion by the suction set up by the flow of the oxygen which is under pressure, reducing valves on the oxygen storage tank, as well as on the torch, regulating the flow. The acetylene gas has merely a few ounces pressure, just sufficient to cause it to flow, the higher speed of the oxygen drawing it into the mixing chamber situated close to the nozzle.

Any ordinary acetylene generator will answer the purpose, but it is essential that the proportion of acetylene should be constant, hence the necessity of controlling valves for the oxygen supply. The oxygen can be used from the ordinary steel bottles or from tanks, provided that the proportion of 1.7 oxygen to 1 of acetylene is maintained. If too much oxygen is admitted, an oxidizing flame results, if too little a carbonizing flame. It is most important to avoid either of these flames, as it is impossible to make a good weld if these plus or minus conditions of oxygen exist. In the low pressure type the gases are, of course, flowing at a speed very little above the speed of propagation of the flame, which causes an occasional back-fire. The safety mechanism of the modern good quality torches, however, traps the flash back and renders them perfectly safe to work with.

The medium-pressure torch is now the one in general use, and is free from the faults of both the very high and low pressure systems. Both the acetylene and the oxygen are used under pressure in the medium-pressure torch, the oxygen for welding purposes being fed at a pressure of 15 pounds, but, when cutting, at any pressure up to 300 lbs., if for very heavy metal. The acetylene is used at 2 to 3 lbs. pressure. The two gases flow through small holes into the mixing chamber, and the correct proportion of the two gases is quite easy to fix and maintain by regulating the supply pressure where the gas enters the torch. In the medium pressure torch the correct proportions for good work is 1 of acetylene to 1.3 of oxygen. Flashes-back are not likely to occur, as the pressure of gas supply keeps the speed above the propagation of the flame. The points to avoid are, clogging the nozzle with small particles of metal, over-heating of the torch, and touching the metal being worked on by the nozzle of the torch. Another advantage with the middle pressure torch is that dissolved acetylene can be used as easily as the gas from a generator. So that both gases can be fed from tanks, and the trouble of cleaning and charging generators is removed.

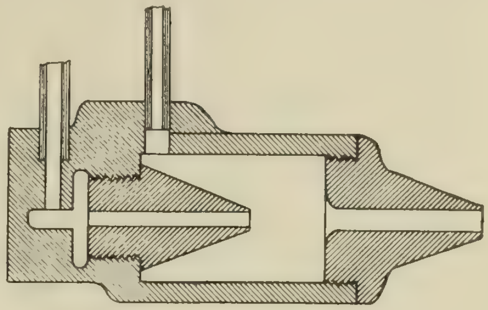


Fig. I. Low Pressure Torch.

Back-fire is the only real danger of the oxy-acetylene blowpipe, and there are several causes for it, but backfire can in every case be rendered harmless if ordinary reasonable care be used in selection and use. A few have attempted to discontinue the use of the back-fire preventer, and they have paid dearly for their folly. Oxygen can be safely used in conjunction with other gases than acetylene, such as hydrogen and various gases made from benzine, petrol, or alcohol, in a torch without a flash-back preventer, but the combinations of all these produce heats of very much lower temperatures than oxy-acetylene, and are only good for certain metals.

No satisfactory weld can be worked except with a flame which is halfway between an oxidizing and a carbonizing one. To obtain this the oxygen must reach the torch at a certain pressure for a given thickness of metal. The pressure of the acetylene can vary, for by regulating the speed of the oxygen the correct composition of mixture can be easily arrived at. The regulation of the pressure is very accurately obtained by means of reducing valves between the storage tanks and the torch, the reducing valve on the oxygen tank regulating the pressure from 15 to 40lbs. per square inch, according to the thickness of metal to be welded, and not less than 50lbs. fur cutting.

The gases used in cutting and welding are hydrogen and carbides of hydrogen, and as hydrogen is a gaseous element it burns with a simple flame, which means that the flame does not hold solid particles in suspension. Illuminating gas, acetylene gas, etc., or carbides of hydrogen decompose into hydrogen and particles of carbon in suspension. The hydrogen easily burning and producing intense heat, the result is that one of three conditions follows :—

1.—If the oxygen is more than sufficient to consume or burn up all the hydrogen and carbon, then the result of combustion leaves oxygen in excess, carbon dioxide and water vapour.

2.—If the oxygen is below the quantity necessary to completely burn the hydrogen and carbon, the hydrogen burns away first, leaving a brilliant flame from the combustion of the carbon.

3.—If the quantity of oxygen is exactly enough for the complete combustion of both carbon and hydrogen, the resultant products are water vapour and carbon dioxide, and the flame, not holding any particles, will be pale.

No. 1 is a non-reducing flame, the carbon all being consumed, with an excess of oxygen remaining. Carbon dioxide and water vapour are obviously non-reducing substances, so any excess of oxygen attacks iron or steel and oxidizes it. The oxide formed is non-reducible by the non-reducing flame, and as it cannot combine with the metal it forms a weak brittle metal in the weld. In the second case part of the carbon is not burned. The carbon in suspension at a red-heat dissolving into iron forms graphitic carbon. Therefore it is perfectly obvious that if the weld is in steel and the joint is made with graphitic iron, it is liable to failure.

In the last case, however, the oxygen is sufficient to burn all the carbon in suspension, and insufficient to transform it into carbon dioxide, therefore it gives off carbon monoxide and water-vapour on leaving the nozzle; but the monoxide is combustible, and immediately it comes under the influence of oxygen it burns, and becomes a reducer of great power. As a matter of fact, it is the reducing agent for oxide of iron when smelting ore in the blast furnace. With the assistance of the great heat, the reducing flame will change and reduce any oxide of iron to the metallic condition. Therefore, with a nicely-adjusted flame a weld is not only possible but, being free from carbon and oxide, the weld is practically perfect, the fusion of metals resulting in an efficiency impossible by other means.

Eye-Test for Correct Flame.

The correct flame for welding is elongated in shape and of two parts. One part takes the shape of a cone and is very bright, forming a point close to the nozzle and measuring from a quarter to half-an-inch in length. An elliptical shaped flame envelopes this cone, this envelope being almost colourless, but giving out a very slight tint of red, and being pointed at each end. In the case of the oxy-hydrogen flame this cone is very difficult to trace, and when regulating the gas to get at the right quality of flame, the cone will suddenly become red violet colour, which tint determines the exact mixture of gases required. With oxy-acetylene there is no trouble whatever in arriving at the right mixture by the judgment of the eye, for the almost colourless envelope of flame and the brilliant white cone contrast very strongly, so a regulation of perfect balance of gases should never be found in the slightest degree difficult.

Referring to Figures I. and II., in the low pressure, as has been explained, the oxygen gas is under pressure and the acetylene at the pressure of the generator, which is never above 10 oz. per square inch, and is only sufficient to cause the gas to flow through the connecting pipe to the mixing chamber. There it is drawn in by the rush of oxygen passing through the mixing chamber. It is impossible to regulate the amount of acetylene drawn in except by altering the compression or flow of oxygen, as the oxygen can only draw in an amount proportionate to the square of its own velocity. Immediately any variation of the pressure of the oxygen takes place the quality of the flame is affected, and any excess of oxygen causes an oxidising flame. In any

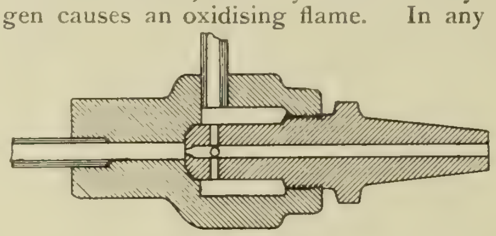


Fig. II. High Pressure Torch.

case it will be clearly seen that there must at all times, with the low pressure type, be an excess of oxygen in the centre, and that is why the low pressure type uses a greater quantity of oxygen than the higher pressure. This excess not only increases the length of time making a weld, but it is certain by reason of the brittle nature to lack in efficiency.

In Fig. II. it will be seen that with the gases both under pressure, meeting at a right angle and having to flow the length of the mixing chamber, the molecular contact is better than in Fig. I., and an average economy of about $33\frac{1}{3}$ per cent. of oxygen is secured. Oxidation is reduced to a minimum and a very big saving (probably on an average of over 50 per cent.) in time is effected, and the efficiency of the weld is very much higher.

The arrangement of reducing valves, combined with the fact that both gases are under pressure, produce a positive mixture in a perfectly homogeneous state, and with these conditions good welding is not only easy, but with ordinary care a very high efficiency can be maintained.

An important consideration is the method of producing acetylene gas. It is essential that the gases used should be as pure as possible, for there are many impurities in the gases which would, unless removed, cause trouble and possible failure. As an instance, in feeding water to carbide a separation of a portion of the carbon from the gas occurs and would result in clogging up the pipes and nozzles. Also the calorific value of the gas is lowered and the production reduced.

Insufficient water to slack the carbide is the cause of this, and it cannot take place if the carbide is fed to the water instead of the water to the carbide.

Professor Pond, who is probably the best authority we have on this subject, has explained in his pamphlet the reasons why the carbide to water system is "under all conditions the best for producing the purest gas," two of the leading causes being that it is impossible for the water to reach a high temperature, and the acetylene having to bubble through the lime water is washed free from all impurities. Also there is a very considerable economy in the use of the carbide.

Dissolved acetylene is generally free from these drawbacks, and is the most convenient form for general purposes.

Oxygen produced under certain conditions contains impurities which must be removed, or better still avoided, chlorine being one that has a very bad effect on welding. Chlorine is present in oxygen when produced in small generators, and can be removed by washing by passing through 9 feet of washing fluid if generated at atmospheric pressure, but when under greater pressure the globules offer only about a tenth or twelfth of the surface, and would require ten or twelve times the washing fluid—easily possible in a stationary generator, but impossible in a portable one. Therefore for small work, average jobs, and where it is essential to have a portable outfit, dissolved acetylene and oxygen in cylinders are best. These are perfectly safe to use, and their purity can nearly always be relied upon. For stationary work or heavy work large generators are recommended, and in the larger sizes the production of impure gas and other faults of the small generator are avoided.

Instructions for Welding.

For welding cast metals, with the exception of steel, special fluxes are used to break down the film of oxide surrounding the molten surface. This can be done by mechanical means, by working up the molten metal with a "feed-stick," but as a rule the operator prefers the flux for heavy work, and on light work it is often impossible to use the feed-stick. For welding cast iron a soft iron feed-stick is used, and for other metals a stick of their own material is employed. For mild steel a soft iron feed-stick is sometimes used.

For heavy work only it is possible to save gas by heating the article to a black or dull red heat beforehand. This is especially recommended for heavy cast iron or steel articles, and can be done in a muffle or upon an open smith's hearth. The quality of the work can be improved by annealing, say by covering up the heated work with dry sand or hot ashes, and allowing it to cool down slowly.

Annealing is not usually resorted to for mild steel, wrought iron, or small cast steel articles, but is necessary for a subsequent soft condition of metal in cast iron work. Copper and the bronzes, on the other hand, are annealed by rapid cooling, as is well known

Welding by oxy-acetylene is really very easy, and can be mastered by any intelligent workman in a very little while. Cast iron can be welded to cast iron, steel and bronze can be welded together, or rolled steel to cast steel. A tooth can be welded on to a broken gear wheel (teeth have been built up or welded on from very small sizes up to 5in. deep and over 20in. wide), or a lug can be welded on to a casting.

On some cars it is desirable to construct back axles of part steel and part bronze, and it is possible with oxy-acetylene to unite the two steel tubes to the bronze centre. In fact, a larger variety of metals can be joined than was possible in blacksmiths' welding. Piston rods can be lengthened, frames can be welded, holes bored through

heavy bars or plates, cracks can be repaired in practically any metal, and if necessary the defective part can be cut out by the cutting blowpipe and a new portion welded in by the welding blowpipe.

In the case of joining two plates, the two edges are each bevelled off at an angle of 45°, forming a groove of 90°. The point of the white cone of the flame is then worked along the bottom of the groove. The metal begins to run first at the bottom and then at the sides, running in and filling up the groove. Additional metal required is added by holding the feed-stick in the groove, the flame striking the stick and causing it to melt and run as a stick of wax might, and when the molten metal reaches the level of the plates the gas is shut off and the metal hardens into one unbroken jointless sheet of metal.

Metal Cutting.

Theoretically, once iron is ignited in oxygen, if a powerful jet of gas is fed to the part to be operated on, the metal burns completely away without any auxiliary source of heat. Oxide of iron, however, forms at a comparatively low temperature and lacks fluidity, thus preventing intimate contact of metal and flame, and cutting became impossible. The oxy-acetylene blowpipe proved a very simple solution of the difficulty, for by the presence of a controllable oxygen jet, additional to the flame feed, the necessary additional heat is furnished sufficient to render the oxide fluid, and to blow it away down through the cut before it can arrest the action of the cutting flame.

The edge or surface of the plate or bar is first heated at the point where the cut commences. This can be done by oxygen and *any* suitable gas. When this spot has been brought to a state of incandescence, a fine cutting jet of oxygen is discharged upon it. This immediately produces combustion of the metal with the resulting formation of iron oxide. The jet of oxygen is then turned on strong enough to blow the iron oxide away in front of it, with the result that a clean narrow cut is effected through the metal at a speed of travel equal to hot sawing. The metal on either side is neither melted nor injured, as the action travels too quickly for the heat to spread and the edges present the sharp metallic surface of a saw cut. Also the amount of metal wasted by being melted away is very small.

THE 17.9 H.P. ARMSTRONG-WHITWORTH CHASSIS.

An exceptionally sound design with some unusual details of construction.

If it be admitted that one of the greatest difficulties to be overcome in the design of a pleasure car chassis is to strike the happy mean between contradictory characteristics, it is necessary to define exactly what the features of an ideal car should be, and to do this is manifestly impossible, as scarcely any two people would agree upon every detail of the catalogue. However, in broad lines it may be said that for all-round work the most suitable chassis is one in which comfort and efficiency are combined as far as possible without interference with each other, and in the 17.9 h.p. Armstrong-Whitworth such a combination appears to have been made with a perhaps unusual measure of success, this being undoubtedly owing to careful attention of both the theoretical and practical functions of almost every detail. Cost of production has received a reasonable, but far from extravagant, amount of attention, while at the same time there are no parts made in an absurdly expensive manner to gain some very small advantage, as is sometimes the case.

The illustration of the engine, Fig. I., is a reproduction from a drawing of the 15.9 h.p., and is therefore not quite true to scale, the cylinder bore of the 17.9 h.p. being 85 mm., and the stroke 120 mm., whereas the smaller car has a cylinder bore five millimetres less. All other parts, however, are the same, and the drawing is otherwise quite correct. Noticeable features of the one-piece cylinder casting are the large free water space round the combustion heads, the four separate exhaust outlets, and the inlet passages leading to two flanges on the offside. The valve chambers are also well cooled, and the method of fitting the valve guide is uncommon. Reference to the transverse section in Fig. I. shows that the casting is first bored to a diameter of $\frac{5}{8}$in., though the valve stems are only $\frac{3}{8}$in. diameter, the reason being that it is less difficult to bore the larger hole quite true to the valve seating than it would be to drill the smaller one. In making the guides the first operation is, of course, to drill them, the outside being afterwards turned true to the bore. They are not intended to be detachable from the main casting, as they are forced into place hydraulically, but they have, on occasion, been removed without injury. Cast steel, with a specification of 63 tons tensile, 60 tons yield, is used for the pistons, which are distinctly light for their size, weighing only 1 lb. 3 ozs. complete with the rings, but without the gudgeon pin. It may be observed that the gudgeon pin receives lubricant from a hole in the upper side of the connecting rod end, and as this method has been criticised on account of the liability for carbonised oil falling from the piston head to stop up the orifice, it ought to be explained that the hole is in reality quite a large slot. It is doubtful whether the drip point on the web has much effect in directing the oil to the desired position, but it can do no harm, and its weight is scarcely sufficient to make its removal desirable on this score.

The gudgeon pin is 11/16in. in diameter and is bushed in phosphor bronze, it being secured by a piston ring, than which there is probably no better method. There are no special points of interest regarding the connecting rod, as it is a stamping of the ordinary section, but the material is of as high a tensile strength as possible (80 tons tensile nickel steel), and the rod is therefore a light one. White metal is used for the main crankshaft bearings, and for the big ends, the crankshaft having a diameter of $1\frac{1}{2}$ins. in each case. Steel of an unusually high strength is employed for the shaft, it being from 90-100 tons tensile. The total length of main bearing is $7\frac{1}{2}$ins., and each big end bearing is 2ins. wide.

An unusual form of construction is adopted for the camshaft, as duplex loose cams are used, as can be seen by reference to Fig. I. These cams have a profile with very slightly concave flanks, and being ground to jig, the relative positions of the exhaust and inlet setting for each cylinder cannot be affected by the pinning to the shaft. The latter is first provided with parallel holes drilled to jig, and the loose cam pieces are prepared similarly. To attach them to the shaft a parallel pin is placed through one hole while the other is being reamered taper, and the first hole is not reamered until the taper pin for the second has been driven home. This method of construction seems to combine the advantages of both solid camshafts and shafts with single loose cams, while there is no probability of inaccuracy in setting as long as the jigs are sufficiently good and they are by no means complicated. The tappets and valves require no explanation beyond that given by the drawings, except perhaps that each tappet head is faced with red fibre. Fibre is also used for the large timing wheel, it being of a hard pale yellow variety, which is not affected by oil. We understand that the discovery of this material has been a matter of some difficulty, but it is at least interesting to know that such a substance can be obtained. It will also be observed that its strength is considerable, as the timing gears are only an inch wide, and the larger of the two is all fibre, not being strengthened by a metal segment.

The lubrication system is explained partly by Fig. I. and partly by Fig. III. Oil is forced to each of the main bearings of the crankshaft by a pump, and passes to the big ends through large passages in the shaft, shown dotted in Fig. I., and the unusual diameter of this oil passage is worthy of careful consideration, because there is no doubt that choakage is almost impossible, while at the same time a free supply to all bearings is made certain, without the use of high pressure. The gudgeon pins depend upon spray for their lubrication, and appear to perform with complete satisfaction. The peculiar feature of the lubrication system is the adjustment supplied, whereby the pressure of the oil in the supply pipe can be regulated. The pipe itself is copper, and is cast inside the aluminium crankcase, together with the leads to each main bearing. At the bottom of the pump there is a hollow set-screw adjustment for a spring, controlling a ball valve, which leads back to the intake side of the pump. Moving the set-screw alters the size of the pressure on the by-pass, and when it is screwed right home all the oil must go to the bearings. If under such circumstances it is found that the engine smokes, then by slacking back the screw the pressure is lessened, and it is intended that the adjustment should be set for the particular oil being used, and then left alone. There is no doubt that this adjustment would be valuable to the makers, and to a user, if employed intelligently, but there is a small danger

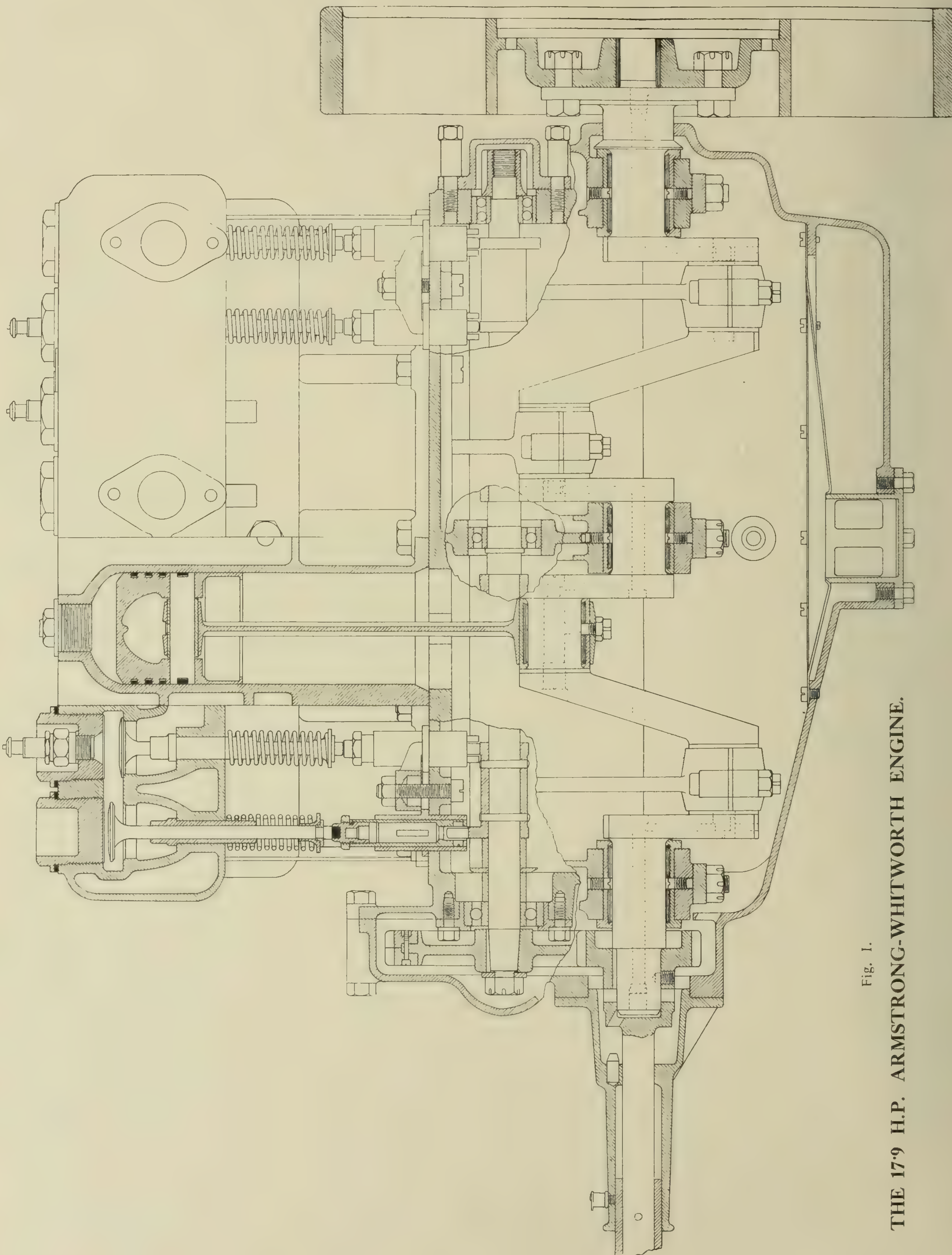


Fig. 1.

THE 17·9 H.P. ARMSTRONG-WHITWORTH ENGINE.

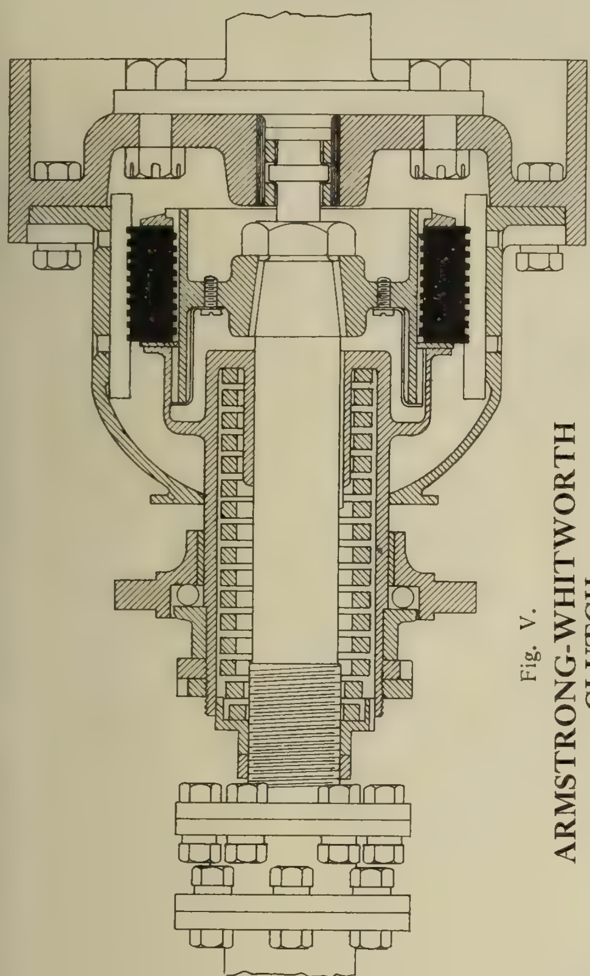


Fig. V.
ARMSTRONG-WHITWORTH CLUTCH.

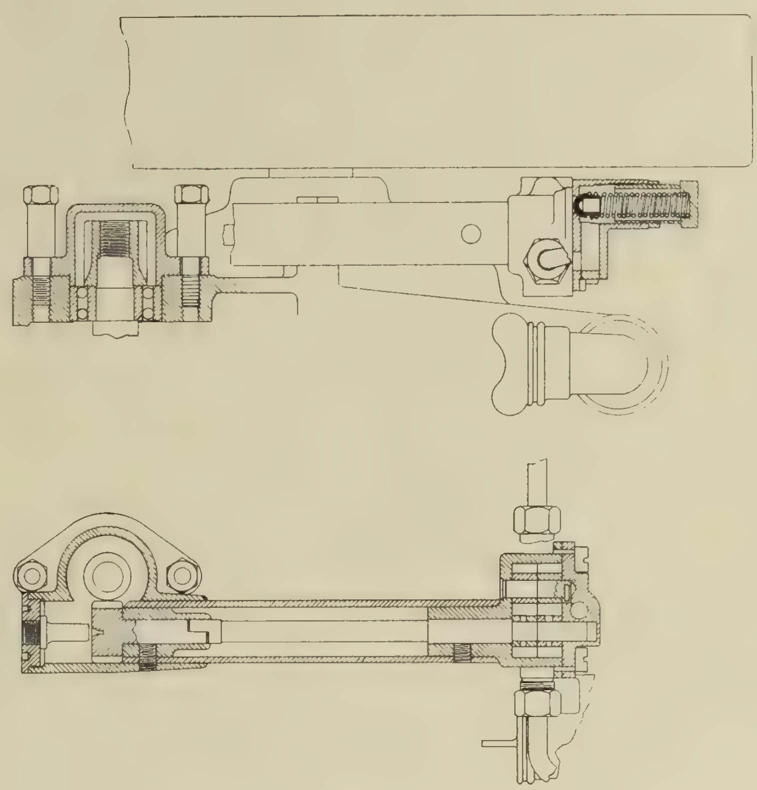


Fig. III.
ARMSTRONG-WHITWORTH OIL PUMP.

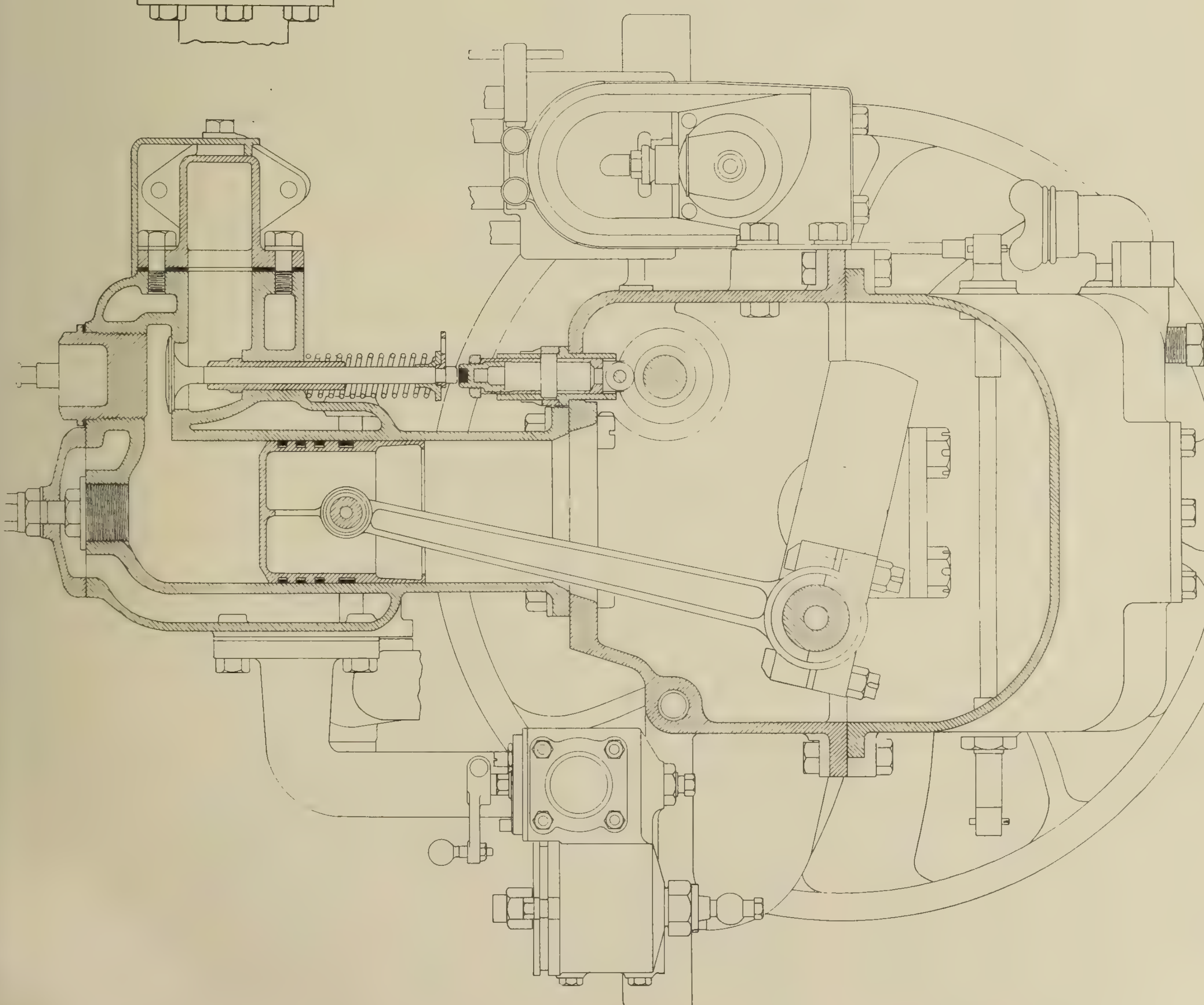


Fig. II.

that the bye-pass might be opened too widely in a zealous effort to stop smoking with thin lubricant. However, the bearings cannot fail to receive enough lubricant, even if the by-pass is fully open, because the compression cannot be taken off the spring within the limits of the adjustment, so the advantages of the adjustment more than compensate for its small danger. The pressure normally in use is about 5 lbs. per sq. inch. The pump itself is the ordinary gear pattern, and is fitted at the rear left hand lower portion of the crankcase. It is driven by a vertical shaft by a skew gear attached to the extreme end of the camshaft, and there is a break connection to enable the pump to be removed easily. There is double filtration of the oil, as on falling to the sump it has first to pass through a gauze grid, and, secondly, to go through another cylindrical gauze surrounding the intake pipe, seen in the centre of the bottom of the crankcase in Fig. I. There is a Bourdon gauge on the dashboard in connection with the main supply pipe, which, to some extent, indicates the condition of pressure in the bearings.

Natural circulation of the cooling water is deemed to be sufficient. We have already described the large size of the water spaces in the cylinder's jacket, and the proportions of the piping may be observed in the side elevation of the chassis, Fig. IV. A honeycomb radiator is used, and a special point in connection with it is the hinged filler lid, an undoubted convenience which is common to all chassis emanating from the Armstrong works.

As regards the gas supply, the standard carburettor is a White and Poppe with both hand and foot throttle control. The ignition is performed by a dual magneto equipment, and the way in which the control connection to the contact maker is carried through the crankcase should be noticed, as it removes otherwise necessary long rods and many joints.

The principal point in connection with the engine which is most obviously open to criticism is the three bearing crankshaft, but it must be admitted that it has many manufacturing advantages in comparison with a five bearing arrangement, and for small sizes of engines—say up to 90 mm. cylinder bore—the three bearing design seems to be quite satisfactory, providing the crankshaft is sufficiently rigid, which it is in this particular instance. Another point is that the oil pump is none too accessible when the shield is in place, but it is no worse than the vast majority of its kind, and is better than many, inasmuch that it can be detached from the crankcase entirely, as its only connection thereto consists of a couple of flanges with a pair of bolts apiece. There is no thrust bearing to support the drive of the skew gear at the end of the camshaft, but the double row ball bearing probably has sufficient axial rigidity for this purpose.

Fig. V. shows a section of the plate clutch, which possesses one or two peculiarities. There are fifty-two thin spring-tempered carbon steel plates, which, with their pressure control, are explained fully by the drawing, and are not different from the ordinary, but the clutch as a whole is supported by what is practically an extension of the main gearshaft, instead of by the crankshaft, and there is provision for a very small amount of universal movement within the flywheel, the forward end of the clutchshaft bearing a collar, which fits loosely in the flywheel bush. This means that if the engine and gear box are not quite in line there must be movement of the plates over each other, and that the need for a fully universal joint to the gear box has not made itself apparent is testimony to the stiffness of the frame and the careful original alignment. Cast steel is used for the outer, and a nickel steel stamping for the inner member, the plate keys being

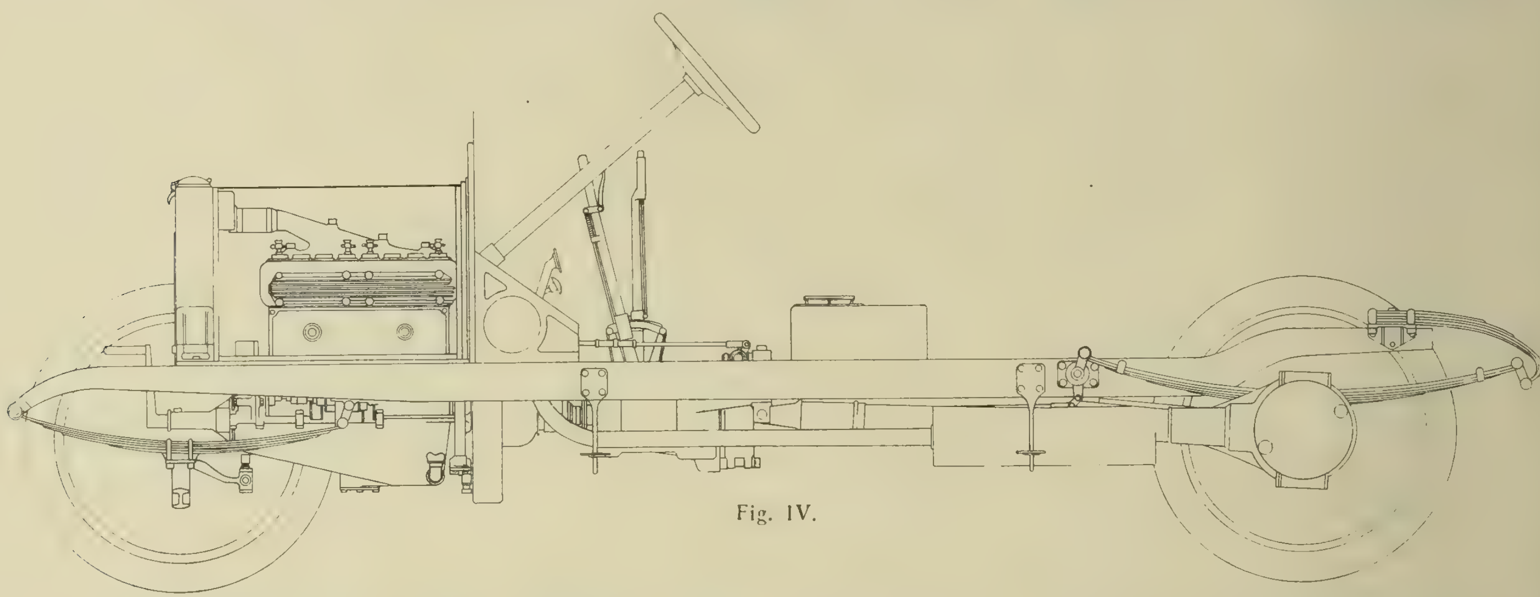


Fig. IV.

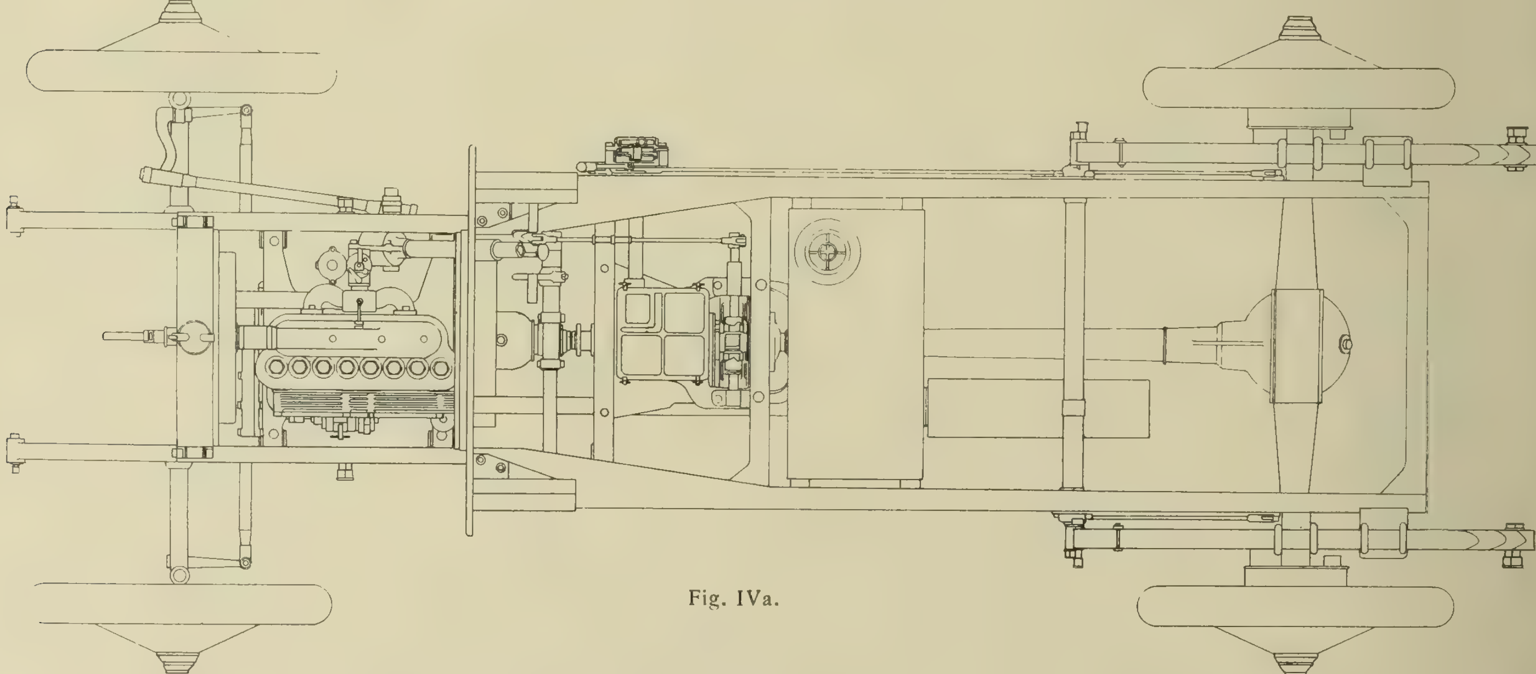


Fig. IVa.

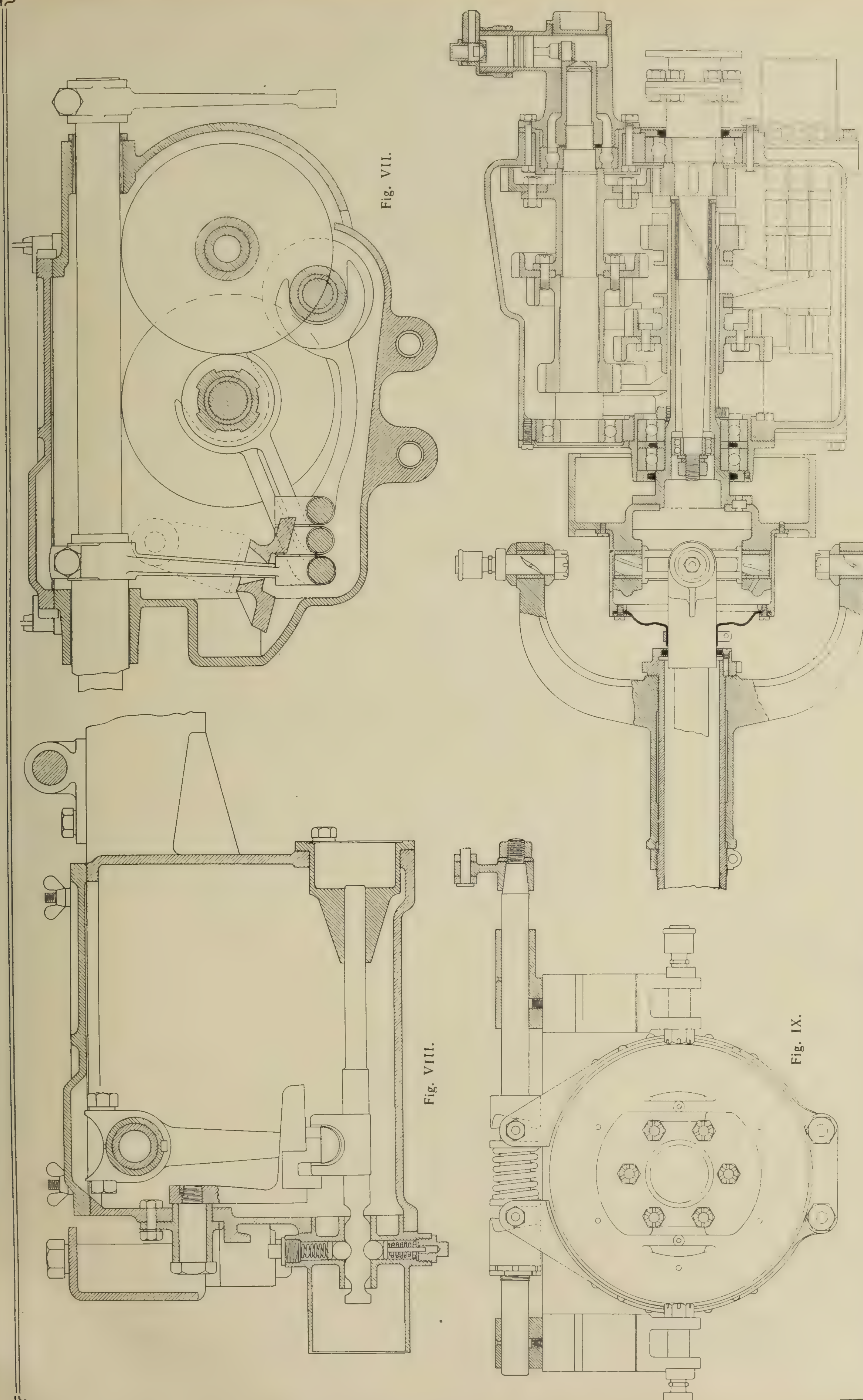


Fig. VI.

Fig. VII.

Fig. VIII.

Fig. IX.

THE ARMSTRONG-WHITWORTH TRANSMISSION.

riveted to the case, so enabling the whole of the inside of the part to be machined, while the central piece of the flange coupling allows the whole clutch to be detached without removing the gear box.

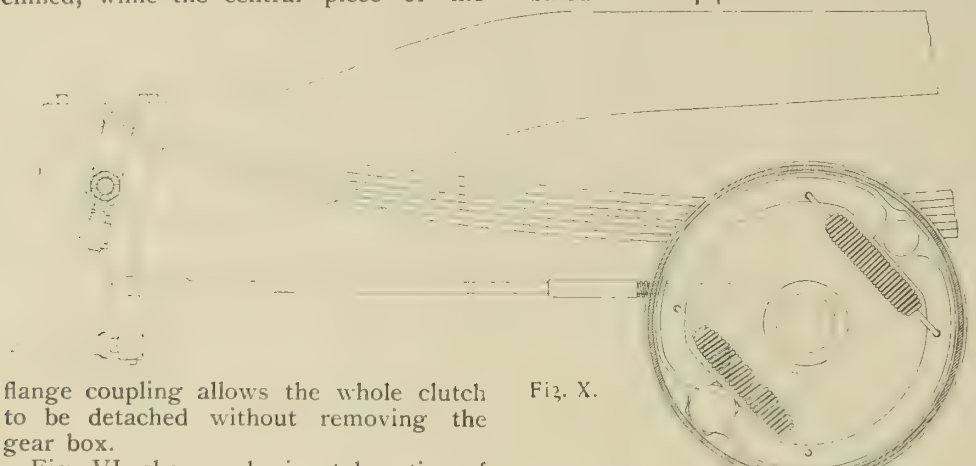


Fig. X.

Fig. VI. shows a horizontal section of the gear box in the plane of the shafts, Fig. VII. a vertical section in the plane of the main shaft, and Fig. VIII. a transverse section at the centre of the striking gear.

In order to secure the utmost possible rigidity for the main shaft, the usual spigot is extended right through, and ends in a ball bearing in the same plane as the usual outside tail end bearing: this should almost entirely do away with any possibility of whip. The box is not split, as the shafts are assembled from the ends, and the bearings are gripped in brass housings bolted to the aluminium. Each bearing is well trapped for oil, and the ends of the striking rods are also enclosed in a brass box. The gears are made to three and a half module pitch, and are of steel with a tensile strength of about ninety tons, which is naturally distinctly hard. After being milled to within a fine limit of their finished tooth form, hardening takes place, and the teeth are afterwards trued up in a manner which we are not at present permitted to disclose. A very low reverse speed is given, the ratios being: Reverse 5.35 to 1, first speed 4 to 1, second speed 2.61 to 1, third speed 1.56 to 1, and fourth speed direct. The small pump seen at the front end of the layshaft is the air pump for providing pressure for the fuel tank, and the adoption of this position makes the drive a very simple matter, while it calls for the use of only a short pipe to the tank, and this does not interfere with any other part. There is, of course, the objection that the engine could not be run for a very long time with the clutch out without resource to the hand pump, but this is really of small moment, and it would seem to be a good plan to relieve the crankcase of yet another attachment, particularly one combined with a pipe line.

The foot brake construction is shown in Fig. IX., cast steel being used for the drum, and the construction of the universal joint within it allows an exceptionally large amount of lubricant to be contained. The absence of a bearing from the front end of the propeller shaft is a little uncommon, as the majority of cars with the enclosed type of shaft are so provided. Of course, there is no need for support to the shaft at this point as long as the torque tube is sufficiently rigid and sufficiently accurately slung. In this case the tube is amply strong to resist bending, being $2\frac{1}{2}$ins. outside diameter at the smallest part, and having walls $\frac{1}{4}$in. thick, while the fork is swivelled on bearing pins secured to the portion of the gear box casting which is bolted up to the frame, and thus completely accurate centring is easy. Any twisting of the axle is compensated for by the lubricated bearing between the fork and its tube, while the length of the latter can be set when erecting, by means of the adjustment shown.

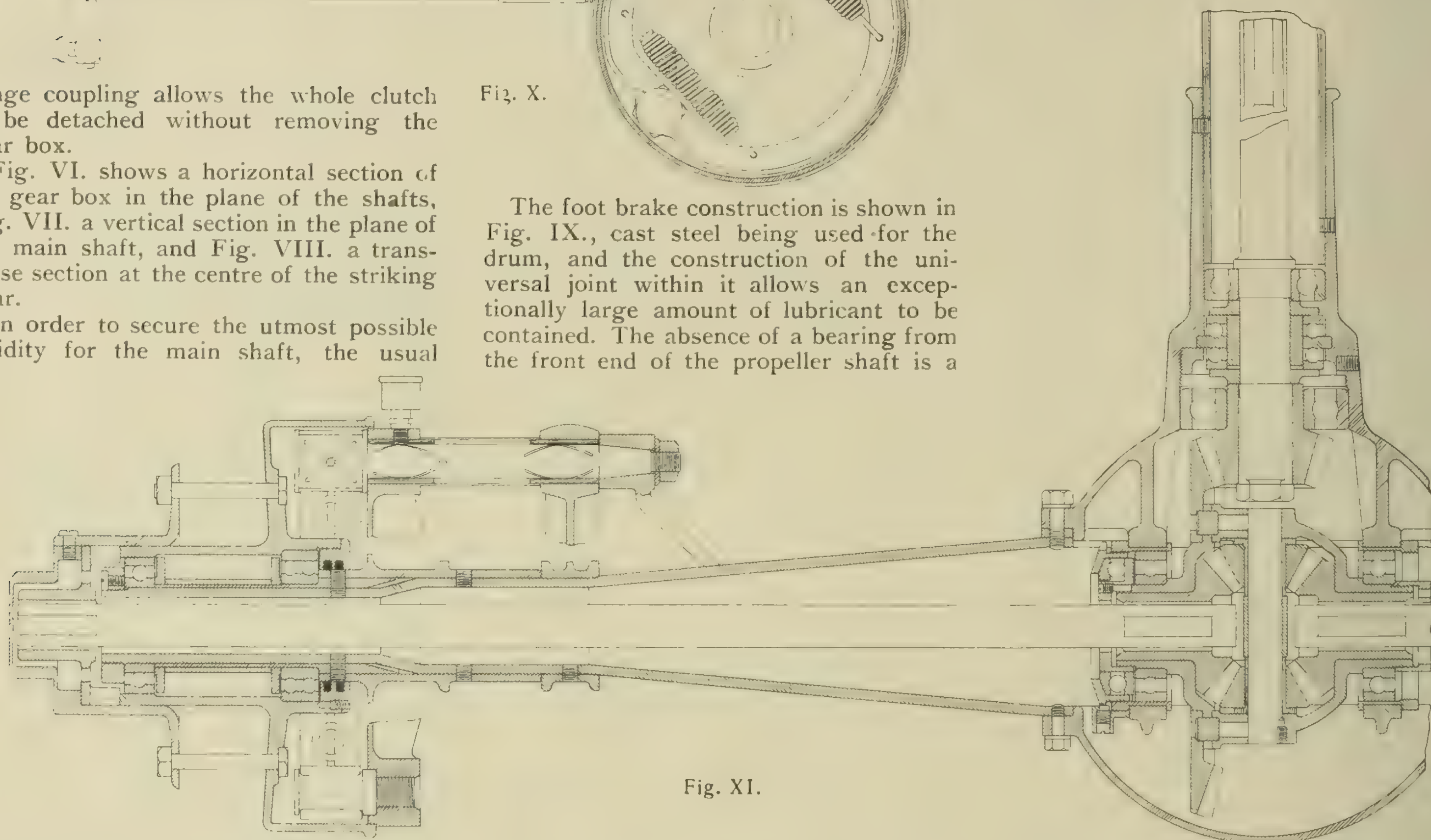


Fig. XI.

The change speed is controlled by a sliding gate, the hand lever being attached to a rod fitted in phosphor bronze bearings inside a tube projecting from the gear box, and is quite independent of the hand brake lever. The form of striking gear is shown in Fig. VI., but the chassis plan in Fig. IV. shows the gate arrangement, and also the method of setting the box in the frame.

The back axle casing is made from two steel pressings, of material which is, roughly, $\frac{1}{4}$in. thick, by the division being longitudinal, and the halves welded together. Cast steel is employed for the

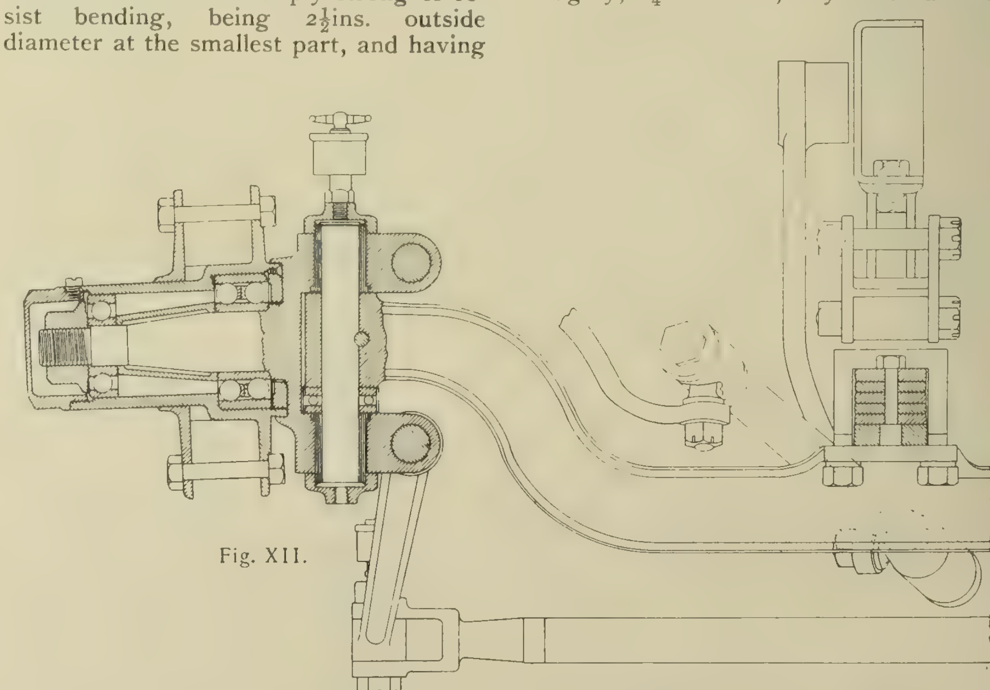


Fig. XII.

part which carries the differential and the driving bevels, and as all the bearings are fixed to this one piece, no portion of the pressed axle requires to be machined inside. Five module teeth are used for the driving bevels, which are cut on a Bilgram machine, hardened, and finally ground in by being run, in alternating directions, for some hours, and fed meanwhile with a mixture of fine carborundum powder and oil. There are only two differential pinions, but they are well above the average of size, having six module teeth—in fact, the whole of the driving gear is a particularly strong job. One ingenious feature is the way in which the thrust bearing is carried in the same housing as the journal, so that the thrust can be adjusted on the bench by the detachment of the aluminium back cover plate of the axle and the removal of the caps securing the journal housings. The driving shafts can, of course, be withdrawn from the hubs.

The castings which carry the hub brakes, and on which the loose spring pads are mounted, are pressed on hydraulically, and the brake mechanism is fully explained by Figs. X. and XI. In the hubs the double bearings on the inside are relied upon to resist all axial thrust, and, though a thrust washer would probably improve the durability a perceptible amount, still the journals are very well mounted, and should be entirely protected from water by the double felt washers. The ratio of the bevels may be either 3.1 to 1, or 3.7 to 1, and the standard wheel diameter is 815 mm.

The chief point of interest with respect to the front axle, Fig. XII., is the precise inclination of the steering pivot. If produced the centre line will be found to intersect the ground line at a distance of 2½ins. inside the theoretical point of contact of the tyre. We understand that this inclination is the result of experiment with different settings varying from the upright to that in which the centre line coincides with the tyre contact on the road. Undoubtedly the steering is excellent, and the car holds the road at high speed on curves in an almost remarkable manner, but it cannot be believed that the pivot setting is alone responsible. Fig. XII. also shows the way in which a bent steering connecting rod is avoided by twisting the steering lever. The worm box, with its complete wheel and square-ended shaft for the arm, was illustrated in the issue of last August.

The general arrangement of the springs can be seen in the views of the chassis, the front springs being normally 33ins. from centre to centre, and the rear springs 45ins. with a camber of about 2½ins. Each of the shackle bolts is hollow, and carries a grease cap fitted over the head, while the springs are all clipped, to damp their movement somewhat. The hand brake gear shows evidence of careful thought, as the cross shaft is carried inside the tubular cross member of the frame, there being a greaser to each of its bearings. Exceptional strength is claimed for the frame, which has four pressed and one tubular cross member; pressed brackets, riveted to the main side members, support the crankcase arms, and the mounting of the gear box has already been described.

It has previously been said that the car behaves very well on the road, and it may be well to particularise its specially noticeable characteristics. The engine accelerates rapidly, pulls well at low speeds, and is capable of revolving at well over two thousand per minute, yet at the same time it is quite smooth running, which is probably owing to the lightness of the reciprocating parts. The springing is good both as judged from the front and rear seats, and is exceptionally good when rounding corners, while the car appears to hold the road quite satisfactorily despite bad surface and high speed. Absence of chassis noises is noticeable, but the engine is not unusually quiet, though it is certainly quite as quiet as the average of well-made engines.

As an engineering job, the chassis is, of course, expensive, and it is rather surprising that it has not been found better to use a worm gear axle drive instead of taking such elaborate pains to quieten the bevels. As regards material, the stampings are of unusually high tensile steel, and most of the other steels are of exceptional hardness. No malleable cast iron is used, the only cast iron part being the cylinders. The details of the small part lubrication and of the control, etc., are very well designed.

MACHINING POPPET VALVES.

Showing the saving effected by present-day methods as compared with former practice.

POPPET valves are usually of high percentage nickel steel, and the evolution of their manufacture is an interesting example of progress in present-day machine-shop practice and methods, as compared with a few years ago. In fact, one may assert safely that a valve costing between five and six shillings in labour and material by former methods can now be produced, with an equal degree of accuracy, for less than 25

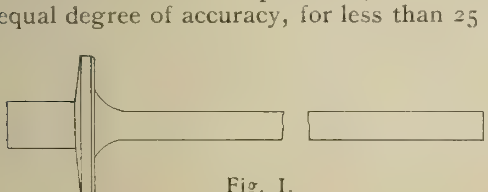


Fig. I.

per cent. of that figure. Against that, however, has to be debited an expensive equipment of tools and heavier overhead charges in the shape of superior shop organisation. The reduction in the cost of production is chiefly due to two things—the substitution of accurate drop forgings, as Fig. I., for the old hand forgings, shown in Fig. II., and the gradual development of the capstan lathe, which is now used for turning these stampings up, whereas they were formerly done on an engine lathe, the latter being a very slow, and consequently expensive, method.

When hand forgings were used it will be understood that a large amount of material was left on by the smiths, which meant considerable time being spent on removing it by the turner, as heavy cuts cannot be taken, owing to the shape and slender proportions of the work. The first operation was to centre the forgings at each end, then put a carrier on the bottom end, and operate on the head by first turning over the top approximately to size. Next the radius on the outside end was put on, which often meant the removal of a considerable amount of material, and this being completed, the inside face and radius, and a straight portion about 1in. to 1½in. long, along the stem, were finished to within about .005in. of final size, that amount being left on to obtain an accurate finish by grinding. The working out of the radius was a painfully slow operation, and required great care and skill on the part of the operator to complete it satisfactorily. When thus far completed the forgings were turned round in the lathe, and the carrier secured on the turned part of the shank, while the straight stem was turned

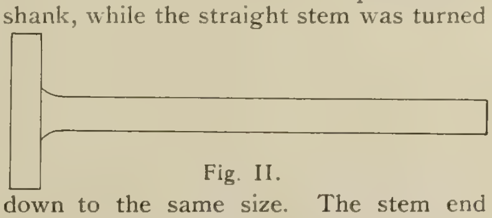


Fig. II.

down to the same size. The stem end was now faced up to length, and the bevel seat turned on, it being particularly important that the distance from the seat to the stem end should be exactly alike on every valve. Turning up of valves by the methods here described not infrequently took from two to two and a half hours each when done in quantities of from twelve to twenty-five. They are now produced in much larger quantities, of course, owing to the enormous growth of the automobile industry, and this fact also does much towards reducing the cost of production.

Present-day conditions have altogether changed the condition of affairs, drop stampings now being universal for this work, and when forged to the shape shown at Fig. II., being very near the finished size, very little material remains to be removed. The forgings are first gripped by the shank, in the automatic chuck, or spring collett, shown in Fig. III., which is of the type common to all hol-

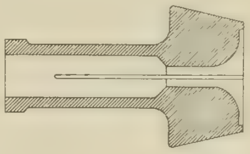


Fig. III.

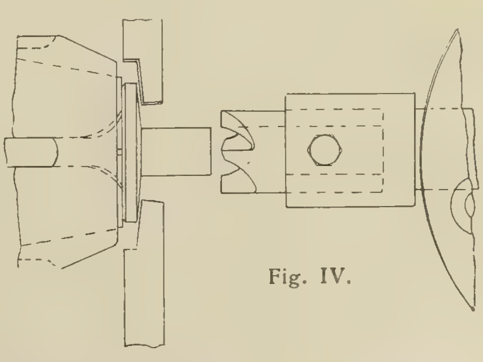


Fig. IV.

low-spindle capstan lathes equipped with automatic bar-feed mechanism. The spigot is first run down with a hollow

mill, which is secured in a holder, in turn fitting in one of the holes in the capstan

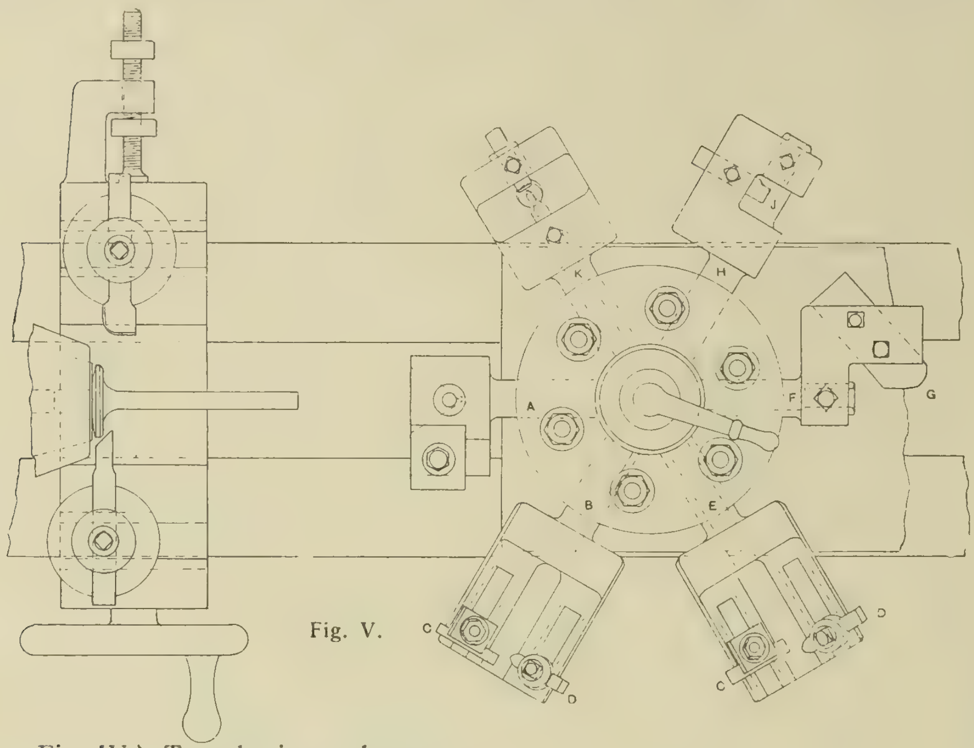


Fig. V.

(see Fig. IV.). Two shaving tools are secured in the tool posts on the cut-off rest, one being the roughing tool, and the other dius, face and periphery to size, the spigot being meanwhile steadied by a the finisher. They machine up the ra- bush in a holder in the capstan to give the work additional rigidity. The time for the above operation complete, including centring the end of the spigot with a centring drill in the capstan, should not occupy above six minutes when the machine is set up for completing a quantity. Setting up time will occupy not more than thirty minutes on a simple job similar to the one involved, and when this time is divided up between a hundred or more valves completed at a time it is infinitesimal. After the ends have all been turned, radiused and centred they are turned round and gripped on the turned spigot, by the automatic chuck, as shown in Fig. V., which is a plan of the capstan lathe, chuck, cut off or cross slide, and capstan or turret tools in sequence. Automatic power feed should be provided for the capstan, and automatic stops to trip at any distance along the working stroke of the capstan. After the work is turned round the first operation is to face up the end of the stem, with the bevel starting tool, in position A in the capstan, so as to enable the hollow shanked roughing box tool B to get a true start. C shows the adjustable vee steadies, which should be set immediately behind the cutting tool D. Position E shows the finishing box tool, which is identical with B, and finishes the straight shank down to within .005in. or .007in. of finished size. Tool G in position F is for rough turning the face and radius as shown, the stem being steadied by the bush shown for the purpose whilst the shank passes up the hollow shank of the tool holder. This obviates any tendency on the part of the stem to spring. Tool H. is shown at J, for finishing the valve stem end to length, and putting a slight bevel on the corner, and the stem is also steadied in the bush shown on the outside whilst this operation is being performed. The only remaining operation on the stem is to centre the end for subsequent operations, such as grinding, and this is done with a Slocombe centring drill, as shown at tool K, the shank being steadied by a bush as before. The drill is withdrawn, the spindle end being still steadied in the bush, whilst the finishing tool in the cut-off slide just takes a finishing cut off the face and radius, and the seating tool puts on the bevel seat, which forms the valve face. These two tools are controlled

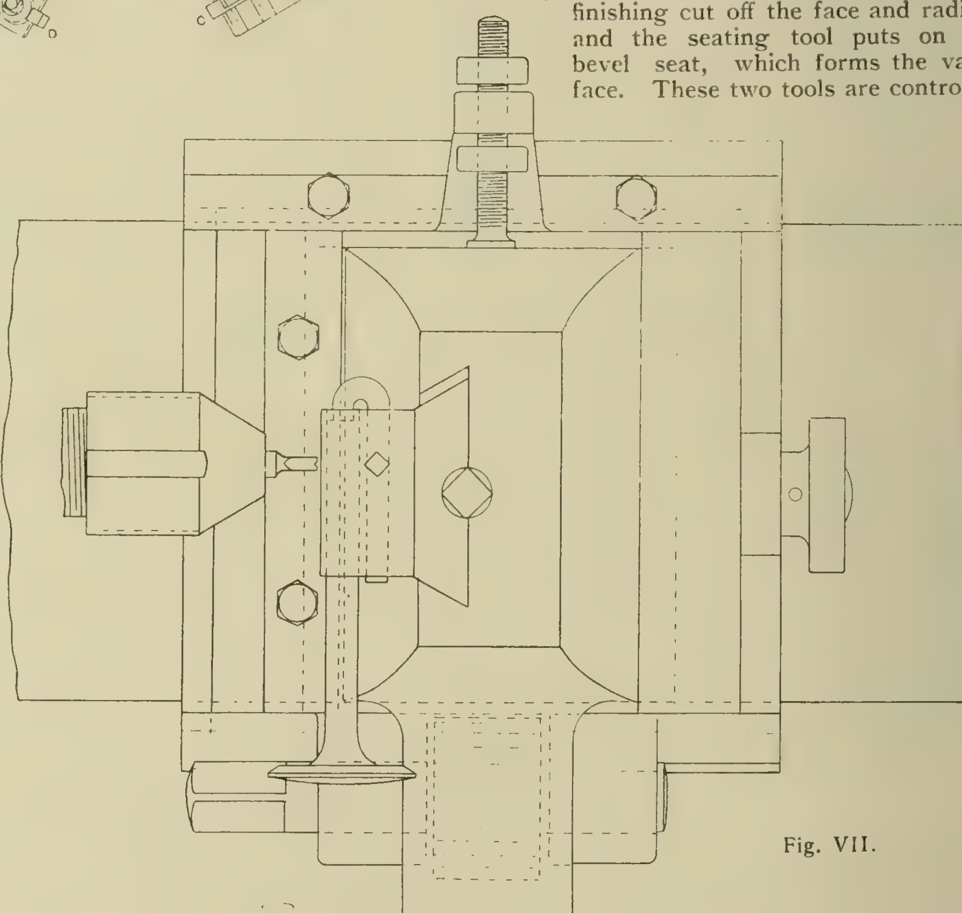


Fig. VII.

by the adjustable stop, shown in the usual position at the back.

The time for this operation, with proper equipment and tools, should not exceed twenty minutes.

The valves are now passed on to the grinding department, and are ground on the seat and stem in a cylindrical grinder, which will occupy another twelve minutes, and an interesting and intricate operation now presents itself, namely, the cutting of the small cotter hole in the spindle. For this job the small home-made slot drill shown in elevation in Fig. VI. (and a plan of the work-holding device in Fig. VII.), is very much superior to the heavy slot drills often used, or the old-fashioned method of drilling four or five holes in a sensitive drill, and then drifting the surplus material out, which is a very indif-

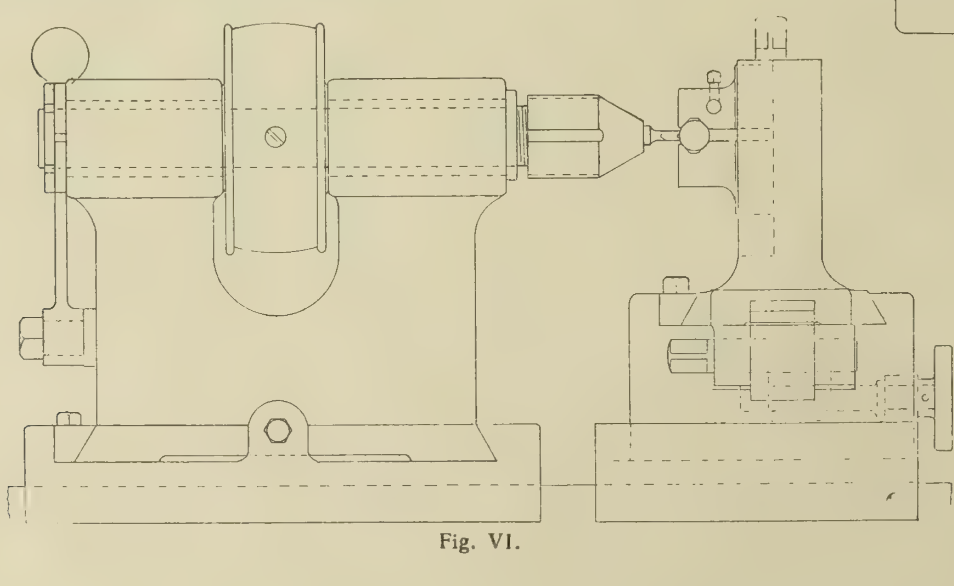


Fig. VI.

ferent piece of workmanship. This device is cheap to construct, and turns out accurate work rapidly, as it is exceedingly sensitive, can be run at a very high speed, and is worked by boy labour.

Referring to Fig. VI., there is a cast-iron bed of suitable box section planed up on the top and sides, and to it the headstock baseplate and work slide are clamped. The headstock for carrying the cutter spindle can be locked in any position in the cross slide, and there is a locking lever with a notch capable of engaging in one of the slots milled across the lock-nut on the spindle end; this holds the spindle while the chuck is slackened or tightened. There is a longitudinal slide superimposed on the transverse work slide, and fed forward by a screw and knurled disc. A screw, with a right and left-hand thread, is used for operating the two jaws, which carry the work dead central, and there is an adjustable stop for setting the work to, as shown in Fig. VII. Transverse feed is obtained by a rack and pinion, controlled by a hand lever and a pair of adjustable stops. Both the handle on the slide and the feed screw are within easy reach of the operator, and with this device a lad can easily complete five cotter holes per hour, which is very much quicker than the time taken by any of the expensive automatic machines I have yet seen for this purpose.

The spigot, which is forged on for gripping purposes, can now be parted off in

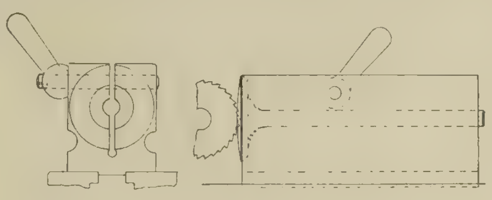


Fig. VIII.

the capstan lathe in exactly the same manner as shown in Fig. IV., a parting-off tool $\frac{1}{8}$in. wide being held in the cut-off rest front tool post, a form tool, in the back tool post, shaving the head to size, and putting on the proper contour. This operation will occupy another four minutes.

The only machine operation now to complete the valve is to mill the slot in the head, for grinding-in purposes, as shown at Fig. VIII. A small Lincoln type milling machine is very suitable for this purpose, and the valves are gripped in the fixture as shown while the table is fed up to the cutter by hand. A dead stop ensures the slot being sawn the correct depth each time. As is shown in Fig. VIII., the valve is held in a cast-iron block, which is slit and provided with a clamping screw, the block itself being tongued to clamp down on the milling machine table.

The total operation should not exceed three minutes, including all setting up and clamping times.

Assuming that good supervision be given in setting up the machines, the whole of the operations here involved, with the exception of the grinding, can be performed by youths, and do not call for a high degree of skill. With the grinding operation higher skilled labour is required, as great accuracy is necessary for the seat and spindle. However, the total labour costs should come out at about 3s. 3d. to 3s. 6d. per dozen, and this is a striking contrast with former methods, the total time expended on the job, as above described, being under one hour for each valve, including all setting-up times.

D. WALTERS.

CARBURETTOR ACTION.

Being a paper read before the Institution of Automobile Engineers.
By W. Morgan, B.Sc., and E. B. Wood, M.A. (Cantab.)

THE work on which the following results and conclusions are based was undertaken at the Daimler Motor Works some three years ago as preliminary to an attempt to design a paraffin carburettor suitable for traction work. The difficulties of the task were twofold, the low volatility of the paraffin giving rise to one set of problems, while the narrower limits of smokeless and odourless combustion, as compared with those of petrol, intensified the difficulties met with in an ordinary carburettor of obtaining suitable fuel mixtures over the wide range of demand required by a modern road engine. It was therefore necessary to find or design a carburettor which could give a mixture of fairly constant composition. Investigations which had been made on carburettor action assumed the steady flow of both air and liquid fuel, with apparently misleading conclusions when applied to designing a carburettor for the actual engine.

Dr. Watson, in his paper on "Thermal and Combustion Efficiency" (Proc. I.A.E., 1908-9, pp. 387-473), shows the character of the pressure variation in the induction pipe of a four-cylinder petrol engine at a speed of 656 r.p.m. with open throttle, see Fig. IX., which is copied from Dr. Watson's paper. In this particular case it appears that actual reversal of flow takes place, so that any argument based on steady flow must be regarded with suspicion.

It was decided to ignore as far as possible all formulæ relating to steady flow, and to measure directly the related quantities of gas and liquid induced under the engine suction. In the first series of experiments, as will be seen, the writers fell from grace, and estimated from pressure the amount of exhaust gas dealt with. The possible danger of error was recognized, but as the necessary apparatus for directly measuring the volume of air was unobtainable, and the correctness of the conclusions based on this method was confirmed by gas analysis, no misgivings were entertained as to the general accuracy of the results.

Last year it was found possible to lay down a more complete equipment at the Merchant Venturers' Technical College, Bristol, and to confirm the results of the previous work.

The outfit at the Daimler works consisted of a standard four-cylinder engine, 124 mm. bore by 150 mm. stroke with tappet valves, driven through a belt and speed cones by a two-phase motor. The engine was fitted with a graduated petrol gauge, and the exhaust passed into a small tank of about two cubic feet capacity perforated with several $\frac{1}{2}$ in. holes; the pressure in this tank was read on a suitable gauge, and converted into volumes.

Fig. I. is diagrammatic representation of the apparatus employed in the later series of experiments. The motor was an 8 h.p. D. C., and the speed regulated by inserting resistances in the armature circuit. At first it was found impossible to maintain constant engine speed, as the temperature of the resistance coils varied greatly in use. Fortunately the resistance was one day overloaded and fused, and a water resistance installed in its place. With this great constancy of speed was obtained, and over a ten minutes' run a speed variation of less than 0.4 per cent. could be maintained.

The petrol gauge shown diagrammatically in Fig. II. enabled accurate measurements of 100, 200, or 300 cc. of petrol to be taken, and was designed with the object of enabling readings of petrol consumption to be taken with an engine running under its own power, without stopping from time to time to refill the gauge. The threeway cock enables the gauge to be filled from the main tank without disturbing the supply to the engine. When the gauge is filled the cock is set so that the engine alone is in communication with the tank, a further movement of the cock cuts out the main tank and connects the gauge to the engine. This has been found a most convenient form of apparatus both for road and bench work. Used with an engine which is driven by an electric motor, as in the present case, it is equally convenient, as the petrol level in the float chamber remains practically constant with the gauge in or out of action. One precaution was found to be necessary in using this instrument: the pressure of the petrol when fed from the gauge chamber differed from that when fed from the main tank, causing a slight difference in float level in the two cases. To prevent error the gauge was always filled some 20 cc. above the upper constriction, so that by the time the petrol had fallen to this point the float level had adjusted itself; the average pressure employed was $1\frac{1}{2}$ lbs. per square inch. The passage of the petrol past the constrictions of the gauge was sharply marked, so that readings of 100 cc. in 20 seconds could be taken correct to 0.2 second; as a rule, much longer periods were taken.

The carburettors used in the first experiments varied in form, but the plain tube carburettor used at Bristol will serve as a type of the others. The section is shown in Fig. III., and it is seen to have consisted of a gas cock in a straight tube fitted with bushes of various diameters, and standing over a petrol jet which stood inside the bush. Pressures at the jet were read by means of a fine brass tube fixed near the jet as shown.

CHARACTERISTIC CURVES OBTAINED FROM DIFFERENT CARBURETTORS.

The vertical scales show the cubic feet of mixture passing through the engine per minute, and the horizontal scales give comparative parts of petrol per minute.

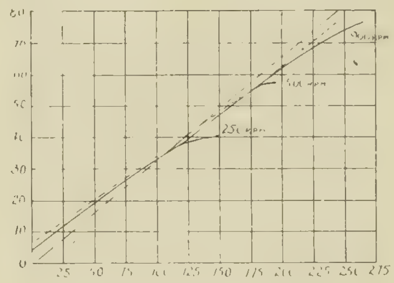


Fig. IV. Characteristics of plain tubes.

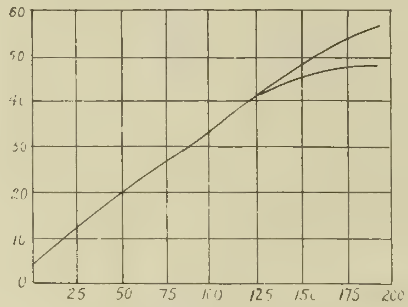


Fig. XI. Effect of back pressure.

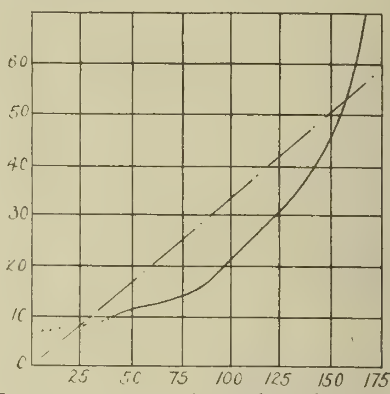


Fig. XVII. Single jet with spring controlled air and fixed throttle.

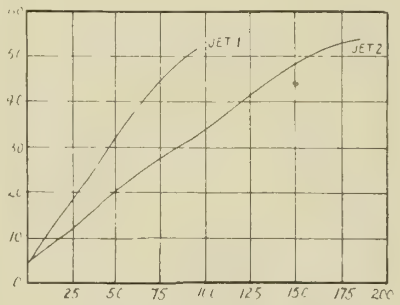


Fig. VI. Effect of jet size.

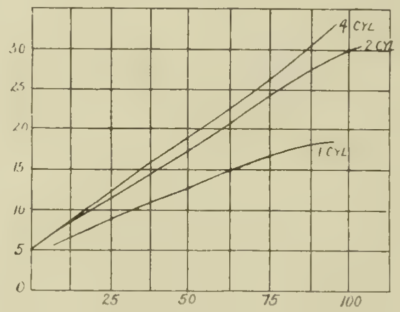


Fig. XII. Effect of number of cylinders.

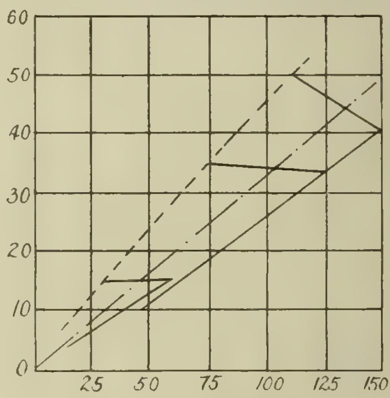


Fig. XVIII. Single jet.

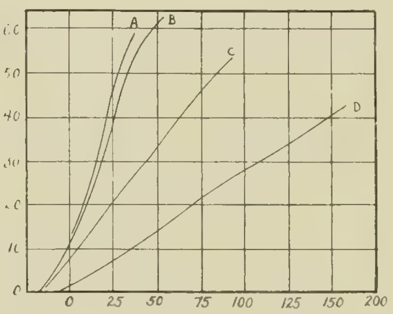


Fig. VII. Effect of throat size.

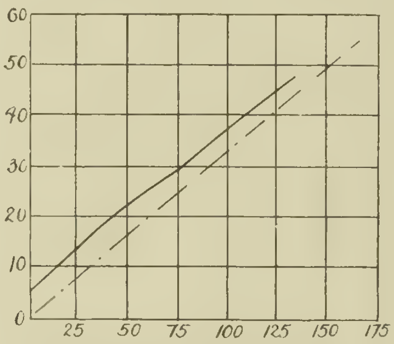


Fig. XIV. Effect of extra petrol supply.

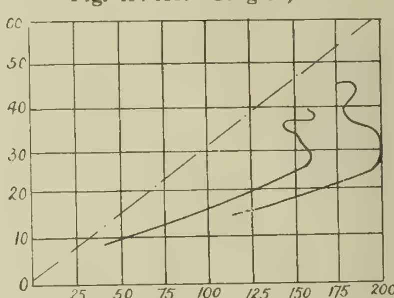


Fig. XIX. Two jets with mechanical extra air.

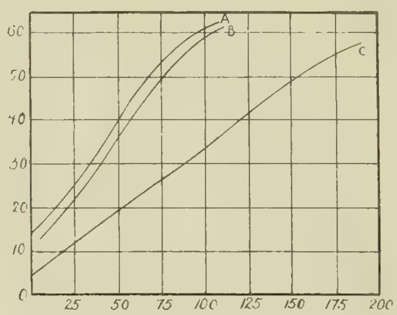


Fig. VIII. Effect of throat size.

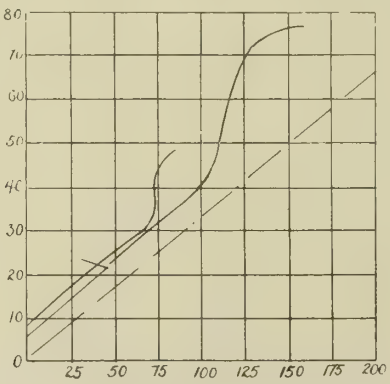


Fig. XV. Single jet with mechanical extra air.

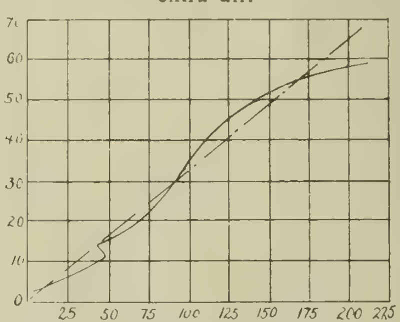


Fig. XX. Two jets with variable choke on one jet.

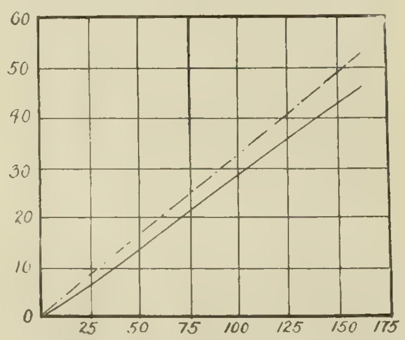


Fig. X. Graph obtained with silencer fitted to intake.

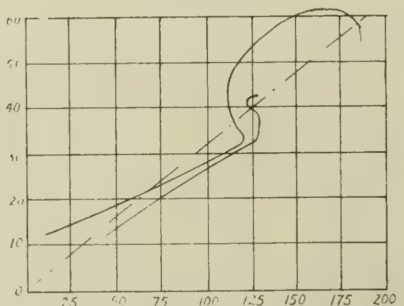


Fig. XVI. Single jet with spring controlled air.

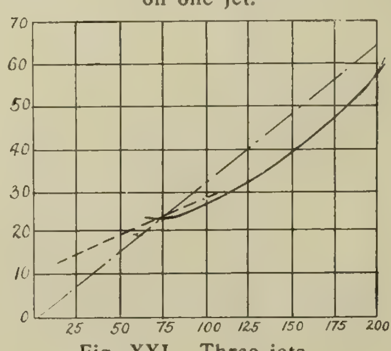


Fig. XXI. Three jets.

The earlier carburettors used were simply lengths of $1\frac{1}{4}$in. copper pipe swept horizontally, so that the air flowed across the jet, and not along it.

In the later experiments the air was measured by observing the time required

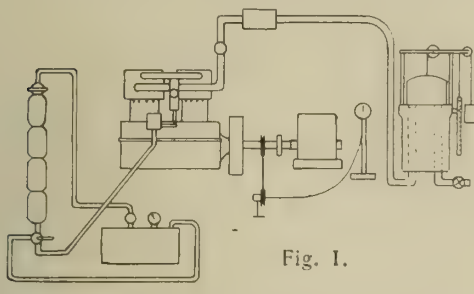


Fig. I.

to pass 30 cubic feet into the gasholder, observations of temperature and barometric pressure being made. In the last series of experiments which were taken to confirm previous work the barometer, temperature, and humidity were so nearly constant that it was thought unnecessary to apply an atmospheric correction which would affect all readings proportionally. In several cases worked out the correction factor was found to be 1.005. Again, no correction has been made for the volume of petrol vapour, as this correction does not affect the main conclusions which follow, as will be shown later.

The gas measurements further required a correction for leakage on the induction and the exhaust side of the engine. On the induction side, beyond making the induction pipe joints gas tight, no attempt was made to prevent leakage. The engine was practically new, and was placed on the bench fresh from overhauling. There was no perceptible leakage past the pistons, but at high vacua there were indications of considerable air flow up the valve guides, although the valve stems were a good fit and were well lubricated. It was decided to allow this leakage to continue, as it was a factor with which the carburettor must deal. Afterwards it was argued that this leakage could have varying relative values with different engines, so that it was desirable to obtain a value for it; accordingly a correction was obtained in the following manner: the engine was first run with closed throttle, and the quantity of air passed into the tank per minute measured; at the same time, the suction in the induction pipe on the engine side of the throttle was taken.

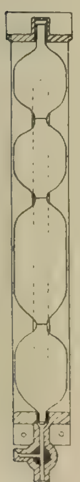


Fig. II.

The throttle was then opened to successive marked positions, with the engine running at a constant speed, and the suction in the induction pipe noted for each position. Since the leakage took place through long and narrow openings it was assumed that the relationship of suction to leakage was linear. Accordingly to every air reading taken with the tank a correction proportional to the suction in the induction pipe was made. The value of this correction is shown by means of the dash line in Fig. IV. In this petrol per minute is plotted against air per minute taken by the engine at different speeds and throttle positions; the plain line graphs are corrected for leakage, and give the relationship between petrol and air which has passed the jet. The dash line is the graph obtained by plotting uncorrected air.

With regard to the correction for losses on the exhaust side, it was found that with clear exhaust pipes the back pressure never exceeded 48in. of water, and with the 500 r.p.m. series of experiments, 24in. The leakage at this pressure was less than one-half cubic foot per minute at open throttle, and fell away very rapidly with closing throttle. No correction was made for this leakage.

The range of speeds over which the experiments were taken was not so great as desired, the electric motor being quite overloaded by the engine, a 124 mm. by 150 mm. Daimler engine with tappet valves.

The results are presented in the form of graphs, which have been obtained from simultaneous readings of petrol and air, as shown in Fig. IV., and those following.

Fig. IV. may be taken as typical of the whole. This is the petrol air graph of the plain tube carburettor at the approximate speeds of 250, 500, 900 R.P.M. with a

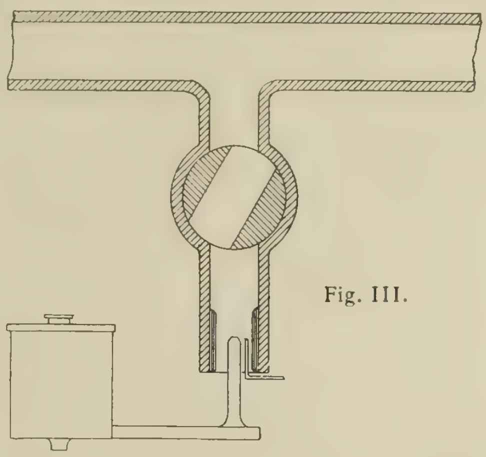


Fig. III.

bush $\frac{3}{4}$in. diameter. It will be seen that the main parts of the superimposed curves lie on a straight line, and that the upper end of the graph for each speed droops towards the petrol axis. Neglecting the drooping parts of the graphs for the present, it then follows that the law connecting petrol and air flow in a carburettor of this type is of the form—

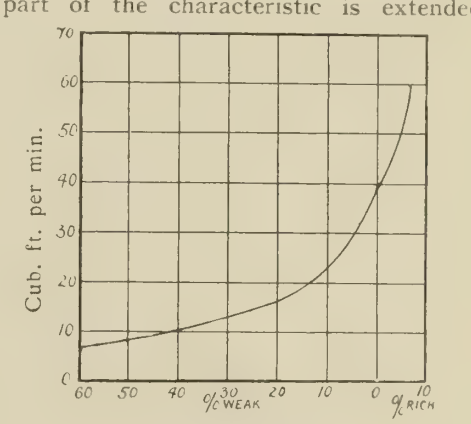
$$y=a+bx$$

It may be said that this conclusion is borne out by every similar experiment made by the authors on various engines, which include four old type Daimlers, two new type Daimlers, one Talbot, and a Darracq.

Accepting this law connecting petrol and air in plain tube carburettors, the running of an engine with such carburettors is at once explained. The chain line in Fig. IV. shows the graph of constant fuel mixture of such composition as will approximately give complete combustion of both air and petrol; this may be called the ideal line or line of correct mixture. Any point on a graph lying between the air axis and the ideal line indicates a weak mixture, while a point on the other side of the ideal line indicates a rich mixture It is now clear that this plain tube carburettor must give a weak mixture until the consumption reaches 30 cubic ft. per minute; for higher gas consumption the mixture becomes richer and richer. Hence the graph shows, as is borne out by experience, that with this carburettor the engine would be difficult to start without flooding, would not run or pull at a low speed, and when opened out the mixture would be slightly rich. It is a defect of this type of graph that it does not readily show the percentage variation from constant mixture. Fig. V. shows the percentage composition of the mixture plotted against air. It should be borne in mind that all the graphs shown are corrected for leakage on the induction side of the engine, hence the intercept on the air axis appears less than actually is the case. This correction leaves the type of the graph unchanged; the same is true of the correction which could be made for the volume of vaporised petrol. The volume of petrol vapour is proportional to the quantity of petrol taken, hence the correction would tend to slightly decrease the inclination of a graph. Here, again, the type of graph is unaltered, so that the main conclusions are unaffected. The value of the intercept on the air axis is of importance, for if this were reduced to zero the plain tube carburettor could be made to give quite good results, at least over that range which gives the straight line part of the graph. The leakage past the valve stems, indicated by the dash line in Fig. IV., is seen to be one important factor in the displacement of the carburettor characteristic from the diagonal; but this leakage does not entirely account for the displacement of the graph. The height of petrol in the jet may have a considerable effect on the petrol delivery at low suctions, and readings were taken with a carburettor set to flood over the jet. The first readings brought the graph on the rich side of the ideal line, but as the suctions went up the succeeding readings showed that the flooding was not then affecting the delivery of petrol to an appreciable extent. If the petrol level be so set that the slightest suction causes petrol to flow, the characteristic is found to retain the same value for the intercept on the air axis, but is rounded at the lower end to pass through the origin.

Characteristic Curves.

Effects of engine speed are at once obvious from Fig. IV. The straight line part of the characteristic is extended



Variation of mixture.

Fig. V.

along the line y=ax+b, which is peculiar to that arrangement of the carburettor, while the droop is only slightly modified if at all. The same holds good for higher speeds than indicated here.

The size of the jet modifies the slope of the straight line characteristic as might be expected. Fig. VI. gives the graphs obtained with two jets, the smaller having an aperture one-half the area of the larger. The characteristic is seen to

swing on the point of intersection with the air axis as a hinge, so that by suitably setting the jet, the characteristic of such a carburettor may be set parallel to, or at any desired angle with the ideal line.

Variation of cross section of throat modifies the suction on the jet for equal deliveries of air, and consequently modifies the carburettor characteristic. Bushes of the following diameters were used with the same jet :—

A	$1\frac{1}{4}$ in.
B	1 in.
C	$\frac{3}{4}$ in.
D	$\frac{1}{2}$ in.

These varying velocities past the jet affect both the slope of the characteristic and the intercept on the air axis (see Fig. VII.). The characteristics converge towards a point on the petrol axis behind the air axis.

The droop from the straight line, seen on the characteristics, appears to a greater or less extent on every curve taken. Fig. VIII. gives the characteristics obtained with throats A, B, and C, at 500 r.p.m. It is seen that the maximum quantity of air per minute with A and B is the same, the curve of the droop in the case B being slightly the flatter. In the case of C the droop is still less pronounced. Measured in percentage deviation from the straight line relationship the droops are :—

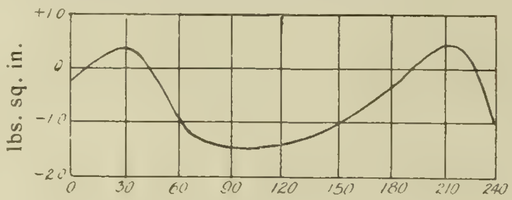
A	23 per cent.
B	17 per cent.
C	5 per cent.

The engine in cases A and B has nearly reached the maximum intake of air with half-open throttle, so that further throttle opening serves to enrich the fuel mixture without greatly increasing the quantity of air taken.

Attention has already been called to Dr. Watson's observation that strongly marked oscillations of pressure occur in the induction pipe of an engine (see Fig. IX.), due to resurgence on opening of inlet valves and to inertia effects generally. These same effects will occur in the partially throttled pipe, but the pressure variations will be damped by the high gas velocities at the partly-closed throttle. On opening the throttle, lower gas velocities are obtained with less and less damping action, so that the surges of gas are more strongly felt at the jet. With open throttle this surging can be so strong as to carry petrol outwards from the induction pipe. The result is that a unit volume of air in surging flow induces a larger quantity of petrol than the same volume in steady flow.

It has been suggested that the high gas velocities in a partly-closed throttle serve to damp this surging. The same would be true of high velocities around the jet, and this conclusion is borne out by the smaller droop observed with throat C. The authors made efforts to wipe out this droop by inserting silencers of various types between the engine and the jet, with quite satisfactory results. Care was taken that a silencer was used which offered no serious resistance to the passage of air. Fig. X is the graph obtained in a silenced carburettor using high velocities round the jet.

Without this droop the characteristic of a plain tube carburettor for all speeds and throttle positions would be a straight line graph. The droop for wide open throttle positions causes the graphs for successive speeds to cover a wedge-shaped area, so that with the same delivery of air various quantities of petrol may be obtained. It should be noticed that this variation is more apparent than real, for in the examples shown, up to 0.8 of the maximum air delivery possible at



Crank angle.
Fig. IX.

any speed the characteristic lies on the straight line

The cause of the droop in the carburettor curve has been stated to be resurgence in the induction pipe, which makes itself seriously felt when the throttle is wide open. It may be urged that the droop may be caused by a change in the type of air flow set up at certain conditions of velocity and throat area. If this were so, it would be expected that the characteristics obtained at different speeds with the same bush would all show the graph falling from the straight line at the same point. Reference to the superimposed curves obtained with the same bush at speeds of 300, 500, and 900 rev. per min. (Fig. VII.) shows that the droop begins to be apparent at points indicating different velocities with different speeds.

Variation in back pressure may be expected to affect the characteristic, for in general the pressure in the cylinder on the opening of the inlet valve is greater than that in the induction pipe, so that a rush out of the cylinder takes place, setting up surging (see Fig. IX.). This effect will be the greater, the greater the back pressure; and as a consequence it may be expected that other things being equal a high back pressure will give a carburettor characteristic with a greater droop than that obtained with a low back pressure. This is brought out in Fig. XI., which shows characteristics obtained with normal back pressure, and with a pressure of 5 lbs. per square inch.

Number of Cylinders.

The adjustable tappets on the valve lifters of the engine were removed first from the two back cylinders leaving the two front ones in action. Afterwards the third cylinder tappets were removed. In this way characteristics were obtained with two cylinders and with one cylinder in action. These are shown in Fig. XII. together with a characteristic taken with four cylinders. The type of characteristic remains unchanged.

Carburettor Characteristics as a Guide to Carburettor Design.

The characteristic of a plain tube carburettor for all speeds has been shown to be a straight line (neglecting droop). The slope of this graph may be varied to any degree by selecting a suitable jet. Let the jet be adjusted so that in a given plain tube carburettor the characteristic lies parallel with the ideal line. The mixture delivered is now weak all along the line, very weak at first, and becoming very nearly correct for large consumptions of air. Another way of stating this is to say that in such a plain tube carburettor there is a constant shortage in the delivery of the liquid.

This being so, the designing of a constant mixture carburettor is within reach of attainment; allow the constant shortage to dribble into the induction pipe, and the problem is solved, except for questions of convenience. Fig. XIV. is the graph of a carburettor set ready for this constant extra supply.

The droop on the characteristic for any given speed does not necessarily lead to waste of petrol, for the graph may be set so that the terminal points of the characteristics be parallel to the ideal line; a suitable constant supply then enables the ideal mixture to be obtained at all open throttle positions. At intermediate throttle positions the mixture would be weak, but if the statement be accepted that such weak mixtures are economical this would be a desirable feature. In any event when maximum torque was required at any speed the required mixture could be obtained by fully opening the throttle; with throttled positions maximum torque is obviously not required, and some weakness of mixture would not be felt.

To sum up, there appears to be no necessity in any carburettor for extra air devices, or the multitudinous contraptions employed, other than a neat form of apparatus for supplying this constant extra supply of petrol.

It may be objected that the foregoing results have been obtained under conditions which do not occur in practice. This

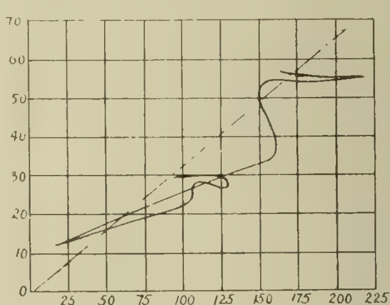


Fig. XXII. Three jets mechanical extra air
Same notation as curves on page 208.

is true, and to estimate the effect of actual running conditions in modifying the characteristic obtained by driving the engine by an electric motor, measurements of air and petrol consumption have been taken with the engine running under its own power. The apparatus employed was the same, except that the electric motor was cut out and replaced by a rope brake. The engine speed was held constant for various throttle openings by varying the load on the brake, and then readings of air and petrol were taken as before. The results obtained bore out completely those obtained by driving the engine by an electric motor. Fig. X is a graph obtained in this way.

Editorial Note.

At the conclusion of Professor Morgan's paper considerable discussion took place, and some doubts were expressed as to the deductions which the author had made from the curves obtained by his experiments. Firstly, Mr. R. W. A.

Brewer expressed a wish that the notation of the curves had been different, so as to give exact quantities instead of merely comparative values. He considered that no surging took place in the inlet pipe at high speeds of engine revolution, and he also said that the velocity of the air passing the jet was of utmost importance, while the temperature of the petrol and air caused large variations, which ought never to be neglected. He referred to the formula, $q = c\ w \sqrt{2\ g\ h}$ (where c = the coefficient of discharge, q = the quantity in c c per sec., w = the area of the orifice, and p = the head of liquid) for giving the discharge from small orifices, and said that he considered the droop shown in each of Professor Morgan's curves was due to the natural droop of the root curve only.

Mr. Remington pointed out that there were advantages in having a carburettor which gave a weaker mixture when throttled than when open.

Mr. F. V. Bickford said that the curve given in Fig. V. was altered in form by setting the air passage horizontally, and so passing the air across the jet instead of in the vertical direction. He also thought that the smallest leakage was of very great importance.

The President, Mr. F. W. Lanchester, in summing up briefly, said that it appeared Mr. Brewer thought the quantity of petrol used varied directly with the vacuum in the intake, while Professor Morgan considered that it varied with the square root of the vacuum, and yet Mr. Brewer had quoted an equation which depended upon a root variation.

From an informal discussion following the proceedings, it seemed that the questions involved were so largely a matter of notation that it would not be possible to base any conclusions upon the discussion as it then stood. As written criticisms upon the paper have not yet been received, we at present refrain from comment, and will content ourselves by expressing the hope that Professor Morgan, Mr. Brewer, and Mr. Bickford may succeed in arriving at some unanimous conclusions which will be of real assistance in carburettor design.

MANUFACTURING METHODS OF THE FUTURE.

A Speculation.

In reading an article in *The Automobile Engineer,* upon the machining of cylinders, the writer was led to consider how far many of the manufacturing methods of the present would extend into the future, and whether some fifteen years hence we should not consider the methods of 1910 somewhat crude.

To explain more definitely the train of thought, let us take the case of cylinder manufacture. After the patterns have been made, and the castings run and cleaned up, the ends are surfaced up, and the barrels bored, angle plates or jigs being used for the purpose. Possibly these two operations have to be done separately, and when they are completed the holes for the holding down bolts must be drilled, the valve chambers machined, the unions for the inlet and outlet of the circulating water provided for, and possibly the sparking plug holes tapped. These processes, as at present practiced, involve the work being set up on different machines a number of times, varying according to the practice of the manufacturers and the design requirements of the cylinder (the two factors being made to harmonise as nearly as possible), and although it is possible that one and the same jig may be made to serve throughout the operations, the cylinders will probably have been worked on not less than three different machines.

Now, it is conceivable, that with carefully-arranged provision for locating the casting correctly, all these processes might be done on a single machine, many of them simultaneously, and that such a method might be profitable, if there was sufficient standardised output to warrant the first cost.

In this matter the question of cost might possibly be kept down by designing the machine to suit the cylinders, rather than *vice versa;* in other words, by building the machine with fixed (as opposed to adjustable) boring and drilling centres. Obviously enough, before having a special machine constructed for the manufacture of a single part, the manufacturer would have to be sure of a large annual output, or, failing that, to be very certain of the permanence of his design. No doubt specialised machine tools on the same principle could be made adjustable within limits, so as to accommodate themselves to varying cylinder designs, and such machines might offer real advantages over the usual present-day methods, but as every movable part introduces a tendency towards additional error in accuracy of work, the simpler tool, built specially to a given cylinder design, would have an advantage, apart from the question of cost.

This latter type would, in effect, be jig and tool combined, and here the writer would like to raise the point, whether the present-day tendency does not err in over-elaboration of the jig, and in not sufficiently modifying machine tools for special requirements, for these are days of large quantities to one pattern.

And as with cylinders, so with other parts the same principles may come to be applied. No doubt some difficulties will make themselves more in evidence in certain cases than in others. For instance, at present, the first operation (say, the drilling of the bolt holes at the base) is often used for locating the work during subsequent processes, while such methods might be inapplicable where all the processes were carried on more or less simultaneously. Under such a system all the operations at various parts of any detail would be located relatively to each other by the machine, and consequently the work would have to be set up very carefully in the first instance. This might lead to demands for greater accuracy being made on forge and foundry, and here we are face to face with the interesting speculation to what extent the casting or forging of the future will require machining. Already stampings have materially decreased the amount of machine work, while the examples turned out by die casting give such promise as to justify us in expecting a great future for this method—a future that would hold very much larger possibilities still if a comparatively cheap method of die making could be discovered.

Again, grinding processes hold vast possibilities of development in the future, and even now are coming into increasing use day by day.

Such developments are, of course, largely dependent on the trend of design and future output requirements. No one regards the mechanism of to-day as the last word in engineering. Improvement, and consequently change, is inevitable until the advent of the car that can be proved by science to be perfect, and as the natural laws of energy and progress alike do not permit of an uninteresting dead level of perfection (for which most thinking men will be grateful), the ideal conditions of manufacturing can never quite be reached. If, however, we elaborate in sufficient numbers a type giving results that warrant some degree of stability and permanence in design, we are surely justified in making details of our manufacturing plant at least as permanent as we expect our designs to be, and surely we have got to such a point? We have the courage of our designers' opinions, otherwise, presumably, we should not manufacture to their designs.

Granted, then, that, whatever radical changes that are introduced in the future cannot take effect without full and sufficient notice to makers of the previous type, it remains to be considered whether the economic conditions of the future give sufficient promise of large enough outputs to justify the methods already outlined, and in view of the big amalgamations which appear likely to become the fashion,—the forces that are at work to crush out smaller firms—and the tendency towards specialisation, the writer thinks there can only be one answer. Already there are numerous firms abroad, and two or three in this country, with outputs sufficient to justify the making of special machines, even for a year's output, and the tendency for each firm to reduce the number of its models makes for the same result. Of course, everyone is perfectly well aware that quite a good few firms do already employ special machines in their work, but it is a practice at present but in its infancy, and the writer will make bold to predict a greatly-increased use of the machine specially built to work on material of a given arrangement and shape—in other words, built to the design of the work to be turned out.

Such speculation would be too nebular, were it not that changes of the further future are the tendencies of the present, which show the lines of development of the near future. "A prophet is not without honour," etc., but the man who does not prophesy, at any rate to himself, about his own business, is likely to be without means as well as honour in his own country.

In such matters prophecy goes by the name of "foresight."

THE LAMPLOUGH TWO-CYCLE ENGINE.

THERE is no doubt that the inefficiency of the ordinary two-cycle engine is due to incomplete scavenging of the cylinder, owing to the natural tendency of the incoming gas to rush across the piston head, direct from the inlet to the exhaust port, and it is also certain that the excellent speed range obtained with the Lucas engine is largely accounted for by the fact that the exhaust port is further from the inlet than any other point in the cylinders, so that unburnt gas can only reach it after all exhaust is expelled. In the Lamplough engine an attempt has been made to still further improve the two-cycle system of operation by the adoption of the Lucas arrangement of cylinders and a special connecting rod setting, which gives either lag or lead to one piston relative to the other, which means that the exhaust port can be opened before the inlet port, and can be shut while the latter is still open. This means that the exhaust port need be only the same depth as the inlet port, and so there is no waste of power, as occurs when the exhaust port opens too early in the stroke.

Opposite a complete sectional view of the engine is given, and the piston arrangement is best followed by reference to the transverse section. It will be seen that there is only one crankshaft, and two pistons are connected to the second and the fourth crank pins, reckoning from left to right, the pistons on the inlet side having ordinary connecting rods, though of a peculiar shape, while the pistons on the exhaust side have rods, which can rock on bearings carried in the upturned ends of the inlet side connecting rods. The crank rotates clockwise, and from this it will be obvious that owing to the greater angularity of the short connecting rod, the exhaust piston will be in advance of the inlet piston when both are nearing the bottom of the stroke, and that the same will be true in the early stages of the up stroke. This gives the early exhaust opening and cut-off that has already been described.

In all, therefore, there are four working pistons, each pair giving an impulse every revolution, and they are charged by a pair of pumps with single pistons contained in the first and third crank pins. In these pumps the same connecting rod arrangement is used, but the second or subsidiary rod is used only to slide one portion of the piston past another portion, so as to open and shut ports through which the mixture is drawn in, as seen in the section. The inlet pipes communicate with about the middle of each of the pump cylinders, rings at the tops and bottoms of the pistons serving to make the necessary gas-tight joints. On the down stroke the angularity of the left-hand connecting rod is greater than that of the right-hand rod, and the ports in the inner and outer parts of the piston register with each other, so allowing mixture to pass into the cylinder. On the up stroke the relative angularity is reversed, the ports are closed, and the gas is compressed. As soon as compression reaches a certain point the diaphragm in the cylinder head lifts, and allows the gas to pass to the two flat reservoirs at the sides of the cylinder castings (seen in section in the transverse view, and being behind the cover plate seen in the elevation). From these containing chambers the working cylinders draw their charges.

As the working pistons are thus their own piston valves it is necessary that they should be gas tight both upwards and downwards from their centres, and this is provided for by the duplex bottom ring (which also serves as a scraper ring), and to get a quick and accurate cut-off a cup-shaped ring is used at the top, it being all cast iron, and not partly brass, like the Gnome piston ring, wherefore better durability might be expected.

It is not easy to criticise or even to comment upon the system of the engine until the results of tests are available, but there is much promise in the idea, and it is obvious that the scavenging and compression should be quite as good as, or even better than, in the case of an ordinary four-cycle motor. On the other hand, for a given total volume of mixture swept out by all the pistons per revolution the four-cycle engine, with an equal frequency of crankshaft impulse, has only four pistons to the Lamplough engine's six, and the wall area of the latter would therefore be the greater. However, the fact that two of the cylinders are pumps only would more than counteract this increase, as far as loss of heat is concerned. Thus it remains to be seen what claims can be made after trials, whether there is any saving in weight, whether there is any increase in efficiency, and whether the balance is as good as or better than the average. Tests are now being made in a very thorough manner, and we hope to be able to publish accounts of them shortly in these columns.

Concerning the mechanical details, the engine is well proportioned and well made, while the lubrication is particularly complete. A section of the pump is inset opposite, and a cross section may be seen at the front end of the crankshaft in the engine elevation. The casing of the pump at its extreme front end is slightly eccentric to the crankshaft, and in this eccentric portion a ring fits, which carries a pin engaging with the plunger which intersects the shaft, as seen in the inset view. The other part of the pump body is concentric with the shaft, so that the plunger reciprocates as the crankshaft revolves. Oil is thus drawn in by the plunger from a passage seen in the lower half of the section, and, as the revolution is continued, the small charge of lubricant is compressed. It then lifts the valve contained in the plunger, and passes to the drilled holes in the crankshaft, which lead right through from end to end. There are leads feeding the secondary big ends of the exhaust side connecting rods, and there are also pipes leading up to the small ends of each of the pistons. The ignition is performed by a magneto driven by a skew gear and a cross shaft.

It is obvious that there is no part of the engine likely to produce noise, and if the intake to the carburettor is quiet and the exhaust silenced effectively, the engine should be practically soundless. Although there are a considerable number of separate moving parts there is some advantage in the absence of the camshaft, and, externally, the engine is both neat in appearance and compact, while the only parts which seem to possess possibilities for trouble are the cup rings on the pistons, which might clog with deposited carbon, and the pump diaphragms, though the latter are so simple that derangement would appear to be very improbable.

THE DEVELOPMENT OF ROAD LOCOMOTION IN RECENT YEARS.

Extracts of a Paper read by Mr. L. A. Legros, M.Inst.Mech.E., before the Institution of Mechanical Engineers.

ON November 18th a paper was read by Mr. L. A. Legros before the Institution of Mechanical Engineers which, although it did not touch the mechanical side of the question to any great extent—but rather dealt with what might be called the political side of the road question—was rich in facts and figures instructive to all those who are connected with mechanical road transport in any form. We therefore make no apology for giving fairly copious extracts of a contribution to automobile literature that at least affords a landmark in the progress of road locomotion.

"*Introductory Note*.—Though attempts were made to adopt mechanical means as aids to locomotion on the common roads as early as 1830, and subsequently about 1860, these efforts were in both cases impeded, and ultimately checked, by the opposition of those interested in horse-drawn vehicles and contingent vested interests. A limited amount of haulage, it is true, was done by means of heavy traction-engines, but this was only effected under conditions so onerous that their field of usefulness was extremely limited.

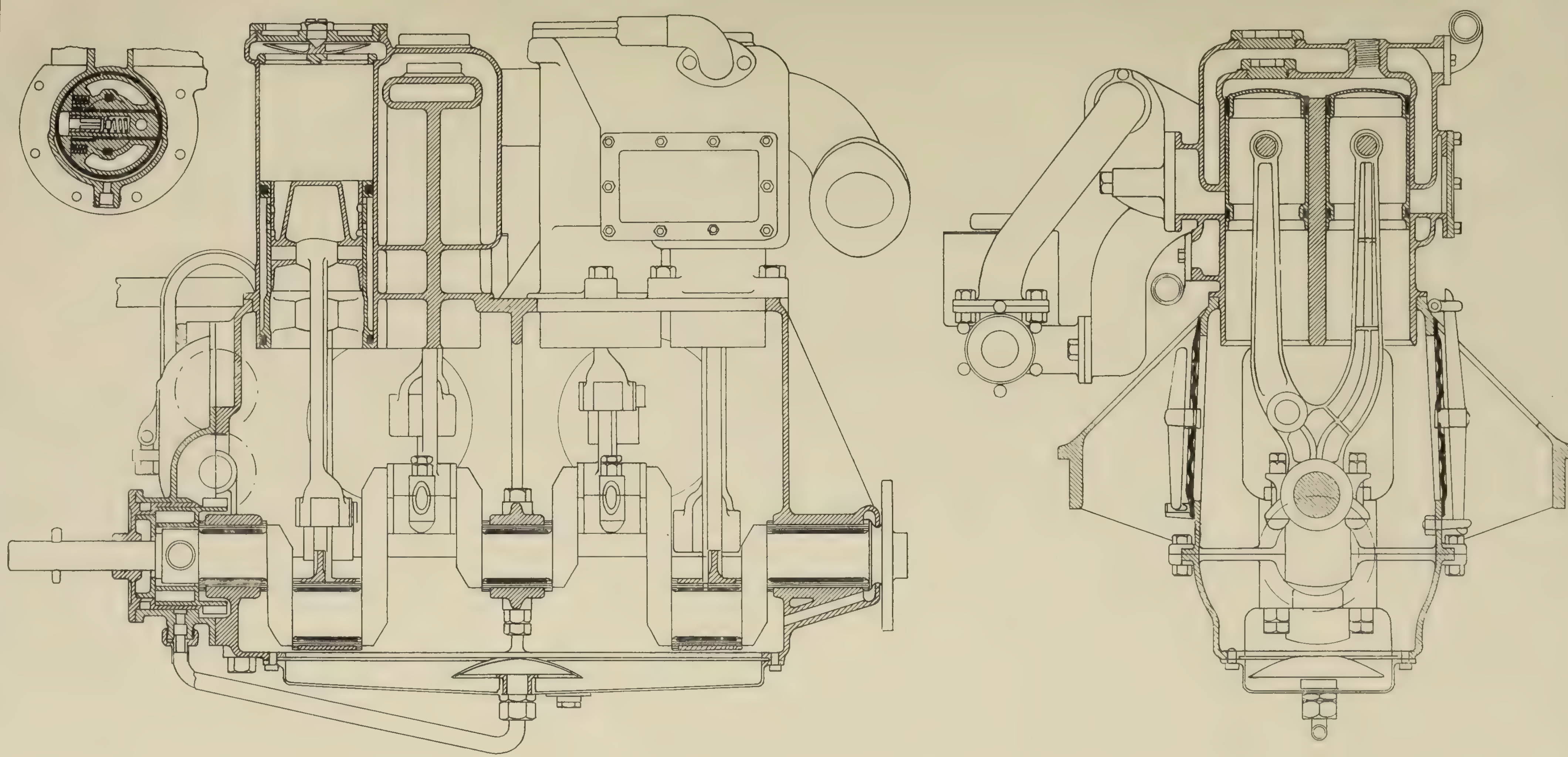
The various forms of road locomotion to be considered may be classed as follows :—

(1) Cycles, including carrier-cycles.

(2) Automobile vehicles used for pleasure purposes.

(3) Light Automobile vehicles employed in public service (motor-cabs).

(4) Heavy motor-vehicles employed in public service (motor-omnibuses).



THE LAMPLOUGH TWO-CYCLE PETROL ENGINE.

The longitudinal section shows one of the two pump cylinders and compound pistons, the transverse section showing one of the two pairs of working pistons. The small section inset is the oil pump at the front end of the crankshaft.

(5) Light motor-vehicles employed for commercial purposes (motor delivery vans).

(6) Heavy motor-vehicles employed for carrying, and in some cases also for hauling (motor-lorries with or without trailers).

(7) Light road-locomotives licensed under the new Act.

(8) Heavy traction-engines of the type previously developed.

(9) Mechanically and electrically propelled tramcars.

It will probably be simplest to deal with the development of each of these classes separately, as most of them have followed widely different lines in their evolution."

From the point of view taken by this journal there is no need to consider certain of the classes of traffic mentioned by the author, and we will, therefore, omit that part of the paper dealing with cycles and proceed to give what Mr. Legros has to say about the second class of traffic that he enumerates.

"Automobile Vehicles Used for Pleasure Purposes.

"A few years prior to 1896 the efforts to produce mechanically-propelled vehicles for running over common roads, had proved fairly successful on the Continent, but the vibration to which the vehicles were subjected gave rise to considerable trouble from parts working loose or breaking. As in the case of the cycle, so also in this case, did the pneumatic tyre prove the solution of the greater portion of these difficulties.

It is not necessary to go into details with regard to the great development which has taken place in these vehicles during the last fifteen years, the history of this being fully given in the technical Press of the industry, but it may be worth noting that a great deal of valuable data have been accumulated by the Scottish Automobile Club, who have each year, from 1902, organized extremely careful trials of automobile vehicles, over peculiarly and increasingly difficult sections of road, with independent observers on all the vehicles. The results of these trials show how

Year.	Miles.	Vehicles entered.	Non-stop runs.	Vehicles finished.	Non-stops.	Finished.
					Entries per cent.	Entries per cent.
1902	468	9	0	3	0·0	33·3
1903	402	25	7	18	28·0	72·0
1904	413	31	7	24	22·6	77·4
1905	595	44	15	36	34·1	81·8
1906	671	84	23	66	27·4	78·3
1907	747	107	19	82	17·8	76·7
1908	772	85	15	53	17·6	62·4
1909	1,007	68	17	58	25·0	85·3

Table I.

high a percentage of the cars subjected to the tests have gone through the trials without stoppage on the road, and this under conditions which were frequently extremely adverse." (See Table I.)

"Light Automobile Vehicles Employed in Public Service (Motor-Cabs).

"Following the application of the internal-combustion engine to pleasure vehicles comes its application to public vehicles plying for hire, such as cabs. Several attempts to introduce this form of locomotion in London in the earlier days of the automobile industry proved abortive, owing to the heavy first cost of the vehicles at that date, and to the excessive repairs they required, and largely to the want of experience on the part of both the designer and the user of the vehicle. Progress in the adoption of the motor-cab in London can be estimated very closely from the returns of licences issued."

An idea of the growth in this branch of automobilism up to the end of 1908 can be obtained from the diagram showing the total numbers of vehicles of various classes licensed in London up to that date. These figures were obtained from the Report for 1908 of the Commissioner of Police of the Metropolis.

"In 1904, 585 new horse-drawn cabs were licensed, and only one new motor-cab was licensed—in 1908 only twenty-one new horse-drawn hansom cabs were licensed (or with four-wheelers fifty-nine in all), whereas 1,715 new motor-cabs were licensed. The total number of motor-cabs licensed in 1909 was 3,956. The motor-cab has proved so much more expeditious than the horse-drawn cab, that the period at which these latter vehicles will become practically obsolete in large towns is obviously close at hand.

"Heavy Motor-Vehicles Employed in Public Service (Motor-Omnibuses).

"The motor-omnibus has not followed along the same lines of development as those taken by the motor-cab. The chassis of the motor-omnibus is of necessity of much heavier construction than that of the motor-cab, on account of the greater weight it has to carry. The total weight is, in fact, so great as to prohibit the use of the pneumatic tyre, and to necessitate the use of solid rubber, together with such constructional modifications in the framing and mechanism as have been found necessary to meet the conditions arising from less efficient cushioning of the unsprung weight. The motor-omnibus has, in fact, more closely followed the lorry than the pleasure vehicle. . . . . .

Whereas seventy-eight new horse-omnibuses were licensed in 1905, only twenty new motor-omnibuses were licensed; the number of new horse-omnibuses licensed fell to two in 1907 and none in 1908; the total number of motor-omnibuses licensed in London in 1909 was 1,180.

Since the amalgamation of the three companies (London General Omnibus Company, the Vanguard and the Road-Car Companies) the number of motor-omnibuses in service, with their mileage, is as follows :—

Year.		Omnibuses.	Year.		Omnibuses.	Total mileage.
1908	Summer	603	1908-9	Winter	536	17,072,120
1909	,,	698	1909-10	,,	664	27,500,000

These figures give an average annual mileage of about 30,000 per motor-omnibus in 1908, and about 40,000 in 1909. There can be no doubt that the motor-omnibus services at the commencement suffered very badly from inexperienced management. The fact that as many as 96 per cent. of the vehicles were kept in service on the road was advertised and encouraged, with disastrous results in frequency of breakdown and in the ultimate cost of repairs. The figures above quoted show the great improvement in mileage, as well as in reliability, that can be effected by proper maintenance. Both steam and petrol engines are employed on the motor-omnibuses in London; the number of steam-omnibuses is, however, relatively small. One of the petrol-buses belonging to a London company was stated to have travelled 148,100 miles up to the 4th of June, 1910.

The use of motor-omnibuses in London does not, however, give a fair indication of the widespread development of this mode of locomotion. Several of the railway companies have adopted the motor-omnibus for service in places where their stations lie at some distance from the towns they serve, and in many cases they are employed where small outlying towns, otherwise devoid of regular communication, could not be economically connected to the railway service by other means. Foremost among these companies is the Great Western Railway; the development of motor traction by this railway having been rapid, which gives the total number of motor-omnibuses in use by it for each of the past six years. Motor-omnibuses are also used by the London and North Western, the Great North of Scotland, the Great Eastern, the North Eastern, the London and South Western, and the Cambrian Railways. Particulars of the services and mileage run by these omnibuses on the regular services are given in Table II."

These figures, the author tells us, were calculated from the time tables of the respective railway companies.

"The author is of opinion that the average cost of working these omnibuses, including wages of driver and conductor, fuel and oil, cleaning and waste, maintenance and renewals—in fact, all running expenses, but not depreciation—will be found to be between 9.5 and 10.5 pence per mile.

The life of the tyres on motor-omnibuses is generally guaranteed at 10,000 miles. The life of the omnibuses cannot at present be ascertained, but it appears that with proper maintenance and supervision, they remain in good working order and have a considerable life after six or seven years' service.

Light Motor-Vehicles for Commercial Purposes.

The development of these vehicles has taken place rather more slowly than that of pleasure vehicles and motor-omnibuses. The necessity for keeping down the weight of the vehicle has required the adoption of light construction resembling that of the chassis of the pleasure car, while, at the same time, the addition of the load has raised the total weight above that at which the pneumatic tyre can be economically employed. Hence a large amount of special experience was required to be ob-

tained before the reliability of this type of vehicle could be assured.

One of the most important individual applications of this type of vehicle is

TABLE II.

Railway.	Total number of omnibuses.	Minimum number of omnibuses for service.	Number of routes worked.	Single trips made per annum.	Total miles run per annum.	Total miles per omnibus on minimum service per annum.	Longest trip, miles.	Average length of single trip, miles.	Mean commercial speed in miles per hour.
G.W.R. ...	96	50	34	114,842	825,458	16,510	20·6	7·19	9·24
L. & N.W.R. ...	13	11	10	85,488	252,861	22,990	11·4	2·96	9·19
G.N.S.R. ...	—	7	5	8,320	130,780	18,683	19·3	15·09	10·04
N.E.R.		6	4	16,900	56,607	9,435	22·6	3·35	9·76
G.E.R.		6	5	14,144	86,736	14,456	10·9	6·13	9·72
L. & S.W.R. ...		4	2	3,172	45,687	11,422	21·4	14·40	8·87
Cambrian ...	—	2	1	2,080	16,224	8,112	7·8	7·80	5·57

that of the General Post Office, who have working under contract at the present time 120 vehicles, in addition to six of their own; the number also of motor-vans so employed is growing weekly. One large firm of general providers in London have eighty vans on the road, each averaging sixty miles per day, while another London firm, engaged in a similar business, have fifty-five vans running from seventy to one hundred miles per day. . . . .

HEAVY MOTOR-VEHICLES EMPLOYED FOR CARRYING (AND IN SOME CASES ALSO FOR HAULING).

Referring to the vehicles of the heavier class, Mr. Legros says:

"Of the figure of 6,500 light locomotives just quoted probably 5,000 are motor-lorries. Many of these are driven by petrol engines, but unlike the motor-omnibuses the majority are fitted with steam engines. The slower speed of these vehicles, and the greater permissible weight render the use of a fire-tube boiler possible. Of forty-two of these vehicles entered for a recent driving competition, two were stated to have run 85,000 miles and upwards, and seventeen to have run 50,000 miles and upwards. With the lorries and heavier vehicles about 10,000 miles per annum appears to be the usual performance, while 25,000 miles per annum appears to be the maximum for the lighter commercial vehicles.

The author said very little about steam tractors (not, in fact, in our opinion, as much as they deserve), and after dealing with the heavy traction engine and tramcar, made a brief reference to railless electric traction, and its various possibilities.

THE ROAD QUESTION.

This brings us to the second part of the paper dealing with the road question, and, in controverting the charges of damaging the roads, so frequently brought against the self-propelled vehicle, the author brings forward the following causes of defective roads:—

"(*a*) The roads were constructed on an insufficient foundation.

(*b*) The roads were made of unsuitable material.

(*c*) The road metal was not broken to proper size.

(*d*) In the rolling operations, road-scrapings, together with other dust and mud-producing matter, were mixed with the road-metal proper, or spread on and watered in with the road-metal, to act as 'binding.'

(*e*) The roads were constructed with an excessive amount of camber."

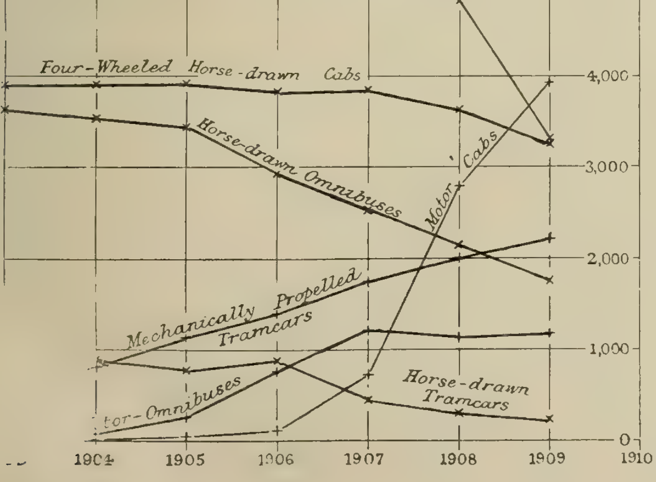


Fig. I.

On the question of road maintenance costs the writer of the paper says:

"It has been publicly stated by Colonel Crompton that in London the cost of rubber tyres worn out on public service and trade vehicles is equal to the amount expended on road maintenance and scavenging in the same period of time. In the opinion of the author this statement is inside the mark. The average cost of road-maintenance and repair, with cleansing, scavenging and water in London was £767 per mile per annum for the two years 1904-5, and for the two years 1907-8 it fell to £680 per mile."*

THE EXTINCTION OF THE HORSE.

Towards the end of his paper Mr. Legros arrives at some interesting conclusions as to future road traffic development (see Fig. I.).

"The total number of public service vehicles of each class licensed in London at the end of each of the past six years is shown in a diagram. This number is not that of the vehicles actually in service at those dates, because, owing to various causes, some vehicles are licensed more than once in the same year. These causes are, however, fairly constant in character and the form of the curves, and hence the rate of increase or decrease in the total number of vehicles is but little affected. Various public service horsed vehicles in London will become extinct as follows; but in the question involved there are so many and varied conflicting factors that it would not be prudent to speak dogmatically:—The horse-tramcar at the end of 1912; the horse-omnibus at the end of 1913; the hansom cab at the end of 1913; the four-wheel horse cab before the end of 1921."

A Point in the Discussion.

An interesting discussion followed, but, as it dealt in the main with the road aspect, we do not propose to give it here. An interesting point, however, from the mechanical point of view, was raised by one speaker, who referred to the inability of an automobile to progress in a straight line. This item was dealt with by Mr. Legros in his reply, and the better to explain his argument he drew a diagram

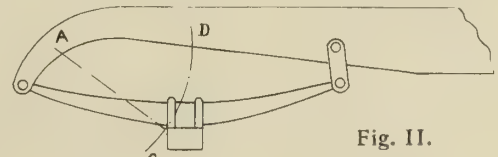


Fig. II.

on the blackboard, which is given in Fig. II. This diagram enabled Mr. Legros to point out that with the arrangement of spring suspension usually employed on the front axle of a car, the movement of the axle relatively to the frame follows the arc C D of a circle having a radius emanating from a point A approximately in the position shown in the sketch. Thus the front axle tended also to move somewhat backwards, when it moved upwards relatively to the frame, and consequently the steering was momentarily thrown out by the spring action—which occurrence repeatedly happening, owing to inequalities of road surface, gave the effect to which attention had been drawn.

*Report of the London Traffic Branch of the Board of Trade, 1909.

THE IDEAL MILITARY CAR.

Being a summary of opinions formed from several years' experience of manœuvres in England.
By One of the Army Motor Reserve.

FOR several years past I have attended with my car the Army Manœuvres, and as an officer in both the (late) Motor Volunteer Corps and the present Army Motor Reserve I have had ample opportunity to note the advantages and disadvantages of various types of machines. At present the Army itself owns no special type of car, and I have not noticed any two alike in any particular other than the noise they produce. At the same time the vitality of some of them is highly creditable to their drivers, for undoubtedly thousands of different soldiers must have learnt to drive on them, for weeks together they have, perforce, to have remained uncleaned, the roads they are required to travel have been fearful, and their overloading simply shocking. That these early ones survive is more than wonderful, that they still do good service is marvellous.

I had the opportunity some time ago to visit Metz, the most military of all German frontier towns, and I could not help being struck with the homogeneity of the military cars I saw there. Being there as a mere tourist, of course, I could not inspect them, otherwise than cursorily, but they appeared to me to be all of one family, and, whether the body was a *limousine* or an open one, I have no doubt that the parts were interchangeable, and the same box of tools, spares, and tyres would fit the lot. Several I saw conveying at one time cadets to the battlefield, and at another provisions, ammunition, or details. These particular cars were not elegant; their sides were as flat as railway-truck walls; their seats were planks placed across, and their entrances were at the back, and opened like our horse-boxes do. But their bonnets looked as if their engines were powerful, and out in the country they appeared to average, loaded, their twenty miles an hour. Possibly their sides were bullet-proof. I see no reason why they could not be made so in emergency, and set at such an angle that, at any rate, some bullets might be deflected. The large closed motors in which high officers moved seemed all of a kind, grey in colour, paint without varnish, and fitted inside for comfort without flashiness. No doubt in Germany it is easier to have all cars of a kind than it would be in France or England, because the number of separate manufacturers and types is—or was—more limited, but it occurred to me during the late manœuvres in Wiltshire that our diversity of type, size, and make cannot but act to our detriment when the unhappy occasion for their real use shall arise. I daresay I came across fifty cars in use by the various staffs, and I do not remember ever seeing two of one kind together. Certainly the Army Motor Reserve cars all differed, and it was common remark that each required different handling and separate spares. Of course, since the cars of the officers of the A.M.R. are all private property, this is not to be wondered at, nor, indeed, is anything else to be expected, but when the War Office itself takes over its own motor work, it is to be hoped that it will see every single car is of an absolutely interchangeable type with every other. Delay and loss might easily be consequent on the driver's inability to borrow a spare detachable rim or wheel, a duplicate magneto, or some other equally important part. On a campaign there would undoubtedly be a repair car, or travelling store, in attendance. If all the cars it served were alike, its contents would be much more condensed, ready, and more likely to fit. As at present constituted—but I can leave this to my readers' imaginations, and we will discuss the type most suitable for the work such a car would be called upon to do.

Concerning size. The work a car would have to do in England and on the Continent would be of quite separate natures, and there would be no need at home to have anything approaching the same power that would be necessary abroad. Even in Germany and France a modern four-inch car would be quite fast and powerful enough for any work it might be called on to do other than the transport of troops in bulk. A forty or sixty horse-power machine would require more petrol, more tyres, and more room, giving in return very questionable extra advantage. A car employed behind the line would for ever be on roads blocked with guns, waggons, troops, and the thousand and one *impedimenta* that are inseparable from service conditions, so that pace would seldom be possible even on the widest of *routes nationales*. On the Continent I believe troops, as there are no hedges, use the fields more than those who only know England can imagine. This, of course, leaves the roads more clear, but, now that mechanical transport is coming in, it is doubtful whether much will be gained, and certainly it is better for a car to run into horse-flesh than traction engines. But here—at any rate, in all parts of England except selected fashionable manœuvring areas—troops will have to keep to the roads, and consequently a car that might suit foreign lands would be much too long and unwieldy to be anything but a snare and a delusion. The car that will be wanted for active service in England will be a short, stout one with a lock that will enable it to be turned in almost its own length. More than forty miles an hour it will seldom be needed to go, but as our poor roads will soon collapse under extraordinary traffic, and as it will oftentimes have to go into fields and muddy lanes, it will require a low speed that will climb the roof of a house, a cooling system that no amount of five-mile-an-hour pace will upset, and a clutch that no amount of slipping will ever put out of action. But ease of turning will be the first and chief consideration, and that this is so can be testified by any motorist who has ever dared to do that pernicious thing, the following of hounds *en automobile*. In that he may get his deserts, and remain in the ditch.

Next we come to the question of tyres, and I always regret that I have never had the experience of having been a combatant in manœuvres in order to judge of the pace possible when confined behind the firing line, and never being able to run about between and on the flanks of the two armies. It would be very interesting to know what pace could be safely averaged by a car carrying, say, a chief of the staff, among an army of a hundred thousand men with a front of forty miles of ordinary English country. Not fifteen miles an hour, I warrant, and it is because of this that I venture to express my opinion that on active service conditions, solids or semi-solids would be preferable to the more comfortable but more treacherous and extravagant pneumatics. This opinion does not apply to manœuvres, for in them there is necessarily much make-believe and a state of politeness that would soon be lost in the excitement and conditions of the Real Thing. An army on the march is very puncturesome. I have seen a tyre with a cartridge case sticking into it, and out of my own Dunlops I have picked something that looked like part of a tin-opener. To say nothing of beef tins themselves.

Therefore if I were asked to suggest a type of tyre for light military work, my opinion would be that it should be fitted with detachable rims or wheels, the spare ones being solid or semi-solid, and the original ones pneumatic and interchangeable. But every wheel should be exactly the same size—the Army should only know of one kind, say, 880 by 120—so that by no possible chance could a wrong-fitting tyre or tube ever be dealt out. For emergencies a tyre that would dig itself out of grass and sand should be carried, and the War Office might institute some splendid trials. I believe the real reason many cars exist with front tyres of a smaller width than their back ones is owing to an exploded idea that it was impossible to steer so correctly with front tyres of a large size. This notion, however, spare rims and wheels have put out of court.

So much for my opinions on ordinary motors for military purposes, and I will leave the question of motors as gun-carriages to those with wider experience. At the same time, however, it would seem to me that a great deal of nonsense has been written and illustrated on this subject, because the recoil of a very small-sized gun fired from a motor might have the most unexpected results. If the brakes were not on and the engine out of gear the car would certainly find the ditch; if the engine was in gear it would do it no good; and if the brakes were on I should expect to find a scar on the tyre very much like one gets when locked wheels slide down a hill. Possibly a sprag could be devised similar to the old patterns that every car was provided with, but even then I imagine something might go, and it is not good for a *chassis* to bear continual stresses it was never intended to have to meet.

O. J.

CORRESPONDENCE.

No responsibility is accepted by the Editor for the opinions of correspondents, and the right is reserved to omit any portion of a letter if it is considered desirable to do so. Letters to be inserted with a nom de plume must be accompanied by the name and address of the writer. If publication in the next issue is desired letters should reach the Editor in no case later than the 20th of the preceding month.

ESTIMATING HORSE POWER.

Sir,—Your leading article on the above subject in November's issue must be very disappointing to anyone who has followed the whole discussion of the subject, by the S.M.M.T. and in other technical journals. (*Engineering*, December, 1908, and January, 1909.) The R.A.C. formula was proposed by a committee of scientific men, who know their business. They assumed a value of yp of 67 lbs. per square inch (y—efficiency, p—mean pressure in working stroke), and a piston speed of 1,000 feet per minute, and although some engines may give higher values of yp, no great objection can be raised against the figures. The resulting formula $d^2/2.5$ gives a very close approximation to the horse-power of all petrol engines *at 1,000 feet per minute piston speed*. The formula has been wretchedly misused by many writers, who have compared the figures it gives with brake-horse-powers obtained at very different piston speeds. The definition of the formula ought to have had this piston speed assumption tacked on to it. At any other piston speed the horse-power is simply proportional to the speed.

The differences between the R.A.C., Dendy Marshall, and Lanchester formulas are purely those of definition. The Dendy Marshall gives the horse-power at a fixed revolution frequency instead of at a fixed piston speed. If yp is the same in both formulæ the Dendy Marshall gives the horse-power at 1,250 revolutions per minute, but whether the horse-power is measured at a fixed piston speed or a fixed revolution frequency is merely a matter of choice. If the value of yp be increased by 20 per cent. to allow for the higher compression ratios used in modern engines the horse-power becomes $\frac{1}{2}d^2$ at 1,000 feet per minute piston speed, or $\frac{1}{12}d^2S$ at 1,000 revolutions per minute. That these two formulæ are identical is made evident by considering the case of an engine having 6in. stroke, in which 1,000 revolutions per minute implies a piston speed of 1,000 feet per minute.

The rating for purposes of taxation is more equitable if carried out at a fixed piston speed than at a fixed revolution frequency, since similar engines are alike capable of running at the same piston speed, and are then equally severely stressed. The R.A.C. formula gives, therefore, an equitable basis of rating for purposes of taxation, if the taxation is fixed at so much per horse-power. Some old engines were designed to run at piston speeds below 1,000 feet per minute, and cannot develop the power given by the R.A.C. formula, but the number of these engines is very small, and is being reduced every day; and if the tax were fixed per horse-power the injustice to these old engines would not be great. On the other hand, most modern engines may safely be run at piston speeds of 1,500, or in some cases 2,000 feet per minute, which enables them to develop proportionately greater power, and by special design and selection of materials piston speeds up to 3,000 feet per minute have been employed in racing. So long as the 20 miles an hour speed limit remains in force, it is practically impossible to utilise *legally* any greater piston speed than 1,000 feet per minute except in climbing hills on low gears. The rating at 1,000 feet per minute is therefore a very satisfactory basis for purposes of taxation.

If the rating is to be employed for handicapping in races, it is essential to vary the piston speed in such a manner that the reciprocating parts will be equally severely stressed when developing the formula horse-power. Lanchester's formula does this, the piston speed being 1,000 $r \times 1/3$ feet per minute, r being the ratio of stroke to bore. It is also desirable in handicapping to allow for the proportionally greater heat loss in small cylinders than in larger cylinders, since this loss affects yp. Professor Callendar's addition to the formula does this by making $yp = k\left(1 + \frac{1}{d}\right)$, and the formula recommended by the S.M.M.T. is a simplified approximation involving Lanchester's piston speed and Callendar's heat loss. It is B.H.P. = $0.2d\,(d-1)\,(r+2)\,N$ at the piston speed $1{,}000\,r^{\frac{1}{2}}$ feet per minute.

J. B. HENDERSON.

Sir,—With reference to the leading article on "Estimating Horse-Power," in your November issue, the enclosed table, giving the h.p. of twelve well-known motor cars, calculated according to six different formulæ, may be of some interest.

It will be observed that of the six formulæ there is only one (Lanchester's) which gives neither a maximum nor a minimum value, thus approaching most nearly to the golden mean.

The third column gives the figures corresponding to a tentative formula, which is the result of some attempts by Mr. F. J. Dykes, of the Cambridge University Engineering Laboratory, and myself, to evolve a formula suitable for handicapping purposes in competitions. It is based on the volume swept out by the pistons per unit distance run by the car, and is given in the subjoined table under the heading "Dykes."

Having obtained a formula, we endeavoured to test its accuracy by applying it to the results of an impromptu hill-climb, the cars used being a 28 h.p. six-cylinder Lanchester, an 18.22 four-cylinder Enfield, and a home-made four-cylinder car, which is familiarly known as the "Dykesmobile." However, the number of cars employed was too small, and the observations were not sufficiently accurate to give any reliable data.

In spite of Mr. F. W. Lanchester's warning that "such tests are entirely delusive" (see "The Horse-power of the Petrol Motor," in the Proceedings of the Incorporated Institute of Automobile Engineers, 1906-07, page 189), I venture to think that a series of trials such as suggested by you, would furnish us with a great deal of interesting information, and with an excellent test-ground like Brooklands available, it should not prove to be a difficult matter to organise.

C. A. BRANTSEN.

Make of Car.	No. of Cyls.	Dimensions of Cyls. m/m.	Diam. wheel- m/m.	Top gear ratio.	Horse-power by formula: Lanchester.	R.A.C.	Dykes	S.M.M.T.	Dendy Marshall	O'Gorman
28 Lanchester	6	102 × 76	880	4·4	33·3	38·7	35·6	41·5	23·7	39·3
12 Sizaire-Naudin	1	120 × 140	750	3·5	9·7	8·9	14·1	11·2	10·1	11·0
40-50 Rolls-Royce	6	114 × 121	895	2·9	50·0	48·4	45·8	57·8	47·2	57·0
15 Straker-Squire	4	87 × 100	810	3·3	20·2	18·8	18·3	21·0	15·1	23·0
12 Talbot	4	80 × 120	800	4·0	19·5	15·9	23·0	19·2	15·4	21·5
15 Daimler	4	80 × 130	870	4·4	20·2	15·9	25·3	19·8	16·6	22·2
15 Napier	4	83 × 127	815	4·0	21·0	17·1	25·5	21·2	17·5	22·8
12-16 Siddeley	4	79 × 115	810	4·4	18·7	15·5	23·3	18·3	14·4	21·0
20 Standard	6	89 × 108	880	3·5	32·5	29·5	30·8	34·2	25·6	36·6
18-24 Austin	4	105 × 127	880	2·6	30·0	27·3	24·8	33·5	27·7	34·4
20 Crossley	4	102 × 140	880	3·0	30·0	25·1	29·8	32·8	29·1	33·5
20 Vauxhall	4	90 × 120	875	3·9	23·2	20·1	26·0	24·2	19·4	26·0

FORMULÆ

Lanchester: $\cdot 4 . D^{1\cdot 5} S^{\cdot 5} N$.

R.A.C.: $\cdot 4 . D^2 . N$.

S.M.M.T.: $\cdot 2D(D-1)(R+2)N$

O'Gorman: $\cdot 4 . D^{1\cdot 6} S^{\cdot 4} N$.

Dendy Marshall $\frac{D^2 . S . N}{12}$

Dykes: $\cdot 965 \frac{D^2 . S . N . Tg.}{Wl.}$

where

D = diameter of cyl. in inches.

S = stroke ,, ,,

Wl = diameter of road wheel in inches.

N = number of cyls.

Tg = top gear ratio $\frac{\text{engine}}{\text{road wheel}}$

R = ratio $\frac{\text{stroke}}{\text{diameter}}$ of cyl.

Table referred to in letter from C. A. Brantsen.

BALL v. ROLLER BEARINGS.

Sir,—The letter of your correspondent, Mr. R. F. Hall, would have been more convincing if it had been less extreme. We hardly expected anyone, however, to take Mr. J. V. Pugh seriously in his advocacy of the cup and cone type of bearing, although his analysis of the forces acting upon the bearings in the hubs of motor vehicles is otherwise most instructive.

That the old cycle form of bearing is dead as far as the modern motor carriage is concerned scarcely admits of discussion, for it has been conclusively proved that this type of bearing is greatly inferior to the more recent non-adjustable annular type from the point of view of load-carrying capacity, and altogether apart from any question of adjustability or ingress of water.

It is impossible to disagree with many of the general observations that Mr. Hall makes, but we, as manufacturers of ball bearings of over twenty-five years' standing, cannot go so far in regard to the arguments or the figure he advances, since these are evidently based upon a very general misapprehension and lack of knowledge as to the actual construction of the latter type of bearing.

To mention quite a small point first, yet one of some importance, Mr. Hall says that "the radius of the bearing surface is struck from a centre taken at 7/10 of the ball diameter, this being common practice." So far is this from being the fact that we do not know one single firm of the many now making annular ball journal bearings who use this proportion. For thrust bearings intended to carry axial loads at high speeds such a proportion would be correct, but even in this type of bearing, when designed for heavy loads, the radius of the bearing path is made much less than the figures given.

The importance of this will become more manifest when it is remembered that the areas given in table 1 will be materially affected by this alteration. Even as they are presented by Mr. Hall there is an evident error between b and c, assuming that B and C in Figure II. are correct representations of the curves used for the purposes of calculation, because the area of contact depend directly upon the radius of the curves, but the error becomes very much greater when the necessary correction as pointed out above has been made.

This error is, of course, repeated in each of the tables that follow, and consequently renders them very misleading and quite useless for the purpose for which they were prepared. As they stand, in every case, apparently, whether radial or axial load is under consideration, the annular ball bearing is greatly inferior to the taper roller bearing in load-carrying capacity, but it is somewhat amusing to find on comparison of the lists issued by the respective makers that the safe working load given for the roller bearing is in some cases even less than that given for the ball bearing, and in no case does it exceed that amount, that is to say, on the authority of the manufacturers themselves, that the safe working load of the taper roller bearing is usually not so great as that of the safe working load of a ball journal bearing of similar size.

It would be interesting to know upon what authority Mr. Hall says that "whether it is a

ball that is carrying the load or whether it is a roller only, three of them can be regarded as load-carrying factors," and "it is well known that only one-fifth of these (balls or rollers) are effective against radial load." Professor Stribeck, indeed, shows that the greatest load on one ball is equal to five times the total load on the bearing divided by the number of balls in the bearing,

$$\text{or } p = \frac{5P}{n} \text{ (Stribeck's Equation II.)},$$

but this is entirely different to the statement quoted above.

With regard to thrust or axial load, Professor Stribeck further says: "It has been experimentally determined that the thrust-carrying capacity of the annular type of bearing is to the radial capacity as 1/10—1/4 to 1, depending upon the relation of the ball diameter, race curvature, and number of balls.

For speeds above 1,500 r.p.m. these radial bearings are *more* efficient thrust carriers than the collar type, and exhaustive experiments carried out by this company in the latter part of 1908 have conclusively shown that the results obtained by Professor Stribeck may be accepted without reserve.

It would appear that the whole indictment centres round the non-adjustability of the annular type of bearing, and Mr. Hall makes a great point of the supposed adjustability for wear of the well-known forms of roller bearings employing taper rollers. A little reflection, however, will show that this feature is not possessed to any extent by any form of bearing available.

In a ball or roller bearing for radial loads any wear that takes place is divided between the outer ring, the balls or rollers, and the inner ring, and it may be further said that this wear is evenly distributed over all the balls or rollers and all round the rotating member, whether the inner or the outer ring, but is confined to one comparatively small part of the stationary member. Mr. Hall again says: "This wear commences and leaves off within 25 per cent. of the total circumference, three-fourths of the circumference showing no signs of wear or reduction of diameter." It is obvious that the wear which is distributed over all the rolling members and the rotating member will be very much less than that of the stationary member—in fact, it may be so small as to be quite imperceptible, but it is equally obvious that it is only this distributed wear that can be compensated for by "adjusting" the bearing. Therefore, the much greater localised wear still remains uncompensated for, because it is quite impossible to design any arrangement, either of roller or ball bearing, that will compensate for it.

It must also be remembered that when any bearing designed as an antifriction device has become worn the surface upon which it depends for its efficiency has been destroyed, and to adjust such a bearing, even if it were possible, would be but to prolong and accentuate the unsatisfactory conditions under which it is working.

The whole secret of successful application of ball bearings consists in the choice of a bearing of suitable size for the load to be carried and the proper protection of the bearing when mounted. The conditions of mounting are now so well known as to need no repetition beyond the very general remark that care must be taken to ensure that no load of any kind is put on the bearing by the method of mounting adopted. This latter point is too often neglected, but given reasonable attention to these conditions the non-adjustable annular type of ball journal bearing is an ideal bearing for every position in the car, with the possible exception of the front hubs, where the loads developed in turning corners at high speed, and what Mr. Hall so aptly calls the "accidental stresses," are undoubtedly great.

Even under the arduous conditions of this position, however, we could cite many cases from our own experience where, when proper attention has been given to the points enumerated above, results have followed which amply confirm the statements made by Mr. Pugh in the closing paragraph of his article, and there appears no reason why similar results should not always be obtained.

One instance only will suffice. We have in our possession a set of ball bearings that have been taken out of the front hub of a Humber car after covering certainly not less than 45,000 miles. In these there is no perceptible looseness or sign of wear. It may not be out of place to remark here that one of the most important points of ball bearing manufacture is the final examination of the assembled bearing so that there shall be absolutely no looseness or side shake, anything like a loose or "sloppy" bearing being discarded, and when this is attended to properly we are able confidently to say that ball bearings if properly protected are perfectly suitable for use in the front hubs of motor vehicles.

THE AUTO MACHINERY CO., LTD.

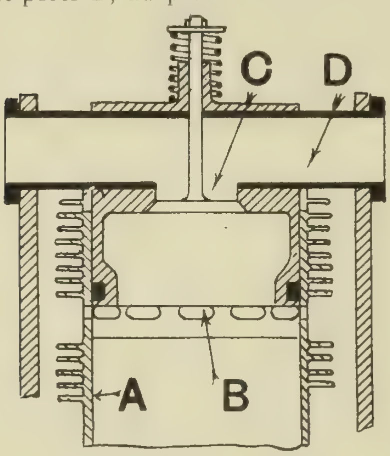
[Concerning this letter and Mr. Hall's letter published in the last issue, the Editor would be glad to hear from any reader who has made comparative road tests of the conical type of roller bearing and ball bearings working under *exactly* the same conditions as regards protection from water.]

RECENT AUTOMOBILE PATENTS.

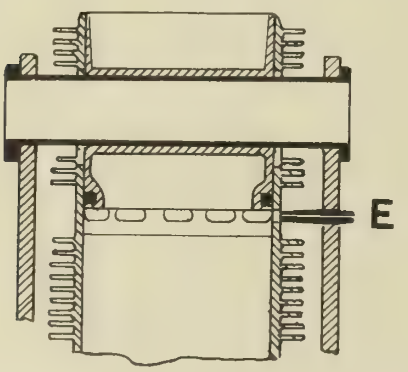
By Eric W. Walford, F.C.I.P.A.

A Valve System.

The piston works inside a sliding cylinder member A, having formed in it exhaust ports B, adapted to be cut off when



moved up beyond the stationary cylinder head. Exhaust takes place directly into the atmosphere, and inlet is effected

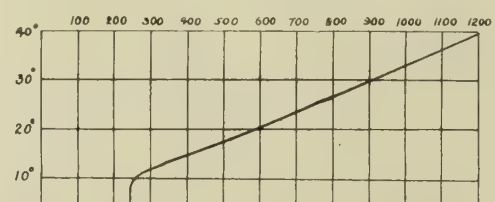


through an automatic valve C, communicating with a hollow pipe or trunnion D, supported directly from the crank chamber. The cylinder head is free to swivel slightly on the trunnion, so that it is self-aligning in regard to the movable cylinder. In one construction there are no separate inlet valves, air being drawn into the cylinder through the exhaust orifices, which open directly into the air, and fuel under pressure is injected through one of these orifices from the nozzle E. It is difficult to see how the valve system and engine construction in general of a four-cycle engine could be made much simpler than in this last case.

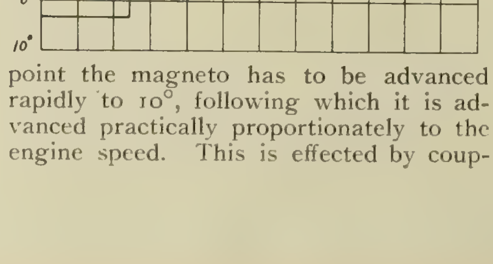
F. W. Lanchester. No. 23,106/09.

Automatic Magneto Advance.

The curve shown demonstrates the requirements in regard to magneto advance. The abscissae represent revolutions per



minute of the engine, and the ordinates angles of retardation and advance. It will be seen that up to 250 r.p.m. a retard of about 5° is desirable, at which

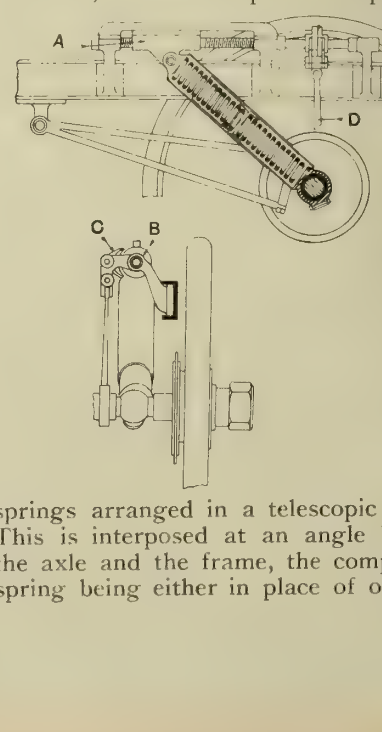


point the magneto has to be advanced rapidly to 10°, following which it is advanced practically proportionately to the engine speed. This is effected by coupling the magneto driving shaft A to the amature spindle through a nut and quick thread device, the nut being adjustable axially by governor mechanism. The quick thread has a very rapid pitch at its commencement, the pitch gradually falling, so with the first increment of speed the magneto is advanced but little relatively to its driving shaft, after which the adjustment becomes rapid until the nut comes up against a spring B, which creates sufficient resistance to give the desired effect. The governor balls are connected together by transverse springs not illustrated.

Ernest Eisemann and Co. No. 9,488/10.

Adjustable Spring Suspension.

The back axle is connected to the frame by triangulated radius rods, as illustrated, and a compression spring or



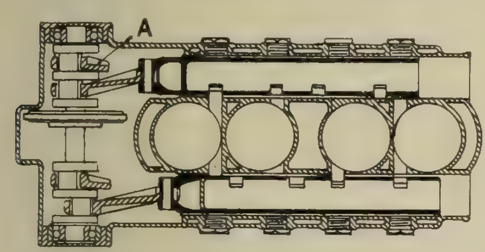
springs arranged in a telescopic casing. This is interposed at an angle between the axle and the frame, the compression spring being either in place of or in ad-

dition to the usual springs. The upper end of spring is anchored to a slide A, working on a screwed spindle, adjustable either by hand or automatically. For the latter purpose the spindle carries a pair of opposed ratchet wheels B, actuated by double-acting pawls C, on an arm reciprocated by a rod D connected to the axle. Thus when the car is overloaded, the frame moves downwards in relation to the axle, and the pawl rotates the spindle in one direction, so as to adjust the compression of the springs accordingly.

G. H. Hamilton. No. 27,150/09.

A Valve System.

On each side of the cylinders is arranged a horizontal chamber, carrying two sliding concentric piston valves. These have ports formed in them, and each is connected to a crank or eccentric on a transverse shaft. According to the

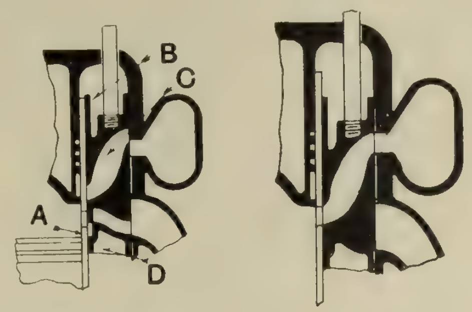


relative positions of each pair of valves, so the various ports are opened and closed, so that the ordinary cycle of strokes is obtained. The drawing is a sectional plan, showing the valve members, but the connecting rod A for the outer portions is in each case broken off.

H. S. and R. E. Morgan. No. 27,809/09.

A Slide Valve Engine.

This engine, which comes from the same source as the present Daimler engine, operates substantially on the well-known Knight principle, but its difference lies in the employment of a single ported sleeve valve A, within which the engine piston works. Outside this lies a sliding valve member B, having passing through it an inlet passage C, communicating with the induction pipe, whilst the exhaust gas is adapted to pass below the

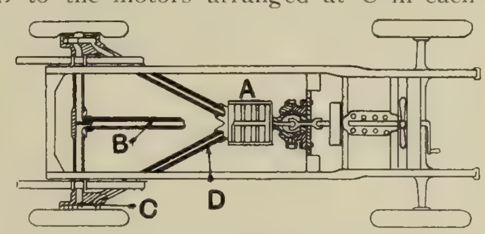


valve member B in the direction of the arrow. The sectional views show induction taking place on the one hand, and the exhaust valve just about to open in the other case. It will be noticed that the exhaust gas passes along the top edge of the water jacket D, so that the exhaust is cut off against a water-cooled surface, and the heated edge of the valve member B practically seats on the water jacket during an induction stroke. As in the Daimler engine, the inner valve member constitutes a periodic valve, and the outer a distributing valve.

C. Y. Knight. No. 21,645/09.

Hydraulic Transmission Mechanism.

The engine drives a pump A, which supplies the fluid through a supply pipe B to the motors arranged at C in each

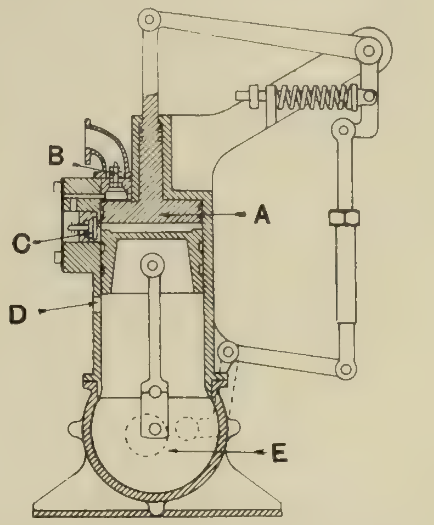


road wheel, the return being through the pipes D. The pump A is free to swivel and oscillate in a spherical bearing E, carried by a plate arranged across the car. Thus, the whole fluid system is free to oscillate and move freely, but is rigid in itself.

A. F. Rockwell. No. 8,352/10.

An Internal Combustion Engine.

In the working cylinder is arranged a supplementary piston A, which is forced downwards by the spring illustrated so as to follow, as far as possible, the move-



ment of the working piston. On the downward stroke gas is drawn in through the automatic valve B, into the space above the piston A. On the back stroke this gas is compressed, and finally injected, through a non-return valve C, into the space intermediate of the two pistons. Ignition takes place, and the working piston descends, but the auxiliary piston remains, owing to the pressure in the cylinder. Directly the exhaust port D is uncovered, the pressure falls, and the auxiliary piston A descends, scavenging the cylinder through the port D. To prevent the two pistons coming into violent collision, the auxiliary piston is arrested, by means of the link mechanism illustrated, coming into contact with an eccentric E, so that the engagement of the two pistons is effected gently. They are then caused to rise simultaneously, to effect compression of the charge of gas above the piston A, when the cycle is repeated, so that the engine operates on the two-stroke principle.

W. Woodland. No. 25,324/09.

THE OLYMPIA AUTOMOBILE EXHIBITION.

IN this issue of *The Automobile Engineer* no attempt will be made to deal with the chassis shown at Olympia, as everything concerning them will be contained in AUTOMOBILE ENGINEERING, the annual of *The Automobile Engineer*, which will be published on December 15th. The plan followed has been to divide all cars into five classes, the basis of comparison being the total volume swept out by the engine pistons per revolution, as it has been found that there is a strong similarity in the design of the chassis which come in each of the classes obtained in this manner. The cars will not be considered one by one as separate and complete in themselves, but engines, transmissions, frames, springs, axles, etc., will be dealt with in separate articles, class by class. Thus it will be easy for readers to obtain information concerning any part of any class of car, as regards general characteristics, without having to pick out paragraphs here, there and everywhere, while the parts of interesting chassis shown will be described in detail in each section, supplementarily to the general consideration.

However, apart from the chassis on the ground floor there were a number of gallery exhibits of interest to manufacturers, and it is proposed to deal briefly with these here.

Engines were shown by Aster, Ltd., and White and Poppe, Ltd., while there were also two new types, one being the Lamplough, which is described on another page in this issue, and another the "H" or Ellis engine, shown by T. B. André and Company. We hope to deal with the latter more fully later, but at present the illustration, Fig. I., will be given as sufficient explanation of the principle. The ordinary four-cycle system is used, and the claims are: extremely small weight for power, good balance, and low cost of production. Amongst the Aster engines there is one with a Coventry-chain-driven camshaft, rated at 16-20 h.p., and there are two other new models of 14-16 h.p. and 30-35 h.p., of a similar pattern, with pair-cast cylinders, enclosed valves, and two oil pumps. The latter are superimposed in a well at the bottom of the crankcase, and the upper one supplies a dashboard indicator only, being intended simply to show whether the level of oil in the base is sufficient to ensure a good supply to the lower pump, which supplies the main bearings under pressure. The big ends are supplied by spray from the main oil channels, which are cast in the crankcase, the ends of the crank pins being drilled and countersunk to catch the jet of oil. It will be understood that these new models are additional to the old types, which are still being made.

We published a list of the new White and Poppe engines in the September issue, and as the types fitted to several new chassis will be dealt with in the Annual, they need not be described here.

Bearings, of course, were there in great profusion, all the principal ball bearing manufacturers having almost complete exhibits of their automobile patterns. The Hoffmann Company have a new type of journal bearing similar to their ordinary form, so far as the bearing itself is concerned, but mounted in a loose ring, the outer ring of the bearing, and the inner surface of the housing being formed to a surface of a sphere, with its centre coincident with the axis of the shaft, so that the bearing is free to roll with any lack of alignment in the shaft. There were also a variety of Hoffmann parallel roller bearings. The Auto Machinery Company exhibited examples of all their small ball and roller bearings, and the Electric and Ordnance Accessories Company showed Timken taper roller bearings, as applied to almost every possible chassis journal.

Some good samples of die-cast bushes in Hoyt metal were shown by the company of that name, together with a small plant for their manufacture, which, we understand, is sufficiently cheap to be worth the serious consideration of any manufacturer who has need of a few hundred die-cast pieces in the course of a year, a pair of dies for a complete big end bush, with oil-ways and pegs, costing only about £2.

Axles and springs were exhibited by Smith, Parfrey and Co., Ltd., David Brown and Sons, Ltd., T. B. André and Co., Ltd., and B. M. and W. D. Fair, Messrs. Brown, of course, showing gears and other parts in addition. Gears

of all kinds were also to be found on the stand of E. J. Wrigley and Co., who are now manufacturing a very large quantity of worm gears for back axle work.

Body fittings and small brass fittings in endless variety, from float chambers to radiator caps, bonnet fasteners, petrol filters, etc., were shown by Benton and Stone. Messrs. Rotherams, Ltd., and Ross Courtney, Ltd., had similar exhibits, but had specially noticeable quantities of lubricators, oil pumps and oil pressure indicators.

John Marston, Ltd., exhibited their honeycomb radiators and fans, as did the Coventry Motór Fittings Co. and Orme, Evans and Co., the latter also having a few bonnets and wings. Some good flat tube patterns were especially prominent on the second-named stand, and Lamplough and Sons, Ltd., had samples of their extra light tubular radiator intended for either car or aeroplane work. This company also had a number of their excessively simple Albany pumps for water or oil circulation, and a new lock-nut washer, which possesses the advantage that it has no influence on the tightening or removal of the nut, which is rendered no more difficult by its presence.

Rudge-Whitworth detachable wire wheels were, of course, to be found in the gallery, but their presence was noticeable on the ground floor,

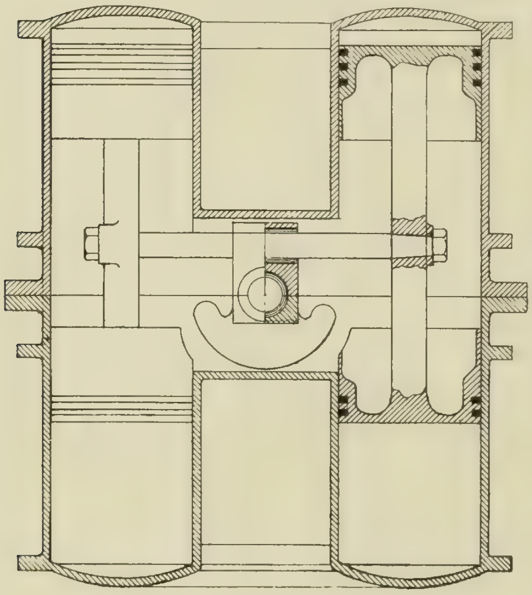


Fig. I.

as a large number of chassis were fitted with them The details of the device are precisely similar to the last year's patterns, but Figs. II. and III. are given to show the alternative methods of transmitting the drive from the inner to the outer shell. Sankey steel wheels, which have the appearance of a wood wheel, but are made from 18 gauge sheet steel, the two pressed halves being welded by oxy-acetylene, were also noticeable in the show, and seem to have gained in favour in the past twelve months.

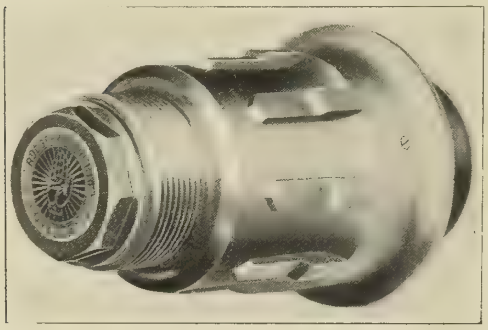


Fig. II.

To deal with the exhibits of ignition fittings adequately would require a volume, but in general it may be said that magnetos seem to tend to decrease in size, and there is also some inclination to provide an automatic advance, as is done most effectively on the Eisemann. The Lodge ignition has become even neater as regards equipment, without any loss of efficiency, but the accumulator manufacturers are turning much of their attention to car lighting equipment, as was obvious from an inspection of the Van Raden exhibit. The C.A.V. lighting dynamo was to be found on one or two cars, and lends itself to neat attachment, owing to its small size and the absence of external projections. Of course, the machine occupied a

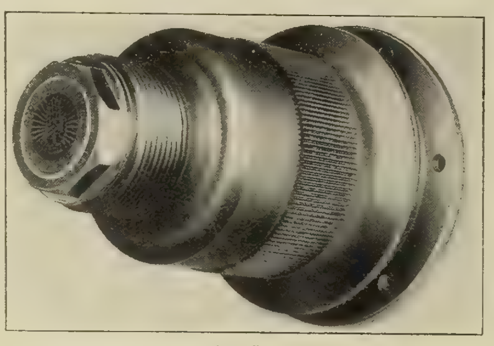


Fig. III.

prominent place on the maker's stand, sharing only with the C.A.V. magneto.

THE BANNER ENGINE.

THIS extremely ingenious engine, working on the four-cycle principle, but with a variety of unique features, has recently been invented, and was described briefly in "Recent Automobile Patents" in our September issue. It has been greatly simplified in a more recent design, and an engine that has been made is now in use in a car. At first sight the design is not particularly attractive, but it improves vastly on examination, and there is reason to believe that the engine will give a good account of itself when tested thoroughly. There are four cylinders, arranged in a single piece casting, as shown before, and a rotary valve of peculiar type turns in a central vertical cylinder. The valve has two ports separated by a web, and the timing is such that burnt gases always pass away beneath the web, while fresh mixture enters from above it. The valve runs at half crankshaft speed, and, of course, serves each cylinder in turn, so that there is a practically continuous stream of hot gas through the lower part, and cold gas through the upper half. This mechanism supplies all that is needed theoretically, and if the valve could be kept tight without any danger of its sticking or scoring, the engine would run satisfactorily.

The pistons are connected to short rods, which terminate in big ends on the beams, of which there are two, and each beam has a connecting rod, which transmits the motion to a two-throw crankshaft, with the cranks at 180°. As the valve feeds each cylinder in turn, the explosions are in similar sequence, and each crank therefore receives two impulses in a single revolution, while the two cylinders on the other crank are sucking and compressing.

To return to the valve gear, there are four poppet valves, in addition to the rotary valve, and their purpose is really simply to overcome the mechanical difficulty of making the rotary valve alone serve for all purposes. There is a cam at the bottom of the valve shaft, and this lifts each of the poppets just before the rotary valve begins to uncover the exhaust passage, on any cylinder. The poppet valve then remains open through the whole of the exhaust and inlet strokes, being cooled during the latter, and only closes when the compression stroke has begun. On the later stage of the compression stroke, and on the firing stroke, the poppet valve acts as a seal protecting the rotary valve, and it is found that the latter can be made quite a loose fit. As a matter of fact, it does not touch the walls of its container, even when hot, and needs no lubrication, as it is mounted in ball bearings at both the top and the bottom.

The foremost advantage is the fact that the whole engine is very compact, a four-cylinder of 4-inch bore and 5in. stroke being little more than a foot square by 2 feet deep. Next in order of importance comes free gas passage, as the poppet valves can open so early and close so late that they may be given a very high lift, so as not to detract from the large openings of the rotary ports. One of the chief claims is good balance, owing to the absence of angularity in the piston connecting rods, and to the fact that variations in acceleration caused by the angularity of the connecting rod between the beam and the crank pin, affect both the rising and falling pistons, on any one beam, to an equal degree. As regards this claim there is no doubt that a very roughly-constructed engine which has been tried in a car, runs smoothly enough at all ordinary speeds, though we have not yet had an opportunity of watching its behaviour at high rates of revolution. The trial engine also had every characteristic of the free port type, obviously giving good power at all low and medium speeds.

The next claim, which is silence in operation, is also borne out by the actual engine. It is obvious that a very stiff and rigid crankshaft (compared with the size of the cylinders) can be used, and that the engine as a whole is a simple piece of machining, while all the parts are accessible. The rotary valve, together with all its bearings, can be lifted out by hand after detaching the inlet dome and taking a pin out of the bottom universal joint on the drive shaft.

A NEW ALUMINIUM ALLOY.

A NEW alloy of aluminium has recently been introduced by Messrs. Gabriel and Co., under the name of "Clarus." It is a light alloy, of good colour, and the claims made for it are that it does not tarnish, and is of very considerably greater strength than the more common alloys. As regards the first claim, samples which we have received with a highly polished surface have retained their lustre for several weeks without any sign of oxidation. Concerning the strength, we give some figures supplied by the makers hereunder :—

Tube.	Tensile Strength in Tons per Sq. Inch.	Elongation.
Aluminium ..	10	6%
Clarus Alloy ..	16.70	Broke out of Limit.
Clarus Alloy ..	17.78	
Wire.	Lbs. per Sq. Inch.	
Aluminium ..	23 to 28,000	3%
Clarus Alloy ..	39,263	2.4 %
Clarus Alloy ..	39,463	2.4%

The metal is recommended for practically all purposes for which either other aluminium alloys of brass are used in automobile work, and is also said to be suitable for any kind of spinning or stamping.

CATALOGUES RECEIVED.

IRON AND STEEL TUBES.—John Spencer, Ltd., have a small and convenient list of their tubes and fittings. It contains some useful tables of weights of tubes and bars, screw threads, etc.

PULLEYS, etc.—The catalogue of the Unbreakable Pulley and Mill Gearing Co., Ltd., besides being very well arranged, and besides giving the fullest possible details concerning the company's manufactures, also contains a number of useful tables which are likely to be greatly appreciated. They include the power transmitted by shafting, conversion of circular and diametral pitch, circumferential velocity of pulleys, and proportions of many other portions of a power transmission system.

TYRE PROTECTORS.—The recently introduced Marshall tyre jacket, for which great durability is claimed, and which has performed well under test, is described and well illustrated in a booklet issued by The Marshall Tyre Fabric Co., Ltd.

CAR LIGHTING EQUIPMENT. — The C.A.V. new system of dynamo for car lighting is described very lucidly in a leaflet issued by C. A. Vandervell and Co., Ltd.

CARBURETTORS.—A leaflet descriptive of the Solex carburettor has just been issued, and it explains the operation of the device with great clearness. It may be obtained from the makers, S. Wolf and Co.

MISCELLANEOUS.

IN OUR COMMENT on the presidential address to the Institution of Automobile Engineers, published in our last issue, we remarked that but little time was left for discussion, and that none took place. This statement perhaps conveys a wrong impression, as it is not the practice of the Institution to arrange a discussion on presidential addresses, and such as have taken place in the past have been of an entirely informal character.

IN OUR RECENT description of the 14 h.p. Metallurgique, acting on information given by the agents in this country, we stated that the back axle case was of cast iron. We understand that a mistake occurred, as cast steel is invariably employed for this purpose.

THE

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Articles of a technical nature relating to the design or construction of automobiles for land, air, or water, will be carefully considered by the editor. Matter must be clearly written or typed on one side of the paper only, and a stamped addressed envelope must be enclosed for return. No responsibility can be accepted for the safety of contributions although every reasonable care will be taken.

Correspondence on interesting subjects, which come within the scope of this journal is cordially invited. All letters must contain the name and address of the writer, but not necessarily for publication. If a reply is desired by post enquiries must be accompanied by a stamped addressed envelope.

CONTENTS.

NOISELESS TRANSMISSION.

AUTOMOBILE engines have now been developed to such a state of efficiency that the once-neglected gearbox has become comparatively most unsatisfactory, principally on account of its noise-producing characteristics. Until quite lately opinion has been divided as to the exact cause of the noise made by spur gears when working under load, but it is now accepted generally that it is owing solely to inaccuracies in the teeth. Probably, however, the sounds arise from two different kinds of vibration, one irregular and caused by inequalities in the teeth: the other a steady note due to the stressing of the material of which the wheel is made at a successive series of points—as tooth after tooth takes up the load. Still it is unlikely that this second vibration is of any importance, as far the greater part of the noise must arise from the "hammering" of one tooth on another, that is to say, to the absence of theoretically accurate engagement and rolling. One of the most puzzling facts in connection with gearbox noises is that the influence of tooth size is far from clear, some makers being able to obtain the best results with coarse and some with fine pitches, while pressure angles and other possible variables seem to have no traceable effect either, at least within very wide limits. It is, of course, a fact that any lack of rigidity in the mounting of the gear wheels is likely to cause increased noise, and great trouble has been taken to shorten gear shafts with a certainly beneficial, but none the less very small effect.

These facts being as they are the action of certain French manufacturers in attacking the gearbox itself instead of the gears is particularly interesting. It is, of course, obvious that vibrations caused by the interaction of the working teeth are transmitted first to the gear wheels as entire pieces, thence to the shafts and, finally, to the outer air and the ear by vibration of the whole box. It is also conceivable that if the vibration of the box could be stopped the noise made by the gears would be so shut in as to be almost inaudible. The contention in the case in point is rather that the noise originated by the gears is actually increased by the box, and not merely transmitted, on account of the box acting as a sounding-board. If this is so it is still not easy to believe that any alteration in the shape of the box will have a marked effect unless it is also made much more solidly. If the natural periodic vibration of the box is an actual noise-producing factor all that can be done is to make the fundamental note as low as possible, so that the box cannot respond to the higher pitched vibrations of the gears. At the best such a remedy is merely palliative and is not a cure, so it seems regrettable that effort should be diverted from what is—and must always remain—the direct issue, namely the manufacture of gears which will run under load without irregular vibration.

Undoubtedly a certain amount of progress has been made, and it is instructive to find that the private experiments of the several firms who have investigated the matter scientifically have in almost every case reduced the problem to that of finding a method of gear manufacture which will give tooth forms accurate to something less than a hundredth of a millimetre. It is found that gears which run quite quietly under a given load, when they are new from the gear cutting machine, will perhaps cause great uproar as soon as they are hardened, owing, of course, to the warping which is bound to take place. As case-hardened steel warps more than air-hardened varieties of tougher alloy steels the latter are being tried by several makers and, in some cases, the improvement is distinct. Still it is safe to say that the improvement is not enough to make the displacement of case-hardening by any means certain, even though it may be probable. There is thus good reason to believe that either the present type of transmission will be modified by the substitution of an entirely different mechanism for the sliding change speed gearbox, or else a means will be devised whereby gears can be rendered accurate after hardening.

In the case of one British manufacturing concern the bevel back axle drives have been made quiet by a process of rubbing down all high spots on the teeth of every pair by hand with oilstone slips. Spur gears for automobile work cannot very well be so treated because of the much smaller size of the teeth and the great expense, so it therefore seems that the chief requirement is a tool for grinding small spur teeth as accurately as, say, the parts of ball bearings are ground. One or two machines have been invented for this purpose and some good work has been done on large gears, so it ought not to be beyond the powers of machine tool engineers to evolve the desired apparatus. Once gears have been made that will run quietly under load when mounted on an open jig or in a very massive gearbox it will be time to turn attention to the resonance of the casing and time to apply the same accuracy to the shafts as to the gear teeth. Obviously, however, every fresh process and every fresh accuracy calls for increased expenditure, unless a grinding machine can be designed which will allow the teeth to be gashed merely before hardening. Therefore it does not seem improbable that the chain drive change speed mechanism may come to be used a great deal in the near future.

CRANKSHAFT OSCILLATIONS AND THE TORSIONAL STIFFNESS OF CRANKS.

By H. Grinsted, B.Sc. (Eng.) and G. S. Bower, B.Sc. (Eng.)

It is a fact of common experience that some engines, especially those of the six-cylinder type, which run with almost perfect smoothness at most speeds, will give rise, over a small range of speed, to excessive high frequency vibration, the phenomenon seeming to be due to resonance between the forces at work in the engine and the elastic vibration of some portion of the mechanism of the car.

This vibration is of much higher frequency than the oscillation of a car upon its springs and as the effect is more noticeable and occurs at lower speeds with six-cylinder engines it appears probable that the crankshaft is the seat of the trouble.

Now a crankshaft can have three kinds of free vibration :—

(1) Longitudinal vibration, like a spiral spring with a mass suspended from it, consisting of elongation and contraction due to bending of the webs.

(2) Transverse vibration, like that of a bow-string when released, in which the shaft, webs, and crank pins bend.

(3) Torsional vibration, due to variation of twist from end to end when under a varying torque.

In addition to these, a disturbance might be caused by whirling.

Each of these modes of vibration will have its own period, and if rhythmic forces are applied to the shaft in a manner which will produce any vibration, then the oscillation will become of large amplitude when the period of force variation corresponds with the natural period of the vibration set up.

The only longitudinal, or axial, force on a crankshaft is the thrust of the clutch when an external spring is used, and as this force does not have a definite period of variation, which depends upon the engine speed, it is not likely to produce a dangerous oscillation of the crankshaft at certain definite speeds.

Transverse oscillation would be caused by the radial component of the pressure on the crank-pin. This kind of vibration is more likely to become violent in four-throw cranks, the cranks being all in one plane, than in the ordinary type of six-throw crankshaft, where three pairs of cranks lie in three planes, and each crank would receive impulses tending to make it vibrate in its own plane.

The possibility of whirling depends, to a great extent, on the rigidity of the bearings, and, for similar bearing conditions, it may be stated that the heavier the flywheel and the farther it is overhung the lower is the first critical whirling speed. Now a four-cylinder engine will require a heavier flywheel than a six-cylinder of the same power, as its torque variation is greater, so we should expect whirling to give more trouble in a four than in a six-cylinder engine of the same power. As trouble is not very often found with four-cylinder engines, it is probably not whirling that causes the oscillation in the six-cylinder type.

We will now consider the possibility of torsional vibration. In all cases of free torsional vibration the parts having the greatest moments of inertia about the axis of the motion will tend to remain steady; their movement will not be so great as that of portions with less inertia, so that in the case of a crankshaft with a flywheel at one end the rest of the shaft can be considered to oscillate with reference to the flywheel. It may not at first sight be clear how that any torsional oscillation of the crankshaft can make itself felt beyond the shaft itself, but it must be remembered that the balance of the engine depends upon the assumption that all portions of the shaft have the same angular velocity, and the rotating and reciprocating masses are only in balance so long as the angular velocity of all the cranks is the same. Now if while the shaft is rotating the cranks are oscillating about the flywheel, the angular speeds of the cranks, if plotted on a time base, would follow curves consisting of the sum of a horizontal line and waves of some sort, the waves representing the relative velocities of the cranks to the flywheel, which is supposed to rotate at a constant speed. Unless the velocity curve for all the cranks is the same the balance of the engine will be disturbed.

We will now investigate the nature of the torsional oscillations of more than one mass on a rod in as simple a form as possible, for although the results obtained from a simplified theoretical case cannot be directly applied to a practical problem of a very complicated nature, they form a guide, which often goes very far towards an approximate understanding of the more difficult cases.

Oscillations of Two Masses.

We will first investigate analytically the motion of two masses on a wire fixed at one end, and, for simplicity, will take the case of equal masses at equal distances along the wire as shown in Fig. I.

Let θ_1 = angular displacement of mass No. 1 from its initial position.

Let θ_2 = angular displacement of mass No. 2 from its initial position.

Let I = moment of inertia of each mass about the axis of the wire.

Let k = "stiffness" of wire, i.e., torque in wire = k × twist per unit length.

Then the equation of motion of No. 1 mass is :—

$$I\frac{d^2\theta_1}{dt^2} = -\frac{k}{l}\left(\theta_1 - \theta_2\right) \quad\ldots\ldots(1)$$

and that for the motion of mass No. 2 is :—

$$I\frac{d^2\theta_2}{dt^2} = \frac{k}{l}\left(\theta_1 - \theta_2\right) - \frac{k}{l}\theta_2 \quad\ldots\ldots(2)$$

The solution of these differential equations is :—

$$\theta^1 = C\cos\left(\cdot617\sqrt{\frac{k}{Il}}\,t\right) + D\cos\left(1\cdot62\sqrt{\frac{k}{Il}}\,t\right) \quad\ldots\ldots(3)$$

$$\theta_2 = \cdot62\,C\cos\left(\cdot617\sqrt{\frac{k}{Il}}\,t\right) - 1\cdot62\,D\cos\left(1\cdot62\sqrt{\frac{k}{Il}}\,t\right) \quad\ldots\ldots(4)$$

where the arbitary constants C and D are to be determined from the initial conditions of motion of the masses.

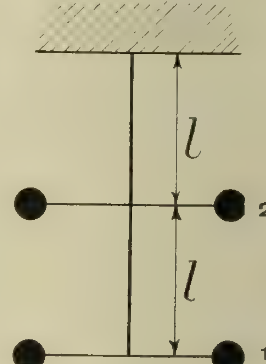


Fig. I.

It will be seen from equations (3) and (4) that the angular displacement of each mass is the sum of two simple harmonic displacements, of different periods, the amplitudes being proportional to C and D, and therefore dependent on the initial displacements of the masses.

Thus, if D is to be zero, we must have the amplitudes when t=0 in the ratio

$$\frac{\theta_1}{\theta_2}{}_{(t=0)} = \frac{C}{\cdot62C} = +1\cdot61$$

in which case, since this ratio is positive, the masses will move together with simple harmonic motion, and will have a common period of

$$\frac{2\pi}{\cdot617\sqrt{\frac{k}{Il}}}\text{ secs.} = t_1$$

No. 1 mass having a greater amplitude than No. 2. Similarly, if C is to be zero, we must have the initial displacements in the ratio

$$\frac{\theta_1}{\theta_2\,(t=0)} = \frac{1}{-1\cdot62} = -\cdot617$$

when the masses will move in opposite directions with simple harmonic motion, and will have a common period of

$$\frac{2\pi}{1\cdot62\sqrt{\frac{k}{Il}}}\text{ secs.} = t_2$$

It will be noticed that $t_1 > t_2$, i.e., the periodic time is greater when the masses swing together than when they move in opposite directions. When neither C nor D is zero, the motion of each mass is complex, and has no exact periodic time.

Experimental Verification.

In order to verify the results arrived at by analysis, we made experiments on the oscillations of two equal masses at equal distances along a wire clamped at the top (see Fig. I.).

The masses were made of variable moments of inertia, and were first of all adjusted so as to have equal moments of inertia by clamping each in turn to the bottom of the wire used in the succeeding experiments, and altering their radii of gyration until the time of swing of each one was the same.

These preliminary experiments also gave us the value of $\frac{Il}{k}$ for the masses and wire, for if T is the time of one complete oscillation of a mass at the end of a wire of length 2l,

$$\frac{Il}{k} = \frac{T}{4\pi^2 \times 2} = \cdot 432 \text{ in our case.}$$

Both masses were then clamped on the wire in their proper positions; the bottom mass No. 1 was given a displacement 1.6 times that of No. 2 and in the same direction, and the masses released. They vibrated together with simple harmonic motion, having a periodic time of about 7 secs., which is equal to t_1, inserting the value of $\frac{Il}{k}$ obtained as stated above, into the expression for t_1.

When the bottom mass was given a displacement of .62 times that of the upper one, but in the opposite direction, the masses vibrated with simple harmonic motion, having a common period of $2\frac{1}{2}$ secs., which is equal to t_2.

When each mass was twisted in the same direction through the same angle relative to the fixed end, both being released at the same instant, it was found that the motion of the masses was very complex.

The constants C and D in the equations (3) and (4) were determined so as to suit these initial displacements of the masses, the equations becoming :—

$$\theta_1 = 1\cdot 138 \cos \cdot 94t + \cdot 4335 \cos 2\cdot 46t \quad \text{.....(3a)}$$
$$\theta_2 = 1\cdot 84 \cos \cdot 94t - \cdot 2685 \cos 2\cdot 46t \quad \text{.....(4a)}$$

using the value of $\frac{Il}{k}$ already found. Fig. II. is plotted from these equations, and it was observed that the motion of the masses was similar to that shown by the curves.

If now we applied to the wire an alternating torque of period 7 secs., the masses adopted the motion corresponding to this

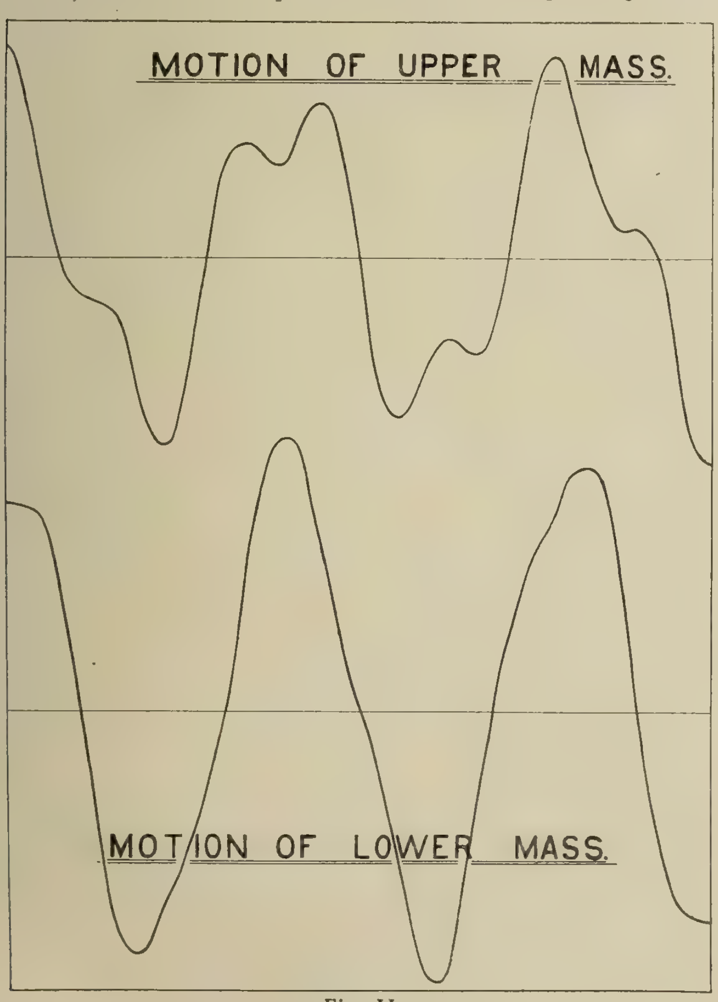


Fig. II.

period, i.e., they swung together with simple harmonic motion, the component of period 2.5 secs. being damped out. It was also found that by no means whatsoever could we obtain a longer period than that of 7 secs., which corresponds to all the masses swinging together. Owing to the stiffness of a crankshaft, its lowest frequency will be rather high, and, as an engine is required to run smoothly at all speeds up to its maximum, it is desirable that the lowest frequency of vibration shall be too high to become synchronous with torques likely to set up such oscillations. We are therefore chiefly concerned with the lowest possible frequency of crankshaft oscillation.

Oscillations of More than Two Masses.

Proceeding now to consider the motion of more than two masses on a wire fixed at the top, we made experiments with six masses, vibrating in various manners. It was again observed that the lowest frequency was obtained when all the masses were swinging together with simple harmonic motion, this frequency being smaller than that obtained when only two masses swung in the same manner. It seems probable that the greater the number of masses and the longer the shaft, the smaller will be this lowest frequency corresponding to all the masses swinging together, and this was confirmed by placing on the wire different numbers of masses of various moments of inertia at unequal distances along the wire. Several other frequencies were observed, at which the masses still moved with simple harmonic motion, but some of them in an opposite direction to the rest, it being possible to start any one of these frequencies, including the smallest one, by applying to the wire a torque of suitable periodicity.

There were also observed several other modes of vibration, in which the masses moved in a complex manner, but these vibrations, by analogy with the two-mass case, we may conclude to be composed of several components, whose periods are equal to those found in the cases of the various simple harmonic oscillations.

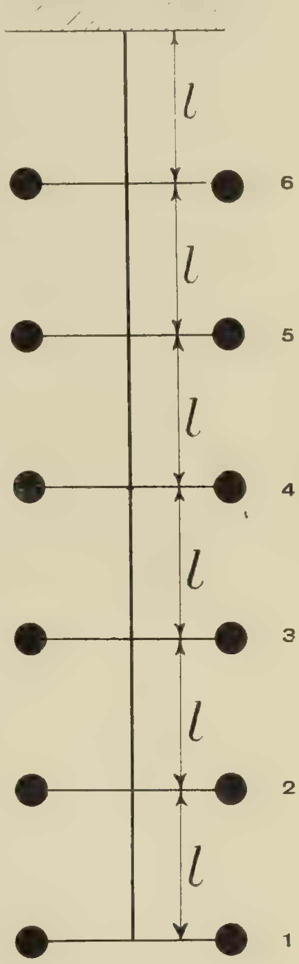


Fig. III.

In order to completely justify our belief that the lowest frequency of six masses is that which occurs when they all vibrate together, and to obtain a method of calculating this frequency, we will give a brief analytical investigation of the motion of six equal masses disposed at equal distances along a wire, as shown in Fig. III. To simplify the problem, we may assume each mass to move with simple harmonic motion, this assumption being justified by experiment as stated above, and the solution will give the initial displacements in order that the masses may move in this way.

Motion of Six Masses.

Let θ_1, θ_2, etc., denote the displacements relative to the fixed end of the various masses of equal moments of inertia, I. Since we assume that they all move with simple harmonic motion, we may write

$$\left.\begin{aligned}\theta_1 &= C_1 \cos pt \\ \theta_2 &= C_2 \cos pt \\ &\text{etc.}\end{aligned}\right\} \quad \text{.....(5)}$$

and therefore, differentiating these expressions twice,

$$\left.\begin{aligned}\frac{d^2\theta_1}{dt^2} &= C_1 p^2 \cos pt \\ \frac{d^2\theta_2}{dt^2} &= -C_2 p^2 \cos pt\end{aligned}\right\} \quad \text{.....(6)}$$

etc.

Where C_1, C_2, etc., are the amplitudes of vibration of the masses 1, 2, etc., p being the common frequency of oscillation.

Then the equation of motion of No. 1 mass is :

$$I \frac{d^2\theta_1}{dt^2} = -\frac{k}{l}(\theta_1 - \theta_2)$$

and, substituting for $\frac{d^2\theta_1}{dt^2}$, θ_1, and θ_2, in terms of cos pt from equations (5) and (6), this reduces to

$$\frac{Il}{k}p^2 = 1 - \frac{C_2}{C_1} \quad \text{.......... (7)}$$

the corresponding equation for mass No. 2 being

$$\frac{Il}{k}p^2 = 2 - \frac{C_1}{C_2} - \frac{C_3}{C_2} \quad \text{.......... (8)}$$

there being similar equations for the other masses. We thus have six equations, the solution of which gives p together with five ratios of amplitudes.

Put $C_6 = 1$, then, from these six equations all the amplitudes C_1, C_2, C_3 and C_4 can be expressed in terms of C_5, leaving one equation of the 6th degree to find C_5, viz. :

$$C_5^6 - C_5^5 - 5\,C_5^4 + 4\,C_5^3 + 6\,C_5^2 - 3\,C_5 - 1 = 0 \quad \text{...(9)}$$

Also, when C_5 is known,

$$\frac{Il}{k}p^2 = 2 - C_5 \quad \text{.......... (10)}$$

Equation (9) we have solved by trial, and each root substituted in equation (10) gives one real value of p, i.e., six in all.

The table gives the values of the ratios of the amplitudes corresponding to the various frequencies.

Values of $\sqrt{\frac{Il}{k}}\,p$. *i.e.*, relative values of possible frequencies.	C_1	C_2	C_3	C_4	C_5	C_6
1·942	—·64	1·53	—2·08	2·175	1·78	1
1·768	·565	1·2	·80	·29	—1·138	1
1·492	—·664	·835	·464	—·944	—·238	1
1·138	·879	—·258	—1·06	—·496	·71	1
·707	—1·41	—·68	·375	1·25	1·5	1
·224	4·36	4·05	3·51	2·8	1·95	1

Upon examination of these values of C, we see that for all values of p other than the smallest, some of the masses swing in opposition to the rest, and that the lowest frequency is obtained when all swing together.

Having shown that the first oscillation likely to be met in a crankshaft is that when all the cranks vibrate together, we will proceed to show how it is possible for the engine torque to produce dangerous oscillations.

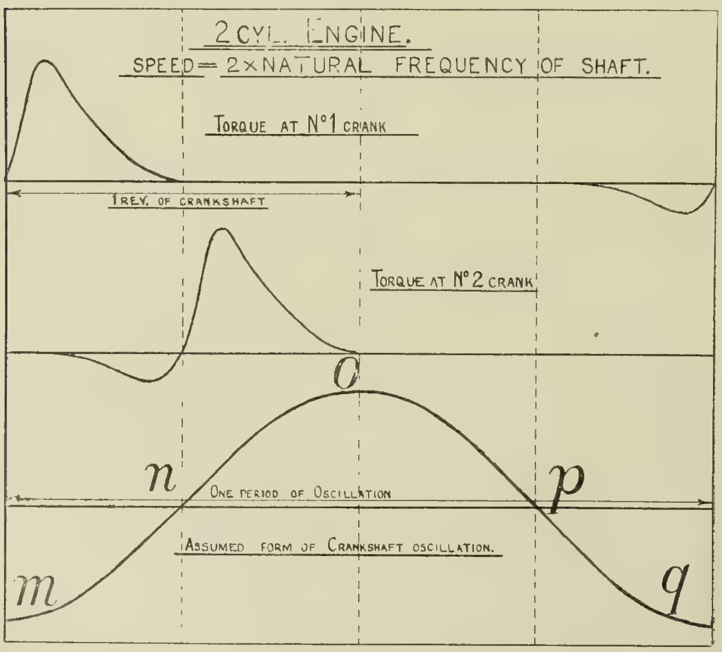


Fig. IV.

Two-Cylinder Engine.

First take the simple case of the two-cylinder engine; suppose its speed to be equal to twice the natural frequency of its crankshaft when the two cranks oscillate in phase. In Fig. IV. the top curve represents the torque on No. 1 crank, and the next that on No. 2 crank. Suppose that when No. 1 begins to fire the cranks are starting an oscillation in the direction of the torque, then the bottom curve may be assumed to represent the nature of the oscillation of the crankshaft about the flywheel. It may here be stated that displacements of cranks in the direction of positive torque are plotted above the mean line in all the diagrams. From m to n (Fig. IV.) the torque of No. 1 is helping the swing of the first crank, while the compression of No. 2 opposes the swing of No. 2 for the first quarter of an oscillation; from n to o No. 2 is helping, from o to p the shaft swings freely, and then from p to q No. 1 is again helping. Thus the two firing strokes and one compression stroke tend to help the oscillation, while one compression tends to damp it. It is evident that the oscillation will persist and grow in force. Thus a speed equal to twice the frequency of vibration of the crankshaft is one critical speed of a two-cylinder engine.

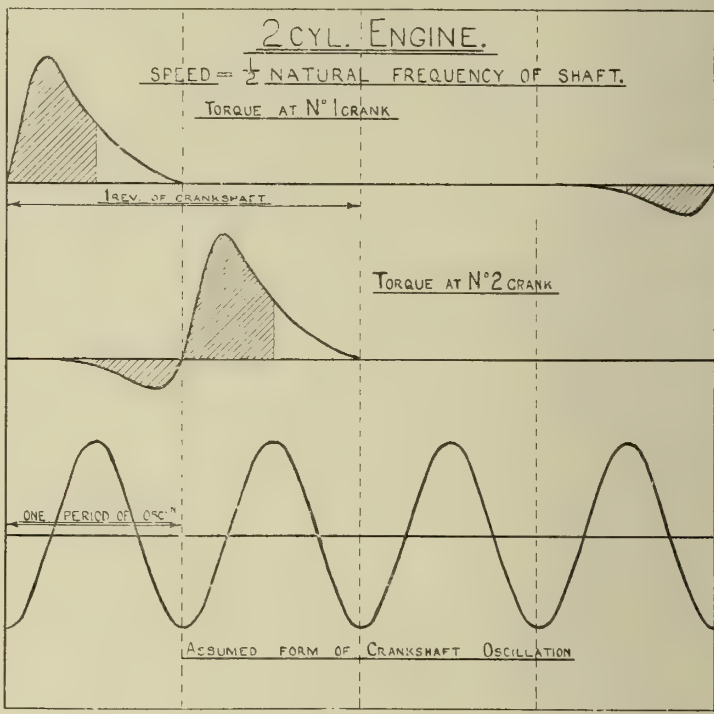


Fig. V.

Fig. V. shows the oscillation of the shaft when the engine speed is one-half this frequency, and the torque diagrams are shaded where the torque helps the oscillation and left plain where opposing it. It is clear that this would also be a critical speed, i.e., one-half the lowest natural frequency of the shaft.

Four-Cylinder Engine.

Let us now take the case of the four-cylinder engine. In Fig. VI. the torque diagrams of the four cylinders are shown above, and a curve of crank oscillation, all cranks being supposed to move in phase, beneath. The shaded portions of the diagrams represent the torque which tends to assist the oscillation, and they are clearly in excess of the parts tending to damp it out. The diagram shows that a speed of one-half the natural frequency of oscillation is dangerous.

Six-Cylinder Engine.

Fig. VII. shows the torque diagrams for three of the cylinders of a six-cylinder engine, and a vibration of the cranks, the engine speed being one-third the frequency of this vibration. The torque diagrams are only shown for three cylinders, as the others obviously act in the same way. Thus for a six-cylinder engine a speed of one-third the lowest natural frequency of the crankshaft will cause large oscillations.

With reference to Figs. V., VI. and VII., it may be objected that were the torque diagrams of a slightly different shape, the unshaded parts would have more effect than the shaded parts; however, if they did, there would still be a danger of oscillation, but the curve representing the crank movement would be advanced some portion of a period on that shown, so that the open portions of the torque curve would help the vibration, and the shaded portions oppose it. The speeds given above would remain dangerous speeds, for the crankshaft oscillation would gradually be forced into the time relation with the engine cycles that would give the greatest effect.

Of course there are other speeds, such as one-sixth, or one-ninth the frequency of the shaft for the six-cylinder case, which may set up slight oscillations, but at these speeds the excess of helping torque over damping torque can never be very great.

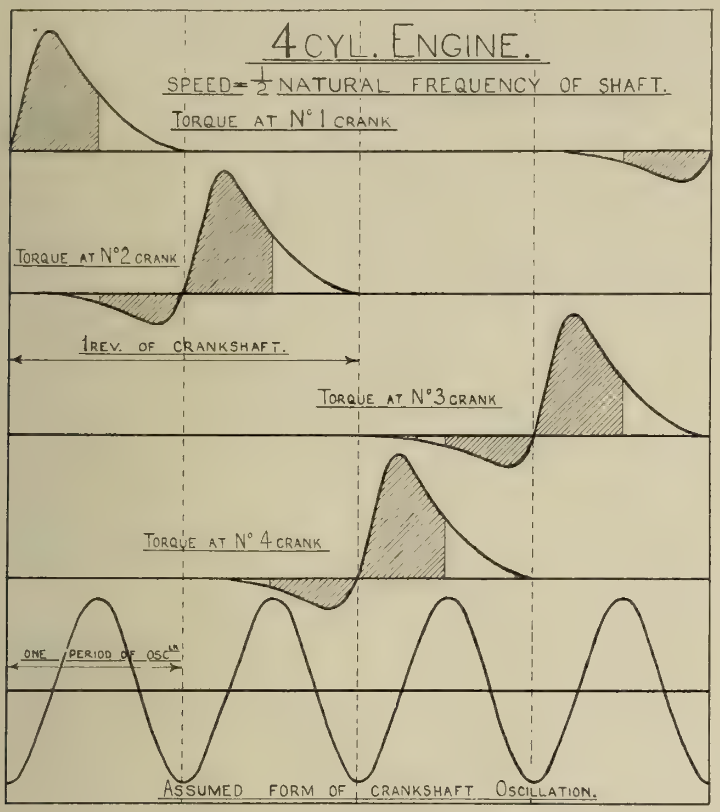


Fig. VI.

It is thus seen that a six-cylinder engine will have a dangerous speed of one-third the lowest natural frequency of its crankshaft, while a four-cylinder will have a dangerous speed of one-half the lowest natural frequency of *its* crankshaft; moreover, the lowest natural frequency of a six-throw crankshaft is less than that of a four-throw, so it is clear that a six-cylinder engine may give trouble at a much lower speed than a four-cylinder of

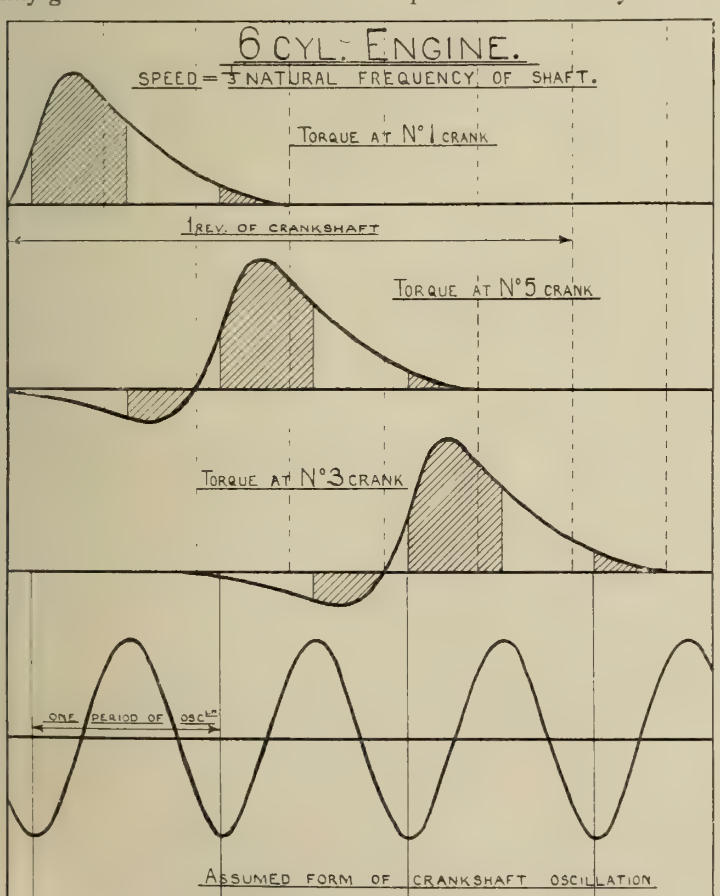


Fig. VII.

the same power. For example, suppose the lowest frequency of a four-throw crankshaft is 4000 per minute, and that of a six-throw is 3,000, then the four-cylinder will run without trouble up to 2,000 r.p.m., while a six-cylinder will encounter it at 1,000.

Effect of Reciprocating Masses.

In the above investigations the crankshaft has been thought of as a rod with masses attached to it; the practical case of a crankshaft with reciprocating masses cannot, however, be very nearly likened to such a rod with masses of constant moment of inertia, but a complete analysis of the motion of a crank rotating and at the same time oscillating, with a reciprocating mass connected to it by a short connecting rod, would be rather complicated and could at best be only approximate. It seems probable, however, that although the effect of the reciprocating masses may be to damp out the torsional oscillations at most speeds, yet at critical speeds, when the frequency of vibration is some exact multiple of engine speed, the reciprocating masses will not alter the general form of the curve of crank oscillation, but will modify it in the same way over each engine revolution. The resultant crank movement would follow a complex wave with a complete period equal to the time of one revolution.

Methods of Damping the Oscillation of the Crankshaft

Some makers of six-cylinder engines have tried to prevent this dangerous torsional oscillation of the crankshaft by placing a fly-wheel at the front of the engine as well as at the back; but this only makes matters worse, for it reduces the lowest critical frequency. If the supplementary flywheel were driven through a friction drive of some sort, which would allow slipping to take place when called upon to rapidly accelerate the flywheel, better results would probably be obtained.

A spring drive for the camshaft has been suggested for dealing with the difficulty, but this does not prevent the oscillation and its accompanying effects on the crankshaft and bearings, it only protects the camshaft to some extent from forced vibrations; this can, however, be more easily done, as some have recognised, by placing the half-time drive at the flywheel end of the shaft.

Torsion of Cranks.

In the preceding work we have considered the simple case of a plain, circular shaft, with several masses attached to it. In practical problems, however, we have to apply our results to the case of a crankshaft, consisting of webs, journals, and crank-pins, and we must be able to find the dimensions of a plain, uniform shaft, which shall be equivalent, so far as torsion is concerned, to the actual crankshaft; that is to say, any portion of this equivalent shaft between two masses must be as stiff as the corresponding portion of the real crankshaft, their twists being the same for the same torque.

In the following, we shall assume that the shaft is so very slightly loose in its bearings that the reactions of the latter on the shaft pass through the ends of the bearings. Then the bearing reactions must evidently be as shown in Fig. VIII., consisting of forces P and Q, which may be reduced into a force F, together with a couple, at each bearing.

Then, for equilibrium of the shaft as a whole, we must have

$$P(a + 2b) = Q(a + 2b + 2l)$$

$$\text{or } Q = \frac{P(a + 2b)}{a + 2b + 2l} \quad \text{i.e. } Q \text{ is} < P$$

Now at the central point, O, of the crankpin the bending moment is

$$M_0 = P\left(\frac{a}{2} + b\right) - Q\left(\frac{a}{2} + b + l\right) = 0$$

Also, the resultant force at this point will be

$$F = P - Q = P\left(\frac{2l}{a + 2b + 2l}\right)$$

Hence, so far as these forces are concerned, the crank-pin may be regarded as two cantilevers loaded at the middle, O, and fixed at the ends, C and D.

Therefore, the deflection of one-half of it, of length $\frac{a}{2}$ will be, using the ordinary formula for the deflection of a cantilever,

$$\frac{F\left(\frac{a}{2}\right)^3}{3EI} = \frac{1}{2}y_1 \quad \text{(See fig. IX.)}$$

$$\text{Therefore } y_1 = \frac{Fa^3}{12EI} \quad \dots\dots\dots\dots (11)$$

where I = Moment of inertia of a cross section of the crank-pin about a diameter.

In order to explain more clearly the method adopted for dealing with the problem, we will first consider the behaviour of a crank with perfectly rigid webs and journals, supported in perfectly rigid bearings, which constrain the journals to remain coaxial, and we will afterwards indicate the changes which must be made, to allow for the bending and twisting of the webs (see Fig. IX.).

Due to the deformation of the crank-pin, supposing the web BC to be fixed, the web ED will rotate slightly, D moving to D[1]

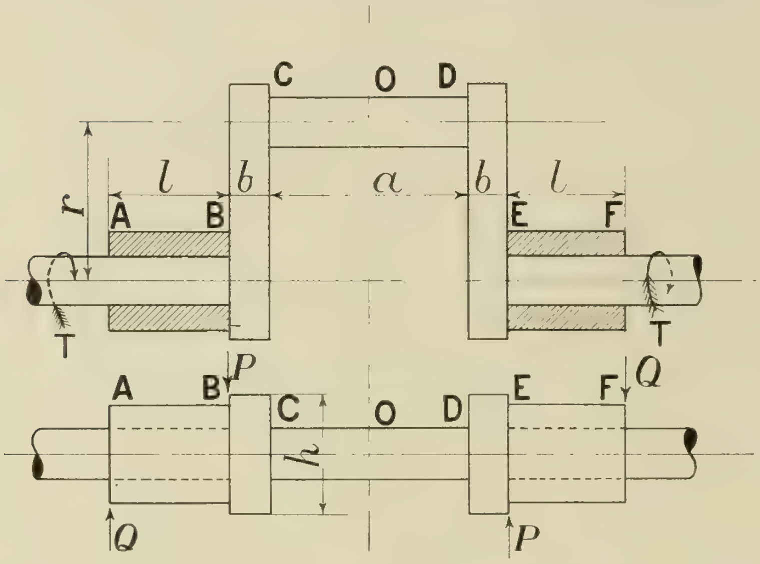


Fig. VIII.

and the angle DGD' = θ measuring the angle of twist of the whole crank. Thus the angle of twist of the crank-pin is

$$\theta = \frac{D\,D^1}{r} \text{ approximately.}$$

But the torque causing this twist is equal to

$$T - Fr$$

$$\text{Therefore, } \theta = \frac{a}{C\,J}\left(T - Fr\right) \quad \text{.................................(12)}$$

J being the polar moment of inertia of a section of the crank pin, and C being the modulus of rigidity of the crank-pin.

Also, $DD^1 = y_1$, since this movement of D is due to the deflection of the crank-pin under the influence of the bearing reactions.

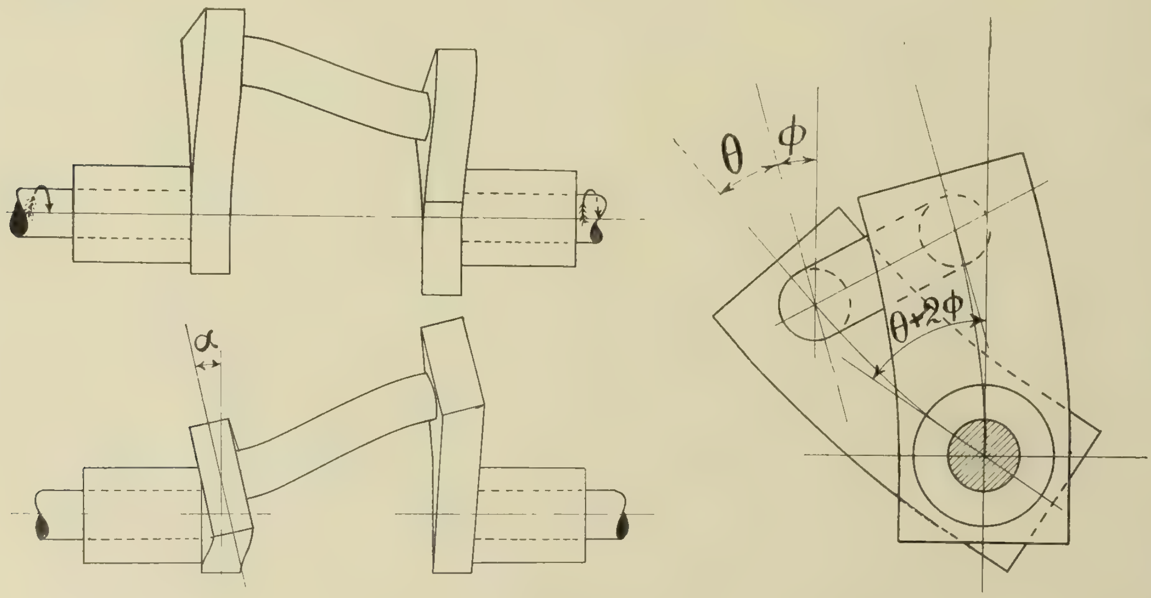


Fig. X.

$$\text{Hence } \frac{a}{C\,J}\left(T - Fr\right) = \frac{Fa^3}{12 E\,I_1}$$

$$\text{or } F = T\,\frac{6\,Er}{Ca^2 + 6\,Er^2} \quad \text{...(13)}$$

Knowing F, θ can at once be calculated from equation (12), thus giving the twist of the crank under the given torque T.

Allowance for the webs.

Due to the twisting of the webs, the ends C and D of the crank-pin, instead of being parallel to the axis of the journals, will each be twisted through a certain angle α, Fig. X., which angle may be calculated approximately in terms of the bending moment, M_c, at the centres of the webs.

$$\text{Thus, } M = Q\left(l + \frac{b}{2}\right) - P\frac{b}{2}$$

$$= F\left(\frac{a + b}{2}\right)$$

$$\text{Also } \alpha = \frac{40\,J_w\,r}{b^4\,h^4\,C}\,M_c$$

Where J_w = the polar moment of inertia of the web section

$$= \frac{b\,h}{12}\left(b^2 + h^2\right)$$

$$\text{So that } \alpha = \frac{40\,(h^2 + b^2)}{12\,b^3\,h^3}\,\frac{r}{C}\,M_c = K\,M_c \text{ say.}$$

The problem of finding the deflection of a crank-pin now becomes that of finding the deflection of a cantilever with its end built in at a small angle α with the horizontal.

Therefore, the total deflection of the crank-pin will be

$$y_1 = \frac{Fa^3}{12\,E\,I} + a\alpha$$

$$\text{or } y_1 = \frac{Fa^3}{12\,E\,I} + a\,K\,M_c \quad \text{...(14)}$$

The webs will bend in a plane at right angles to the axis of the shaft, owing to the effect of the torque T and force F.

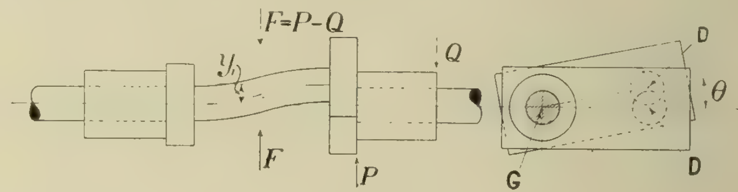


Fig. IX.

If y_w is the deflection of the web due to these, and ϕ is the change in inclination of the ends, then it may easily be shown that

$$y_w = \frac{r^2}{E\,I_w}\left(\frac{T}{2} - \frac{Fr}{3}\right)$$

$$\text{and } \phi = \frac{r}{E\,I_w}\left(T - \frac{Fr}{2}\right) \text{...(15)}$$

The twist of the crank-pin will be as before,

$$\theta = \frac{a}{C\,J}\left(T - Fr\right) \text{.........(16)}$$

and the total twist of the crank will be $\psi = \theta + 2\phi$

Also, to find F, and therefore P and Q, we have

$$y_1 = r\,(\phi + \theta)$$

which, substituting for y_1, ϕ, and θ, from equations (14), (15), and (16) becomes

$$F = T\,\frac{\dfrac{r^2}{E\,I} + \dfrac{a\,r}{C\,J}}{\dfrac{a^3}{12\,E\,I} + \dfrac{Ka\,(a+b)}{2} + \dfrac{r^3}{2EI_w} + \dfrac{r^2a}{C\,J}} \quad \text{..........(17)}$$

Knowing F, we can at once calculate ψ, the total angle of twist of the crank, by substituting for F its numerical value in the expression :—

$$\psi = \theta + 2\phi$$

$$-\frac{a}{CJ}(T - Fr) + \frac{2r}{EI_w}\left(T - \frac{Fr}{2}\right) \quad \ldots\ldots\ldots\ldots\ldots\ldots(18)$$

To exemplify the use of the above analysis, we will calculate the angle of twist of an actual crank, using equations (17) and (18), for the torque to which it will be subjected in practice, and will then compare the result thus arrived at with the result of the approximate theory.

We will take a crank whose dimensions are

$$a = 2{\cdot}26'' \quad b = {\cdot}67'' \quad h = 2{\cdot}68''$$
$$r = 2{\cdot}56'' \quad l = 2{\cdot}56''$$

Diameter of crank-pin = 2.36in., bored out to 1.45in. diameter. The maximum torque, to which this crank would be subjected, would be T = 4,200 lb. inches. Assuming $C = \frac{2}{5}E$, from equation (17) we find :—

$$F = 382 \text{ lbs.}$$

Inserting this value of F in equation (18), we find

$$\psi = \frac{9865}{C} \text{ radians.}$$

The angle of twist of a crank not constrained in bearings will be, neglecting the effect of the webs,

$$\psi = \frac{3630}{C} \text{ radians.}$$

Again, working on the theory which assumes the webs to be rigid, from equation (13) we find

$$F = 1560 \text{ lbs.}$$

$$\text{and } \psi = \frac{173}{C} \text{ radians.}$$

It will be noticed from the examples that the effect of the bending and twisting of the webs is to very much reduce the force F and to greatly increase the angle of twist ψ.

This shows how important it is to make the webs of a crankshaft as rigid as possible.

Engines without Intermediate Bearings.

Similar treatment can be applied when there are more cranks than one between two consecutive bearings.

In the case of a two-throw crankshaft with cranks opposite and with bearings only at the ends there is no tendency for the end journals to get out of line, and hence there are no bearing reactions due to the torque.

Thus the full torque has effect in twisting the crank-pin and bending the webs, whereas, when three bearings are used, the reactions at the bearings diminish the effective torque on these members, and thus the total twist for a given torque will be less than when no intermediate bearing is used.

This result can be applied to a four-throw crankshaft, which can be regarded as consisting of two two-throw crankshafts.

Thus, in a four-cylinder engine with only three bearings the stiffness of the crankshaft will be less than when five bearings are used, and its lowest torsional frequency will be less.

CARBURETTOR ACTION.

By Robert W. A. Brewer, A.M.I.C.E., M.I.M.E., M.I.A.E., F.S.E.

THE interesting paper read by Professor Morgan before the Institute of Automobile Engineers, on November 9th, 1910, called for some remarks from me at the time, but the matter is of so much importance that an explanation of my statements, accompanied by diagrams, will help to make clear my views upon this subject. Professor Morgan's method of testing the flow of petrol through small orifices, by means of a motor-driven engine, is more elaborate than the methods I originally employed, and is more likely to be accompanied by errors on account of the tortuous path of air and gas flow. At the same time, the observations have been carefully made and corrections for leakage taken into account, but in his deductions from the results obtained, I disagree with Professor Morgan, particularly with regard to what I term his "surging flow" theory. The curves he gave were plotted slightly differently to mine, as will be seen on referring to the abscissae and ordinates, and I claim that my method is the better one of the two.

When I commenced to investigate carburettor problems seriously some years ago, I used an old type Longuemare carburettor, diagrammatically depicted in the accompanying Fig. I. In this particular carburettor the area of the inlet at the small end of the taper shown is 4.9 square centimetres, i.e., the 25 mm. diameter taper increasing to 34 mm. diameter in the vicinity of the jet. The outlet to the engine was 30 mm. diameter, the aperture being shown on the right of the diagram, and fitted with a flange. In these early experiments I took direct measurements of the suction produced by the engine, the observation point being situated close to the jet in the intake. This carburettor was fitted to an engine having four cylinders 90 mm. diameter by 110 mm. stroke. This same engine was employed throughout the series of investigations, with various carburettors, but in the work under discussion other engines of different sizes and powers were also used, to obtain further data and make the whole investigation more complete. By calculating out the piston displacement at 100 per cent. cylinder filled, curves were obtained showing the velocity of air, and the suctions were calculated on various areas while throughout the investigations I have adopted constants allowing for wire drawing, etc., affecting the quantity of mixture entering the cylinders at the suction stroke, the smallest constants being 0.75 at engine speeds of 1,300 revolutions per minute.

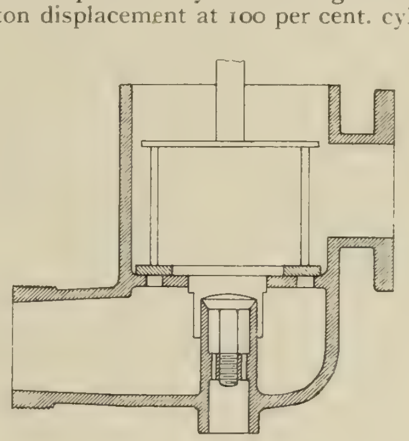


Fig. I.

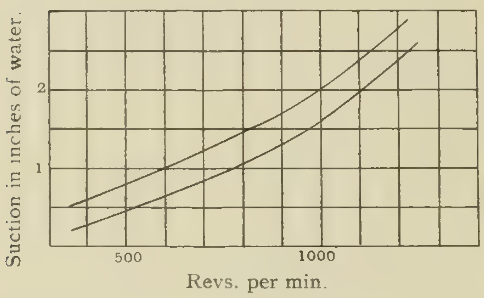
Curve "E" shows the suction in inches of water, at the carburettor orifices, at engine speeds varying from 350 to 1,250 revolutions per minute, and the fact that the experimental curve lies higher than the calculated one is accounted for by the presence of the observation tube partly restricting the air inlet. Taking the dimensions of this engine as given, the results of a single road test will give an indication of the methods of procedure in the preliminary work. The spirit used was Borneo 0.760 sp. gr., the approximate composition 91 per cent. "C," 9 per cent. "H." The amount of air theoretically required for a complete combustion = 13.64 lbs. per lb. of fuel = 182 cubic feet of air at 62° Fahrenheit.

During the test of 20.5 miles the engine made 38,700 revolutions, and consequently twice that number of cylinder charges were admitted. The cylinder capacity was 698.5 cc., swept by a piston, and one gallon = 4,543 cc. The total petrol used was 3,860 c.c. net, and by a simple calculation we find that the total volume of mixture at 14 lbs. per square inch and 110° Fahrenheit = 54 million cc., which reduced to 15 lbs. per square inch and 62° Fahrenheit = 46.7 million cu.cms. The relation, therefore, between the liquid petrol and the air by volume = 8.25 volumes of petrol per 100,000 volumes of mixture. Working this out further, we find that 59 volumes of air are mixed with one volume of heavy petrol vapour = 1.7% mixture of vapour to air. Theoretically, with petrol of 0.700 s.g., the best mixture is 1.86%, and with petrol of a s.g. 0.680 the best mixture is 2.5% for perfect combustion without previous compression (Redwood).

The foregoing results were taken at practically constant engine speed, the speed of the car averaging 20 miles per hour, and the apparatus used being fairly crude. A Claudel Hobson carburettor was then fitted to the engine, and a large number of tests carried out, both with this carburettor operating in the ordinary way with various jets, also in a laboratory using a special instrument, which I designed for the purpose. I at once found out that the temperature of the liquid considerably affected the behaviour of the jets, and I made an investigation, the results of which are shown in curve "B." This indicates what corrections should be made in cases where the petrol is previously heated before passing through the carburettor jet orifice.

In practical language, it may be taken that if my curves and constants are used, they apply to petrol at a temperature of 55° Fahrenheit, but at working carburet-

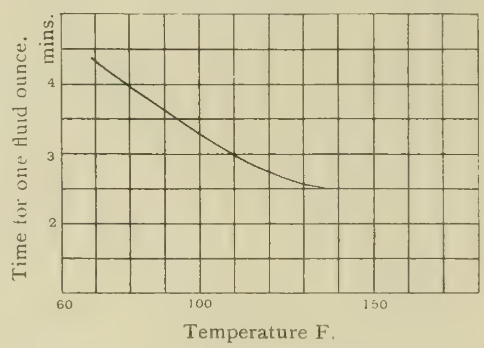
Curve E.



tor temperatures these effluxes are 10% higher; in other words, where a certain jet is shown by the curves as discharging the correct volume of petrol at 55° Fahrenheit, the carburettor should be fitted with a jet 0.05 mm. less in diameter. It is a well-known fact that the discharge of petrol from such a jet (and in this case the restricted orifice is 5 mm. long) is affected by the friction of the orifice, which tends to prevent the correct quantity of petrol flowing at low suctions, and also when the air velocity past the jet reaches a high value, the discharge from the jet becomes more than directly proportional to the air velocity. My curves, which are given, show the relation between the discharge from the jet orifice, and the velocity of air which passed the jet in a direction parallel to the flow of liquid agree generally with those given by Professor Morgan, but my curves were obtained in the first instance in a vacuum tube fitted with special means for regulating the suction upon the jets with considerable accuracy.

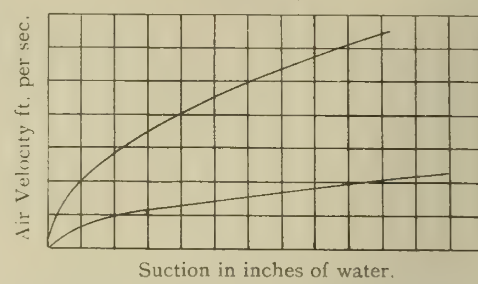
I will first proceed to describe the vacuum instrument, as it had been suggested to me that probably the discharge from the jets into a pressure below that of the atmosphere might differ in some respects from a discharge into ordinary atmospheric pressure, where the flow of

Curve B.



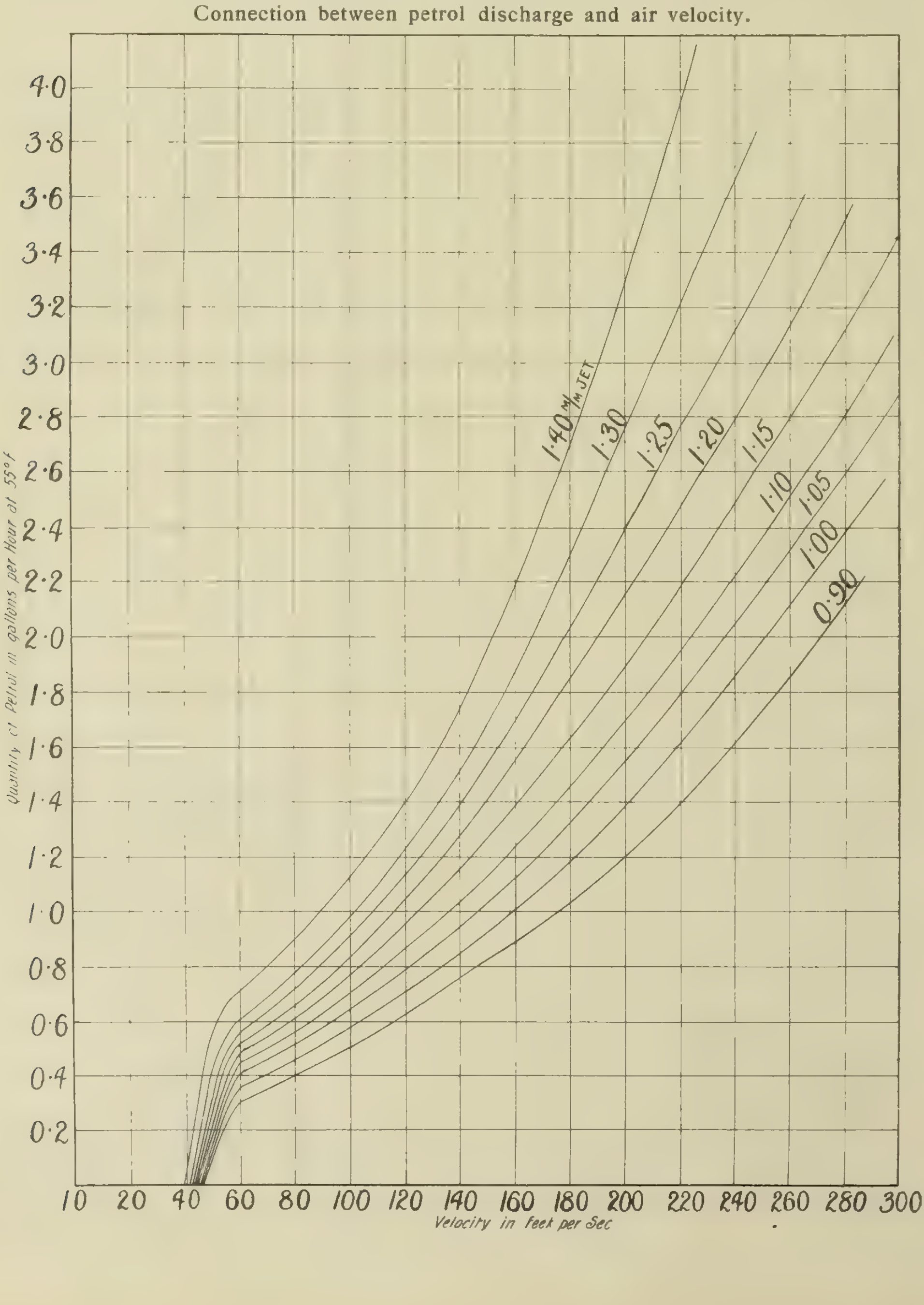
petrol was produced by a liquid head over the orifice. As a matter of fact, I did not find this to be the case within the limits of experimental accuracy. The vacuum instrument consisted of a long vertical glass tube, having at its upper

Curve D.



end two outlet tubes of small diameter projecting from it. The lower end of the tube fitted into a brass casting, which formed a seating for the jets, and also had branch pipes leading from it in three directions. Firstly, there was the inlet for the petrol, and, secondly, an outlet for the petrol which had passed through the jet, both governed by special types of air-cocks. There was also a removable plug, through which any jet could be withdrawn after testing. To the top of the instrument was attached a pipe leading from a vacuum pump, and also a second pipe connected to a "U" gauge for registering the extent of the vacuum in the tube, and at the lower end of the instrument a small petrol tank was connected up, the level of the top being equal to that of the jet orifice.

Connection between petrol discharge and air velocity.



Various jets were tested in this instrument, the suction head was varied in intervals of two inches, and the observations calculated out and plotted in curves. Further observations were made with the same jets screwed into the end of an upturned pipe of "J" shape, the jets being fitted to the smaller leg of the "J," and a constant level tank attached to the long leg.

At all the lower values up to 150 mm's. of fuel head, Mr. A. S. E. Ackermann, A.M.I.C.E., found that my coefficient of discharge only varied about 4%, i.e., from 0.738 to 0.770, the mean coefficient being 0.75, which agreed approximately with that given by Professor Unwin (on page 88 of his "Treatise on Hydraulics") for small orifices where the ratio of the length of the orifice to its diameter was five or more. The formula applied in this case is :—

$$Q = c\omega\sqrt{2gh}$$

where

Q was the discharge in cubic cm. per sec.

c was the coefficient of discharge.

ω was the area of the orifice in sq. cm.

g was the acceleration of gravity in cm. per sec. per sec.

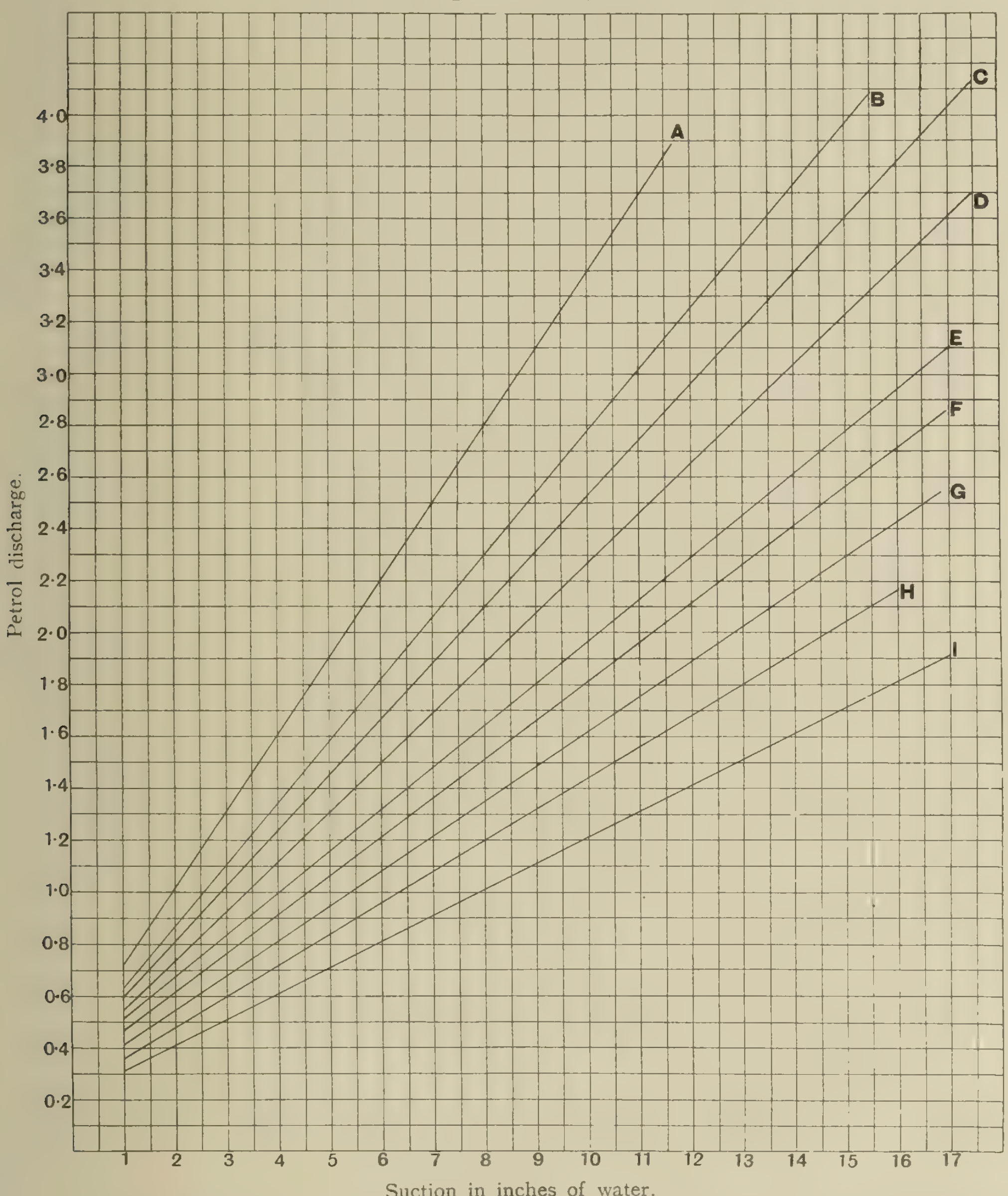
h was the head in cm. over the centre of the orifice.

The curves showing the relation between petrol discharge and air velocity have undergone a large amount of verification and amplification from time to time, as my various tests have provided me with further information during the past three years. Tables which I have founded on these curves are in daily use in connection with the fitting of a certain type of carburettor to various engines, and have proved eminently satisfactory.

More than two years ago, on an occasion when I publicly explained my system, I was severely criticised by a well-known motor engineer, who stated, amongst

other things, that, in spite of what I had done, my investigations "did not enable one to make a carburettor without having to do a good deal of experimenting to get it quite right for any particular engine," and that "he should like to warn the author that he (the author) aspired to too much, if he wished it to be supposed that those who had done the practical work in the past in producing the best carburettors were to be helped from a condition of rule of thumb to one of theoretical accuracy in building up a carburettor for a given purpose," and also "that certain some of which give variations of 50% in the proportioning of the mixture. A study of the drooping curves, or, in my case, rising curves, as my abcissae and ordinates are reversed as compared with Professor Morgan's, led me to combine some of these curves with what I show as curve "D," this being the relation between suction and air velocity, and a series of curves, which lie along straight lines, resulted. These curves are very interesting, and they agree with Professor Morgan's straight line parts of the characteristics, the formula for which he

Petrol Discharge Plotted against Suction.



The Jets corresponding to the curves have the following diameters: A 1·4 mm. B 1·3 mm. C 1·25 mm. D 1·2 mm. E 1·15 mm. F 1·10 mm. G 1·05 mm. H 1 mm. and I 0·9 mm.

experiments the author claimed agreed very fairly, and were within 10% of this and that."

The past two years have proved the practical utility of my work, and that the foregoing remarks were not in any way justified, as practical workers had been materially assisted during that time, and carburettors fitted acording to my specification have absolutely proved to be right the first time, without all the extra work my critic referred to.

I go further in claiming that the results obtained by my method are certainly within 10% of giving a correct mixture, and this is far and away better than those shown in many curves in the paper for other carburettors, which are no doubt claimed to be mechanically correct, but quotes as

$$y = ax + b.$$

In my case, however, I give a definite value for the discharge of petrol from an orifice such as I have described, and taking three cases in point, I used a formula

$$Q = aV + b,$$

where Q is the quantity of petrol delivered in gallons per hour, and V equals the vacuum in inches of water at the jet orifice. Three equations are given below as follows:—

Jet diameter 1.00 mm. $Q = 0.12V + 0.24$.
,, ,, 1.20 mm. $Q = 0.19V + 0.38$.
,, ,, 1.40 mm. $Q = 0.3V + 0.4$.

The above equations are given to show the simplicity of the calculation required to ascertain the flow of petrol when any given vacuum is present in a carburettor in which the vacuum varies, but they are also useful for application to another type of carburettor which is becoming more popular, namely, that in which a constant vacuum is aimed at. On the face of this argument, it seems somewhat difficult to understand how some of the claims of the designers of constant vacuum carburettors can be substantiated. For instance, we read of constant vacuum types in which the pressure of the air inlet is caused to vary with the demand of the engine, which means that the quantity of air passing the jet, either in its direction of flow or at right angles to it, varies in like manner, but at the same time the vacuum is constant. My curves, therefore, do not apply in relation to air velocity, but solely to the constant vacuum, although, of course, it may happen that in certain arrangements of jet relatively to the air stream the efflux of petrol may be increased by reason of the air stream, but from ordinary reasoning it would be somewhat desirable to increase the dimension of the petrol orifice in proportion to the quantity of air passing, and consequently as the engine speed increases.

It will thus be seen that in types of constant vacuum carburettors, in which a moving spindle operates a valve, so that the area increases in a certain ratio either in direct proportion to lift or movement, a connecting arrangement between the air valve spindle and the spindle operating the jet-varying devices can be arranged so that the area of petrol orifice can be varied in proportion. This can be calculated from my curves of petrol discharge. It is somewhat difficult to give an equation relating the discharge between the various-sized orifices, as it will be seen that there are two constants which differ with each jet orifice, although in some cases these constants are the same in any one equation. For instance, 0.19 and $0.19 \times 2 = 0.38$, while in others the two constants in any one equation do not bear the same relation to one another. Then again, the shape of the orifice is of great importance, and in my experiments I found that the flow of petrol varied through two orifices, presumably the same, but in one case the hole was perhaps not truly drilled, and in small holes of this kind it is often a difficulty to find a truly circular hole.

To overcome this difficulty, more than one well-known carburettor designer has given up reliance upon a circular jet orifice, and the ideal would seem to be an annular slit, the width of which can be varied at will, and the proportion of the width through which free discharge can take place can be truly regulated by the volume of air passing to the engine. Whilst discussing variable jet carburettors, we naturally proceed to the consideration of types fitted with multiple jets, and in Professor Morgan's paper the curves shown for this type of carburettor are in no way surprising, though they are of extreme interest.

Firstly, from my experiments I found how very difficult it was to obtain consistent results from any jet orifice of a diameter less than 0.9 mm., and in multiple jet types it is always necessary, at any rate in carburettors for small engines, to have large single jet of very much less diameter than this. And again, it must be obvious that as each jet comes into play in turn there will be considerable

lag, due both to the inertia of liquid with its surface tension, and also to the friction of rest which had existed in the jet itself up to the moment of the orifice becoming uncovered. This effect is clearly shown in my curves, and is indicated by the hesitancy of the jet to commence discharging, shown in the sudden rise in the left hand corner of the curves. This is most marked in the vacuum tube instrument to which I have referred.

In order to allay any doubt which may exist in the mind of the reader as to the practical corroboration of these curves, which I carried out on Brooklands track and elsewhere, I may state that the volume of air which I calculated to enter the cylinders was carefully approximated both by means of an anemometer placed in the air inlet to the carburettor, and also by so arranging the condition and the size of the jets that I was running at the most efficient mixture at the times when the observations were made. I generally found that at the high speeds the percentage of cylinder full of mixture was 75 at 1,200 r.p.m. (reduced to 60% at 2,300 r.p.m. when using another engine). Of course, the volume of cylinder filled varies with the design of the engine, and some correct measurements for these values were given by Dr. Watson in his paper some time ago before the Institute of Automobile Engineers. My first trial engine must have been somewhat similar to one of his, as the figure of 75%, which I obtained, practically agrees with what he measured. In designing a carburettor system for any particular purpose this must be carefully borne in mind, as also must the question of the length of the stroke.

We cannot say that, because a certain piston speed appears as a maximum in any one instance, the speed of the gas through the carburettor orifice will be equal to the piston speed multiplied by the diameter of the cylinder squared and divided by the diameter of the carburettor orifice squared, although, of course, this is a desirable state of things. A six-cylinder engine presents another difficulty wherever over-lapping suctions occur, in instances where a single carburettor feeds the six cylinders and it is necessary to obtain some idea as to the combined curve of piston speeds, and to make due allowance for the lag of air flow before assuming a maximum air velocity through the carburettor. In many cases, however, we find that, where a pulsating air flow itself occurred, the pulsations are so rapid that they do not effect the petrol discharge, as the liquid itself has a very much greater inertia than has air.

It is my opinion that far too much stress is laid upon the possibility of pulsating flow taking place in a carburettor system, and this argument has on several occasions been raised against my deductions, but as against this allegation I think, if the volume of petrol in the jet and the pipe leading to it are considered—and also the additional volume in the float chamber which would have to be effected if any such pulsations did occur—this mass would be sufficient to nullify any such action by reason of its magnitude, no actual effect, therefore, taking place at the jet orifice. We may classify carburettors into the constant vacuum type and the variable. I have dealt with the former, and I think that I have proved from my curves that in the variable vacuum type the discharge of petrol related to the vacuum acting on it gives a straight line curve. The variable vacuum type can again be subdivided into a fixed jet orifice with variable air velocity; varying jet orifice and varying air velocity; fixed jet orifice, and partly varying air velocity, i.e., when additional air is admitted between the jet chamber and the engine. There is, as we know, a very wide difference of opinion as to which of these types is the best, but as an axiom we may state that the best is the simplest.

When well designed it is unnecessary to employ contraption types, which are far more liable to give grave and varying errors in their performances, some of which are graphically shown by Professor Morgan. I do not, however, agree with him in his deduction that a plain tube carburettor with a set dribble at the jet will attain the desired result, although, in a measure, it would tend to bring the characteristic curve into a straight line. I think the suggestion is too crude. However, any simple means can be arranged, so that the vacuum acting on a single orifice can be controlled proportionately to the air passing to the engine. What we have so long strived for in carburation would be then well advanced in the direction of simplicity and perfect results at all engine speeds.

FRENCH AUTOMOBILE DESIGN FOR 1911.

As exemplified by the chassis shown at the 1910 Exhibition in Paris.

THE similarity between present French and British practice with respect to pleasure chassis design is so considerable that if two shows are held in quick succession, one typical of one nationality, and one of the other, the second exhibition, whichever it may be, is bound to be much the less interesting. In the particular case of the London and Paris shows the disappointing nature of the second event is heightened greatly by the fact that so very many of the best French chassis are to be seen in London, and indeed, if this state of affairs between the nations was reciprocal, the shows would be almost identical. Thus speaking on broad lines, there was so little of complete novelty at the Paris Salon that it cannot be said that the wider knowledge of Continental work, which a visit to it gives, is likely to cause any alteration of opinion as to the general trend of design as shown at Olympia. In matters of detail the differences are quite noticeable, chiefly to the advantage of British manufacture, but the main lines of current practice, as discussed in these pages last month and dealt with more fully in the Annual of *The Automobile Engineer*, certainly need no qualification.

It would be hard to say whether the average of excellence of exhibited chassis was greater in London or in Paris, as so much depends upon the point of view, but if sound design alone is considered, it may be said at once that Great Britain is now leading her chief Continental rival for all classes of pleasure car work. The lead is not great, and it will need the assiduous attention of home producers to maintain it; but it is none the less real. The difference is not to be found so much in large matters as in small—in general convenience and care for detail is where it shows up most strongly—and there is no object to be gained by the citation of particular examples, rather on the other hand, is it more advantageous to study the cases where something is to be learnt.

Engine Design.

Amongst the engines at the Salon there were many which could only be described as experimental, to use no stronger word. Experience in the recent past has shown that it is entirely unsafe to predict the behaviour of any new construction, but some of the engines exhibited do not seem to possess much promise in that their advantages are far from obvious, while their drawbacks are plain. As regards engines of the ordinary type, the chief difference between French and British practice is in the matter of lubrication. Whereas in this country forced lubrication or trough-controlled splash systems are almost universal, there are still very many French made engines with the old-fashioned plain splash, a very favourite modification being to divide the crankcase into two parts by a transverse division, and to fit overflow passages by which oil returns to a small sump directly the level becomes too high. This design necessitates a pump, and cannot be cheaper than the trough system, while it is certainly less effective. On the other hand there are a number of excellently-designed complete forced systems, amongst which that of the De Dion (which is well known, and was described fully in the *The Automobile Engineer* for July, 1910) may be mentioned as one of the best. External pipe work, in place of passages cast in the crankcase, are more common in French work, and while they are less neat, they have the advantage of being cheaper and more easy to clean. Pumps when used are almost always of the gear wheel type, and are placed as inaccessibly as in the majority of British designs. If anything, the pumps used are slightly larger than is common here, but the difference is very slight. Filtering arrangements, level taps and drains, are all similar to British practice, and the Bourdon gauge is the most frequently employed method of indicating the approximate pressure of the oil, and its passage to the bearings.

The compression ratios would appear to be somewhat higher than are common here, especially with regard to the smaller engines, but none of the few pistons which were available for examination at the Salon were as light as the best to be found at Olympia. With the large stroke/bore ratio (about 1.4 for the new engines) which is usual in French as well as British practice, and the high speeds of revolution which are desired, the piston weight has a direct and powerful influence on smooth running, so it is reasonable to assume that hard running is a common fault of the engines under consideration.

For cylinders, the one-piece casting is almost universal on the small chassis, and fairly common for large engines, there being some elaborate examples of foundry work with all pipes and passages cast-in with the cylinders. The six-cylinder Delage, which was illustrated in our annual issue, is as complicated as any, perhaps, but it is comparatively small, and a more noticeable job was the new four-cylinder piston valve Berliet. This engine has a bore of 100 mm. and a stroke of 140 mm.; the valve pistons are driven at only a quarter the crankshaft speed, as they open the cylinder ports on both the up and the down strokes, while even the automatic air adding valve chamber is cast as a part of the block, the only external attachment being the float chamber and jet of the carburettor, with a short pipe feeding a very rich gas. Particulars as to the behaviour of this engine are as yet unobtainable, but it is said to have given every satisfaction to the makers.

For large engines, however, pair casting is far more common than one piece, though the pairs are not infrequently

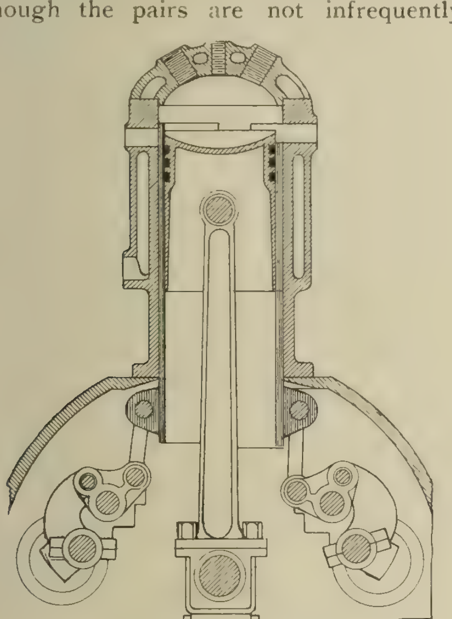


Fig. I.

bolted together, Maudslay fashion, and the Panhard Company make one chassis in which six separate cylinders are assembled thus with long bolts passing from end to end of the row. Otherwise singly-cast cylinders have almost disappeared, and pump-maintained water circulation seems to be giving way to thermo-syphon for cars of all sizes, though, judging by the enormous radiators which are necessary for big engines, it is likely that the pump will come back into favour, if indeed, the big engine does not disappear altogether as small engines are made more and more powerful.

Poppet valves generally are slightly smaller on French engines than on corresponding British motors, and there were many more enclosed valves at Olympia than in Paris, so it seems that British designers can teach rather than learn in this respect. The solid camshaft is common, and cams with concave flanks appear to be the rule rather than the exception, which looks as though more French attention was being given to power production than to silent running. For shafts the material used is usually of a high tensile strength, and many crankshafts are stated to be of nickel-chrome steel. Three bearings are usual for a four-throw shaft, both the two and five-bearing arrangements being equally rare, while brasses lined with white metal are the standard type of bush. Several French makers are trying ball bearing engines, and it is noticeable that great care is then taken to relieve the journals of all thrust, while the latter are always of great diameter except in the case of one or two quite small engines.

Altogether there is but little to be said concerning standard engines that has not already been dealt with very fully in these pages, but the few peculiar engines are worthy of a little more attention. Leaving the Knight engines of the Panhard out of the argument, it may be said at once that the sleeve valves shown were interesting but not convincing. The Rolland-Pilain bears the closest resemblance to the Knight, but has a single sleeve only, the ports being staggered to a sufficient extent to permit of their being uncovered in correct sequence. The valve diagram is probably not so good as that of the Knight, but to have a single sleeve, instead of two, is a very decided manufacturing and lubricating advantage, so the behaviour of the engine in ordinary use should be watched with interest. Barring the one sleeve, the engine is very similar to the Knight, having the separate water-cooled head, with a junk ring. An equally interesting engine, but one which would seem to possess greater possibilities for trouble, is the Mustad, shown diagrammatically in Fig. I. This has two half sleeves operated separately from two shafts, the action being made quite clear by the sketch, and the obvious weak point is the sliding junction of the two half sleeves, as it is reasonable to assume that as wear takes place leakage will occur past the piston down the junction on each side. It is easy to obtain a fairly good valve diagram with this form of construction, and tightness between the sleeves and the cylinder walls is, of course, maintained by internal pressure, and so it simply remains to be seen whether the wear at the sleeve junctions will be enough to cause leakage. This engine has not got the deep head of the Knight and so there is no special cooling of the upper ends of the sleeves, whence other trouble might arise.

Perhaps the most interesting engine in the whole show was the Cottereau, because it is known to have been in actual use for some time, and a car fitted with it has given a good account of itself. The construction is indicated by the diagram, Fig. II., but some explanation is needed to make it clear. Firstly, there are two ports in the cylinder, placed on the same level, and directly opposite each other. Inside the cylinder there is a rotating cylindrical valve, having a single port, which registers with the two cylinder ports in turn, and it is, of course, driven at half crankshaft speed. The valve itself is cast iron, and is horizontally of piston ring

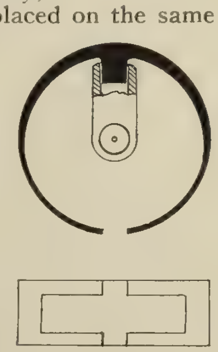


Fig. II.

section, that is to say, it is slotted, and from the slot tapers to the maximum wall thickness at the opposite centre, while the port is situated on the thin side, the vertical slit being central to it. Thus the ring or valve has sufficient spring to keep it always in close contact with the cylinder wall. To drive the valve there is a vertical shaft situated centrally in the cylinder head, which it enters through a variety of stuffing box to prevent leakage of the explosion, and this shaft has a sort of tee, or L piece at its lower end. The end of the tee piece is hollow, and fits quite loosely over a projection cast on the valve ring at its thickest part. Thus, as the valve is not connected rigidly to any other part, it is free to expand, and no great stresses can be imposed upon it. For lubrication, the driving shaft is hollow and oil is fed down through it, there being a ball valve at the lower end to prevent blow back. The design is certainly not of a kind to inspire confidence, and it may

Fig. III.

at first sight appear rather complicated, but the number of parts is not really very great, and they are mostly cheap enough to make, even the valve ring being a simple machining job. The friction must be considerable, and this assumption is borne out by the decidedly substantial nature of the driving gears, but troubles from carbon deposit are said to be non-existent, because the continual rubbing prevents any accumulation.

It may be remembered that in *The Automobile Engineer* for September last we published a section (taken from a patent specification) of an engine with a spherical cap valve inside the combustion chamber. This design has made its appearance in France on an aviation engine manufactured by Messrs. Ballot, the cap valves being driven from the top. There is, of course, no special need for gas-tight enclosure of the driving mechanism as the valve itself acts as a permanent seal to the opening. The particulars obtainable were extremely scanty, but we hope to be able to deal more fully with the engine in another issue. Lubrication in this case is performed from the top, as in the Cottereau, the cap being grooved to conduct the oil well over its surface.

An engine of a quite different type, but none the less peculiar, is the Henriod, which has only a single rotary valve, shown in the diagram, Fig. III. Commencing with the end of the exhaust stroke, the cycle is as follows:—The piston descends a sixth of the stroke before it uncovers the port, and the valve previously opens the inlet passage, closing again just as the bottom of the stroke is reached. Compression and firing take place while the back of the valve is towards the cylinder, thus there is a tendency to lift it from its seating within the limits of clearance, but this is said to be sufficiently counteracted by the low position of the cylinder wall port, the greatest pressure of explosion being when the valve

is entirely covered and protected by the piston. At the end of the firing stroke, auxiliary exhaust ports cut in the cylinder wall are opened for an instant (they are, of course, also uncovered on the suction stroke), and the valve has by then assumed the dotted position where the exhaust port is uncovered. For a four-cylinder engine a single casting is used, and the valves for each cylinder are on one shaft, being contained in a common chamber running down the whole length of the casting, while there is a single driving gear at the front end, and so the whole construction is simple and fairly cheap. The setting of the port so low that the piston can protect it is a serious disadvantage on the exhaust stroke (partially counteracted by the auxiliary ports), and is as great a drawback on the inlet stroke. It is, of course, not possible to criticise such a design as this without knowing the comparative power and economy with other engines, but it may be remarked that if the claimed speed of four thousand revolutions per minute has been obtained with a normal stroke/bore ratio, then there must be some feature of the engine that is well worth studying.

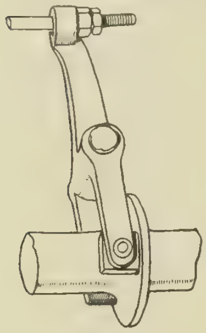


Fig. IV.

It thus becomes obvious either that the French manufacturers have been experimenting with new engine designs to a greater extent than have their British colleagues, or that they have been less cautious than the latter in publishing the results of experiment. It seems the last-named probability is the stronger, and it is perhaps significant that very few of the peculiar engines are standard for all their makers' chassis, they being mostly only made in one size.

Transmission.

At the British exhibition the changes and novelties in gearbox and clutch construction were so small that it was possible to say that scarcely any progress had been made in the past year. While this is in a sense true also of the Paris show, still some quite new designs were to be found there, both separately and on chassis. Clutches were almost all of the leather cone variety, pressed steel cones of the Panhard pattern being very common and, as in London, clutch stops were the rule. There is thus very little to be said concerning clutch design, the

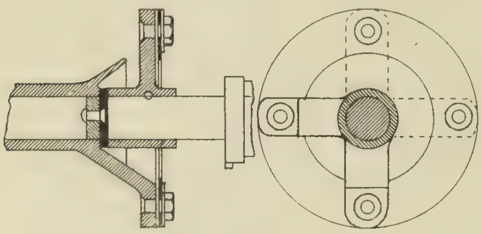


Fig. V.

chief point of interest being the method of coupling to the gearbox. In passing, however, it may be remarked that there was a neat clutch striking gear on the six-cylinder Darracq, in which the stop was formed by fitting fibre blocks on the ends of the striking pins. This is shown in Fig. IV., and it seems a very convenient way of combining two parts. The most interesting coupling was certainly that on the Th Schneider, shown in Fig. V. Here there is a ring of spring steel some 200 mm. in diameter that forms the link between the two shafts, each of which carries two arms at right angles, which are bolted to the ring (in Fig. V. one pair are shown full and the other pair dotted for the sake of clearness). This seems to be a good joint because it cannot possibly cause noise, and should be sufficiently universal to compensate for the extremely small variations in alignment likely to occur between the crankshaft and gear shaft. The commonest coupling is apparently the unsatisfactory type with a rounded "square" block on each shaft and a box connector, but the pin and ring is also frequent, as is the De Dion joint. On the Barré chassis there was an aluminium cover for the joint attached to the clutch, the striking fork being situated behind the coupling, and this is rather interesting because it is quite the opposite extreme to standard practice, which is to leave the joint exposed.

Certain of the French makers have never become supporters of the gate change-gear system, but in one notable case—that of the Turcat Mery—a return from the gate control to the straight-through arrangement has been made. Both on the chassis shown by the firm just mentioned and on the Chenard Walcker cars a very long gearbox is now being used with the smallest possible square section and exceptionally large castellated shafts. Particular silence is claimed for this construction because no shaft rigidity is lost, owing to the increased diameter, while the narrow box is less resonant than the more nearly cubical ordinary pattern. The Turcat Mery firm say that their gears will run with almost complete silence when exposed, and that the small sounds made are magnified enormously by the sounding-board effect of the thin aluminium casing. However, this may be, and although the straight-through design has manufacturing advantages, there is no doubt that the gate change is much more easy to handle and is much to be preferred for ordinary comparatively bad drivers. Accidental gear damage is much more likely to occur with a straight-through change, and if the secret of silence lies partly in the design

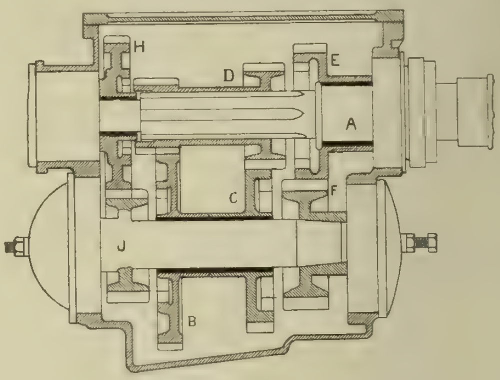


Fig. VI.

of the gearbox itself it can surely be found by altering the method of support on the section of the walls without so radical a retrogression.

For the great majority of French cars the plain gate with a sliding tube to carry the lever is employed, and all the remarks made regarding standard practice in our annual issue hold good for these. Turning to the entirely novel designs, there is a chain change speed gear now fitted to one of the Aries chassis, very much on the lines of that employed by the London General Omnibus Co. In this two of the three forward speeds are indirect, and the top is a through drive, all the chains running continuously, while the reverse is given by a supplementary spur gear train, which is normally out of action. Judging by the success of this transmission for omnibus work it should be entirely good for pleasure car work, and it is a little surprising that so old a French firm should be the first to make it standard on any chassis.

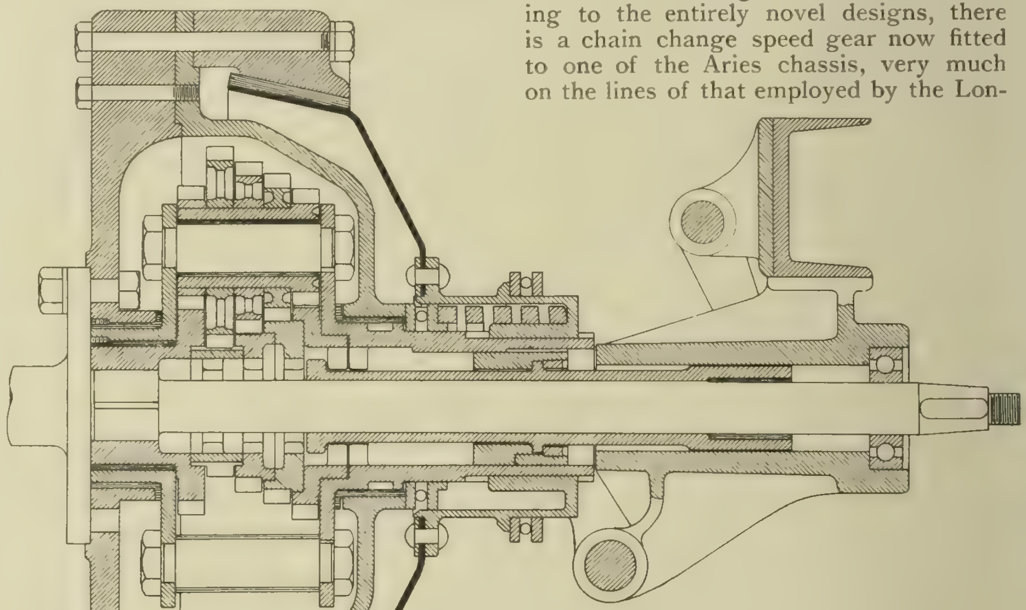


Fig. VIII.

Contrary to the Turcat Mery practice, a box found on several chassis is the design shown in Fig. VI. In this almost everything has been sacrificed to obtaining very short shafts and also to so dispose the parts that all the working surfaces of the gears are in constant mesh. Every engagement is by means of internally toothed rings, and the action is as follows:—For the first speed B is slid over J and the drive from the engine shaft A then passes in the sequence D, B, J, H, for the second speed C slides over F giving sequence A, D, C, J, H, for the third speed D slides into E giving E, F, J, H, and for the fourth speed D slides into

H, giving a through drive. For the reverse H and B are connected by a separate double pinion. In this design no wheel is ever idle, even in the neutral position. Another curious construction is the Henriod gear, which is of the epicyclic pattern, and is contained entirely within the flywheel or clutch. This is shown in Fig. VIII., and it will be seen that the engagement is by means of arresting the motion of the different sun wheels by means of a sort of sliding key clutch. The ordinary clutch can be used with any gear, and must be taken out for changing speed just as for a sliding gear. It will be seen that the clutch carries the planet pinions, and that any of the sun wheels can be held stationary by the sliding dog. On the outside of the stationary shaft carrying the dog there is another "key," itself running on a feather in the clutch sleeve, so when the stationary dog has been pressed right home the clutch becomes locked to the end sun wheel and then the whole gear revolves solidly. This gear is certainly neat, but it has the disadvantage of giving an extremely inefficient reverse; still it is less complicated than the ordinary type of epicyclic gear, and the speed changing is more easy than with sliding gears, though less easy than with brake bands. The stationary shaft would seem to be none too strong, but the actual dog is robust enough to withstand considerable rough usage.

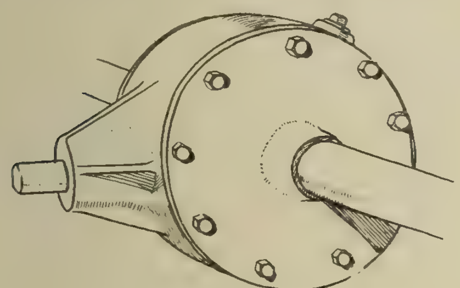


Fig. IX.

On one or two chassis there was yet another unusual gearbox, which appears to be an imitation of the Linley design fitted to all Commer cars and also to some Vauxhall chassis, the system consisting of constant mesh gears engaged by dog clutches with spring control, so that the spring is compressed by movement of the hand lever and the dog is moved by the spring only when the driving pressure is relieved by taking out the main clutch.

Axles and Universal Joints.

For front axles the forged H section is practically universal, tubular constructions being used only on quite a few small cars. In back axle design, however, much greater variations are to be noticed, and perhaps the most striking difference between British and French practice is the rarity of the pressed steel axle in France. Conical steel tubes with a cast central casing of aluminium or iron is the standard form, though the shape of the casing varies. A common type is that which follows the contour of the differential as closely as possible, and here it is obvious that the internal clearances are small, while the amount of lubricant contained cannot be great. Almost the opposite extreme is the type shown in Fig. IX., where the central piece is usually cast iron and the tubes and flanges steel. This design was used on quite a number of small cars, and should be cheap, while not being greatly inferior in strength to any other pattern, especially if a tie rod is used. One axle in the Salon was entirely of cast steel, the two halves being divided vertically at the centre, while the hub bearings were mounted on the sleeves. Torque rods of the triangulated pattern are most usual, and the torque tube with an enclosed propeller shaft was rather less noticeable than at Olympia, although present in fair quantities. One of the new Renault features was the aluminium cover enclosing the universal joint at the front end of the torque tube and also containing the brake compensation. This is shown in Fig. X., and it is necessary to explain that the brake shaft is free to swivel in its bearing on the frame side, its inner end being connected to a small differential gear exactly like that used on the Rolls-Royce. There is another brake-actuating rod on the opposite side of the propeller shaft and not shown in the sketch.

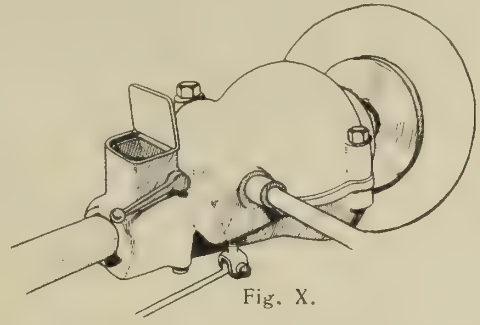


Fig. X.

Of universal joints, the ring pattern is the most common as here, but the De Dion joint is a close second, and practically all joints are now protected efficiently by pressed brass covers, the old leather wrapping having disappeared. On the Isotta Fraschini, however, the joint itself is leather, there being a kind of cup on the end of the gear shaft and a small disc on the propeller shaft connection between the two parts consisting of a simple thick disc of leather firmly gripped between flanges on the cup and the centre piece on the shaft. It seems likely that rough handling of the clutch would damage the leather, but the design is said only to have been adopted after extended trial, and there is no gear shaft brake, which relieves the joint of the more violent stressing. If this coupling is quite satisfactory it seems to be an ideal way of overcoming the noise-making tendencies of the all-metal joint, and ought to be cheap as well.

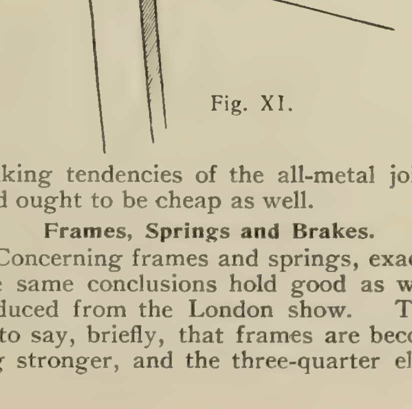


Fig. XI.

Frames, Springs and Brakes.

Concerning frames and springs, exactly the same conclusions hold good as were deduced from the London show. That is to say, briefly, that frames are becoming stronger, and the three-quarter elliptic spring has displaced the transverse and threatens to become standard practice. It is perhaps worth noting that on several chassis the top and bottom flanges of the pressed rear cross member were extended out beyond the frame sides so as to form a housing for the end of the quarter-elliptic portion of the spring. This is neat, and should be satisfactory where the pressing is of sufficiently heavy section, but in some of the instances noticed the gauge was decidedly on the light side and did not look strong enough. Practically the only peculiar spring mounting was the new Renault design, which has the spring pads above the springs so as to bring the frame below on the back axle, the sides being upswept under the rear end of the body to allow sufficient movement.

As regards steering gears also, no new departures were to be found, and the most interesting detail was a fitting on a Pierron (or Mass) chassis. Here there was a complete worm wheel to allow wear to be taken up by bringing a fresh part of the wheel into mesh with the worm, but instead of providing four keys, or even leaving it necessary to cut a new key-way for each adjustment as is sometimes done, the end of the worm wheel shaft was octagonal, as shown in Fig. XI.

Amongst brakes, the expanding type is universal for wheel hubs and the pair shoe external most usual for the propeller shaft, though an increasing number of makers are using expanding brakes at this point also. There were very few front wheel brakes at the show, but the hydraulically-operated device, shown in Fig. XII., was fitted to the sleeve valve, Rolland Pilain, and another rather interesting form was on the Isotta Fraschini. In this there was a cross shaft carried in brackets on the front axle, and at the ends were cams for spreading the brake shoes in the ordinary way, but instead of being flat-faced the cams were so rounded that they had point contact only with the shoes, thus allowing the wheels to be moved by the steering gear without affecting the brake shoe position, as the faces of the cams are curved in such a manner that their projection on a horizontal plane is a circle. It can, of course, only be accurate for one position, but perhaps the external compensation is sufficiently flexible to allow for small irregularities.

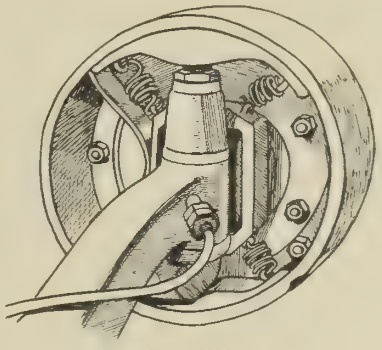


Fig. XII.

In conclusion, although a man who had not seen the Olympia Show could have learnt much by a visit to the Parisian exhibition, there is no doubt that much more was to be discovered in London, and those who made a study of current design here need not regret it if they have been unable to make a visit to the second show.

THE 12 H.P. ARGYLL CHASSIS.

A long-stroke engine with a two-bearing crankshaft and an interesting front wheel braking system.

THE aim of the designers of the 12 h.p. Argyll has admittedly been to copy the best features of the latest Continental practice with small improvements where possible, and taken as a whole their efforts have not been unsuccessful, although there are several debatable points. The engine design was published in our Annual last month, and so will not be reproduced here. It needs but little explanation; the cylinders have particularly ample water spaces surrounding them, and the valve pockets are equally well cooled—in fact, it is doubtful whether the cooling will not be found to be almost too complete, having regard for the 70 mm. cylinder bore. Concerning the two-bearing arrangement of the crankshaft and camshaft, it is not easy to express an opinion. Each shaft is large enough to be stiff under normal loading, and the crankshaft is of 60 ton steel, but it is probable that there is considerable whip when the load is heavy, and considerable stresses are set up in a camshaft when the valve springs are strong and the rate of revolution is high. Time alone can show whether the system is reliable, and whether the bearing durability is satisfactory, whence it will be interesting and instructive to see whether the Argyll Company continue the type in their 1912 designs. The object of the two-bearing crank is presumably to cheapen cost of production, because the length of engine saved is extremely small—in fact, a small sacrifice of width of water space between the front and rear pair of cylinders, leaving a slightly larger central space, would enable a three-bearing shaft to be employed without any increase in overall dimensions. It is rather curious, too, that it should have been considered worth while to put a cap bearing on the front end of the camshaft.

Lubrication is performed by an oil pump carried in the upper half of the crankcase, and driven by a skew gear situated between the two rearmost cylinders, where it is in an extremely accessible position. The pump forces oil under small pressure to the main crankshaft bearings, and also supplies a series of sprays, which maintain the level in the troughs, and feed the camshaft and cross-shaft driving gears. A rather unusual point in connection with the lubrication is the duplication of the oil holes in the small end, there being two in the upper portion, spaced wide apart, and two drilled up from beneath at the centre of the pin.

The reciprocating parts are reasonably light in weight, though the pistons are 90 mm. long, and the engine is claimed to be able to accelerate to 2,500 r.p.m. under load, while we are informed that it gives 20 b.h.p. at 1,800 r.p.m. In actual practice the engine runs very smoothly and quietly at low speeds of revolution, or when only lightly loaded, but makes its presence felt somewhat when accelerated on either gear. The acceleration is satisfactory in itself, and when running dead light the silence of operation is unusual, particularly as regards the smaller engine noises—in fact, it is very difficult to tell whether the engine is running even when seated in the stationary car.

Natural, or thermo syphon, circulation is used for the cooling water, and it will be noticed on the chassis elevation that there is a good head of water in the radiator above the outlet pipe, which leads from the combustion heads—a point often overlooked when the water pump is first dispensed with, and the radiator is of the increasingly popular, flat, vertical tube type. Plenty of space is given between the back of the radiator and the cylinders, so as to accommodate the magneto comfortably, and the near side end of the cross shaft is arranged with a cap, so that a commutator for accumulator ignition can be fitted if so desired. The spring-lifted cap which carries the fan is a very neat fitting, and worthy of more than passing notice. The inlet passages are cast-in, and take the usual double-branched form, the carburettor being attached on the valve side and low down. A separate ribbed exhaust pipe is attached by four separate flanges, and the valves are not covered by either the inlet or exhaust pipes, though they are enclosed by the covers, which can be seen in the elevation of the chassis.

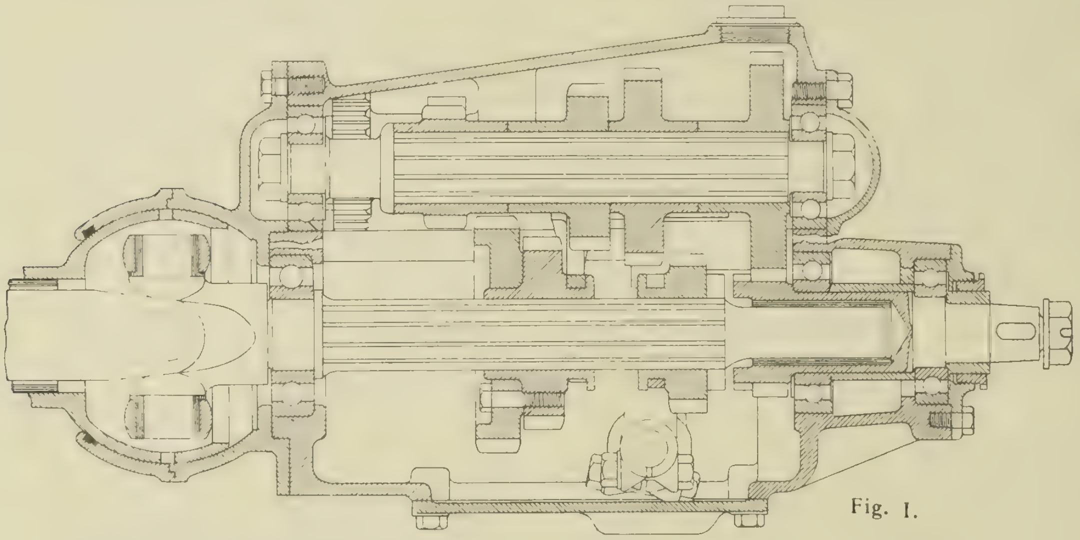


Fig. I.

Both the engine and gear box are mounted on an underframe, consisting of two pressings raised at each end and attached to the front and middle cross members of the main frame, an extra stiffening cross member being situated just in front of the flywheel. The clutch (Fig. II.) is of an ordinary disc pattern of good size carried on a crankshaft spigot, and is connected to the gearshaft by a square block and box coupling. A distance piece is fitted to prevent any longitudinal movement of the female portion, and this ought to counteract the tendency to rattle, which is often noticeable with semi-universal joints of this character.

The gearbox is shown in Fig. I., which is a section in the central vertical plane. Spined shafts are used for both the fixed and sliding gears, and the bearings should be sufficient, though they are not very large, the diameter of the layshaft pair being 1in. internal, and of the main shaft $1\frac{1}{2}$ins. for the extremes and $1\frac{5}{8}$ins. for the centre bearing. Seven diametral pitch is used for all the gears, which have the following ratios: First speed 3.8 to 1, second speed 2.1 to 1, and third speed direct. All the bearings are provided with protective washers to prevent the entry of large metallic fragments, and the only uncapped bearing has an adjustable oil-excluding washer. In each case the outer ball race is housed in the aluminium of the main casting, where it is free longitudinally, and this should be satisfactory practice with the comparatively light loads on the shafts, though there are some advantages in fitting ball races in the brass caps, owing to the harder nature of the material and its smaller liability to deform under alternating stresses. A simple striking gear is the result of the adoption of what may best be described as a Daimler type of gate, as this does away with any kind of locking gear within the box, the only internal fitting being the actual striking arms. A very neat casing encloses the whole of the gate, the reverse being obtained by coming through the first

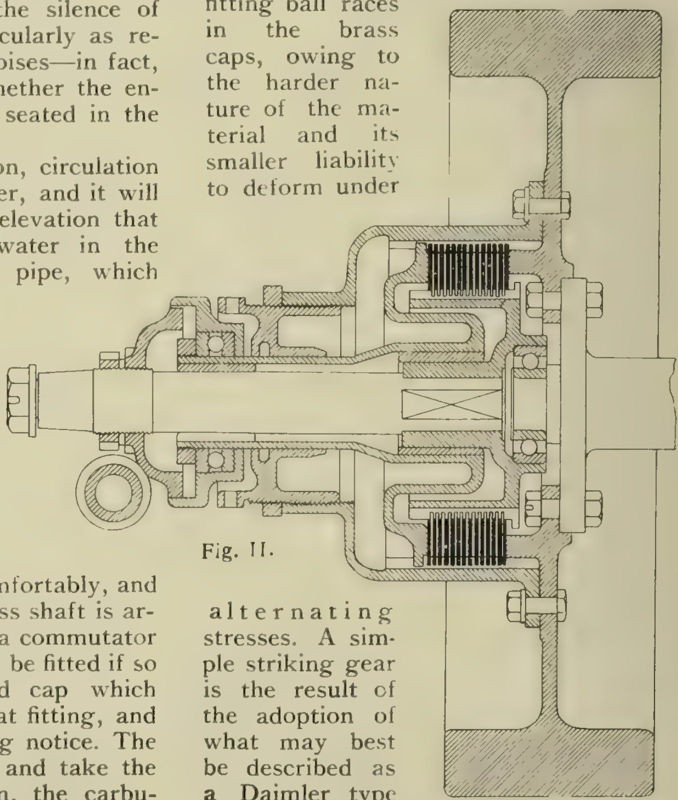


Fig. II.

speed, as is also shown in Fig. I.

Malleable cast-iron forms the rear end of the gearbox, and the outside part of the spherical torque tube end, which part is decidedly neat and well proportioned. This type of tube end is gaining in favour, and seems likely to become almost as common as the forked end, which is now the most usual design. The spherical end is the better of the two in that it encloses the universal joint, and makes the lubrication of the latter from the gearbox both reliable and effective. The torque tube is fixed rigidly to the axle case by being socketed in the malleable casting, and the propeller shaft has no bearing in the tube, it being supported at each end of the worm in the customary manner by a pair of $1\frac{1}{4}$in. ball journal bearings and a double-thrust ball bearing with half-inch balls and a ball centres diameter of $1\frac{3}{4}$in.

Fig. III. shows all the principal details of the axle, the only one requiring explanation being the small thrust ring between the ends of the two driving shafts, though it is easy to see on reflection that this takes any thrust from the road wheels, and transmits it to one or other of the large thrust rings which flank the worm wheel. A rather uncommon detail also is the fitting of oil trapping washers to prevent lubricant running into the axle sleeves, and this is presumably meant to assist the maintenance of a high-working level of oil in the differential case without the necessity for filling up the sleeves. Owing to the top position of the worm, it has been considered desirable to set the underframe carrying the engine and gearbox at an angle of $2\frac{1}{2}$ degrees to the horizontal, and the engine is thus tilted downwards in front. Although the appearance of the chassis is thereby rendered peculiar, there is no

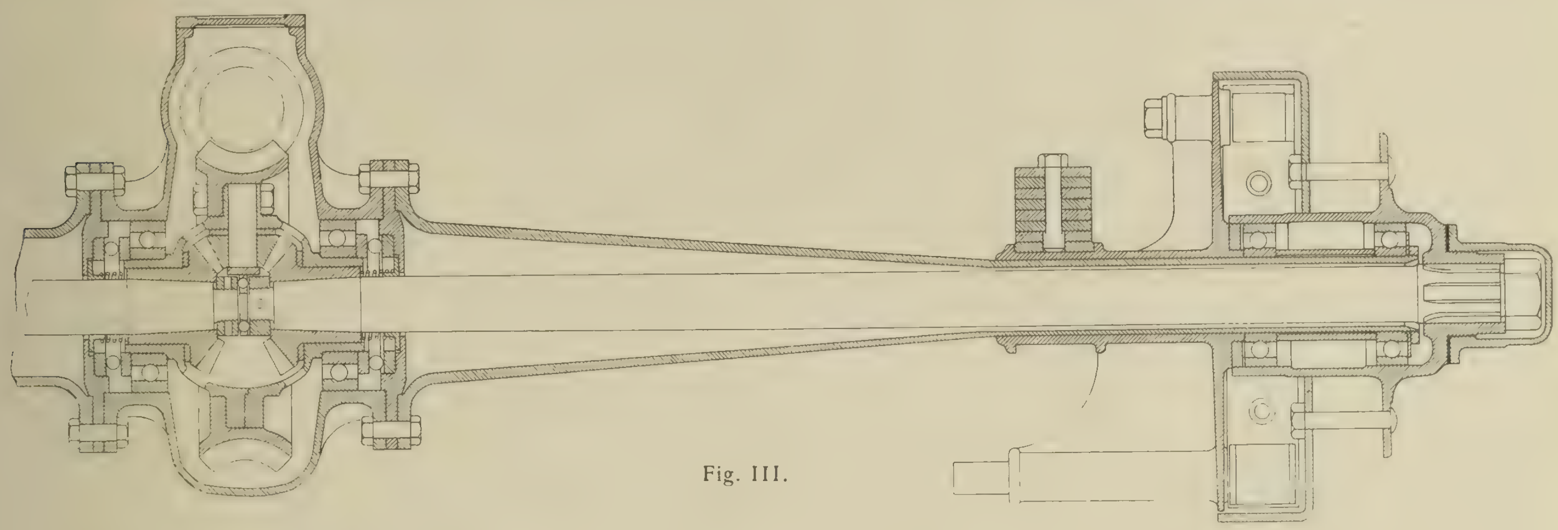
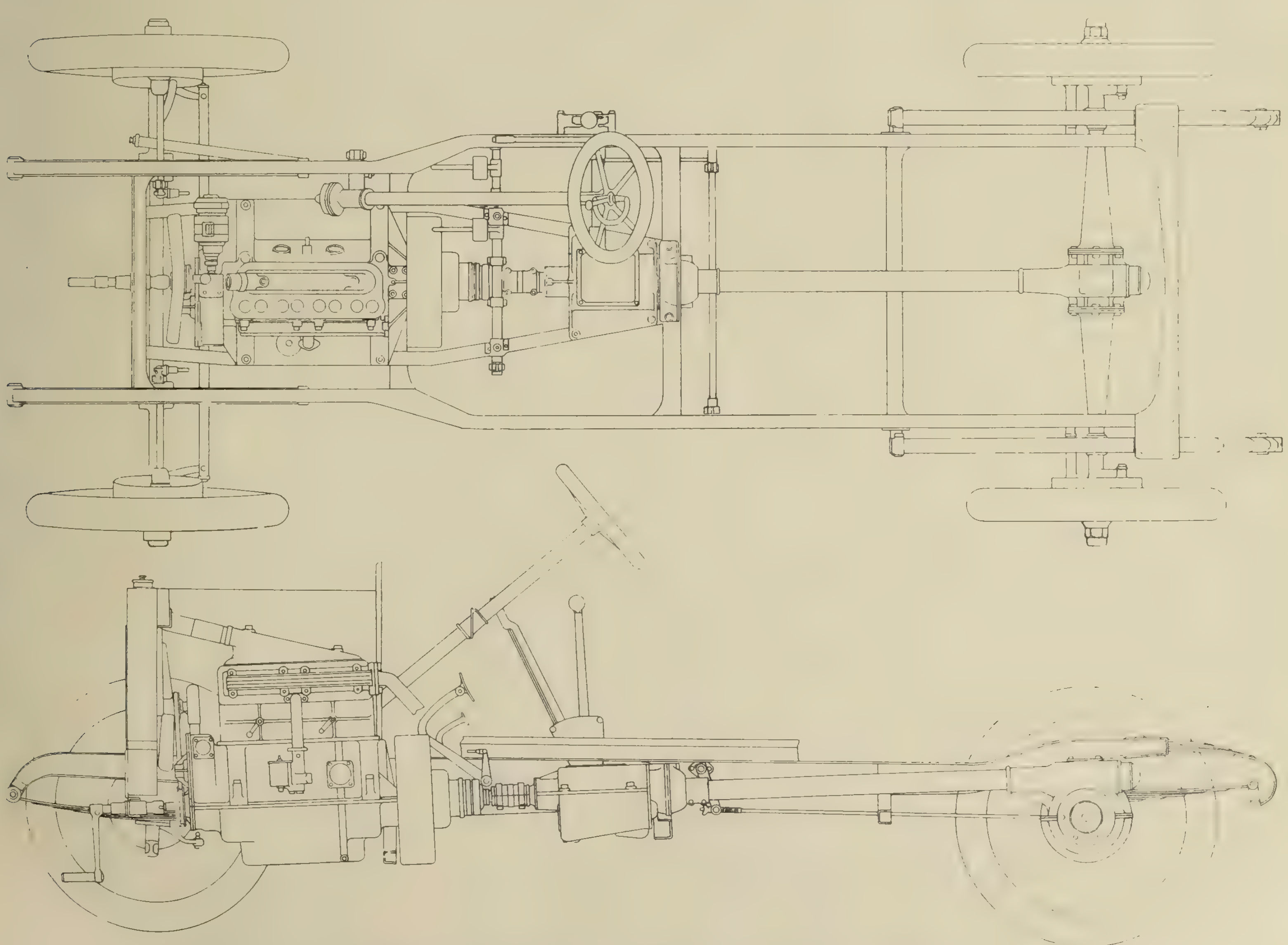


Fig. III.



General Arrangement of 12 h.p. Argyll Chassis.

doubt that the arrangement is a good one, and far better than raising the whole mechanism so as to get a straight line level drive. The extra clearance given by placing the worm above the worm wheel is often useful, and the suggestion that the lubrication is less reliable than in the case of a worm in the under position is not to be regarded very seriously.

The frame is all pressed steel, there being no tubular parts, and the only noticeable point is the construction of the rearmost cross member, so as to form the spring end boxes for the rear suspension. As regards the braking arrangements, the rear wheel brakes are shown in Fig. III., and it is only necessary to add that the hand lever is mounted on a bearing external to the gear striking shaft, and is therefore inside the gear changing lever. The front wheel brakes shown in Fig. IV. are rather more substantial than the majority of their kind, particularly as regards the control shafts and links. From Fig. IV. it will be observed that the steering pivots are canted, so as to intersect the point of contact of the tyre with the road when produced, and the cam which spreads the brake shoes ends in a universal joint of the slotted ball, or Rover, type, the centre also coinciding with the produced centre line of the pivot. The other end of the actuating shaft passes through a ball, which is free to swivel in any direction in a socket fixed to the frame, and the shaft is free to slide in this ball. Thus no perceptible motion of the brake shoes takes place owing to the action of steering or to the flection of the springs.

It is perhaps to be regretted that the hubs are not designed to take a detachable wheel—in fact, a completely new hub shell would have to be made for this purpose, but it is perhaps not fair to criticise a particular design on this score, as the fault is so general, and the detachable wheel has not yet become by any means as common as the fixed artillery type. The arrangement of the pivots, of course, makes it necessary that there the tie rod joints should be capable of universal movement, and this is advantageous, because the ball joint is more easy to adjust and lubricate than a pin joint. Adjustable thrust bearings are used in the worm box, and all the steering connections are straight, a good point being that the steering arm on the offside stub axle is not one piece with the tie rod arm, but is an entirely separate piece, and is attached separately to the stub axle.

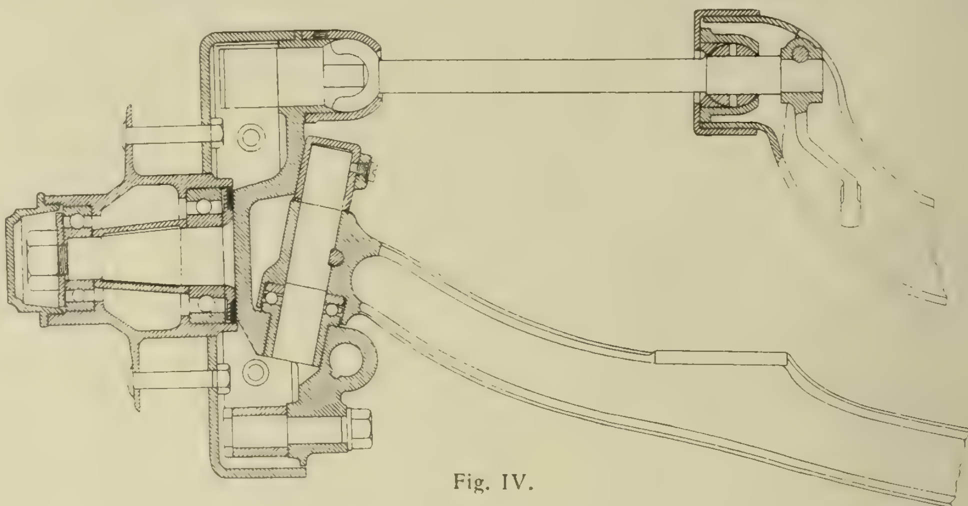


Fig. IV.

We have already made mention of the behaviour of the engine on the road, and need only add that it appears to give fully the horse power which it is claimed to develop on the bench. As regards the transmission, this is naturally quite silent on the direct speed, but is not noticeably so on the first and second gears. The springing is very easy, and the car appears to hold the load well at reasonable speeds. Taken as a whole the chassis should be particularly suited to town work, and would certainly be improved for high speed work on country roads by an increase in the gear ratio, which is a small fraction under four to one on the worm alone. It might be well to fit alternative ratios on this chassis.

AN INVESTIGATION OF THE THERMAL EFFICIENCY OF A TWO-CYCLE PETROL ENGINE.

Being a paper read before the Institution of Automobile Engineers by W. Watson, A.R.C.S., D.Sc., F.R.S., and R. W. Fenning, B.Sc. (Eng.).

A FAIRLY complete series of tests on the efficiency of a typical four-cycle petrol engine having been carried through, the results of which were communicated to the Institution in a previous paper,* it seemed that it would be of considerable interest to carry out a similar series of experiments with a two-cycle engine, a type which, though hardly employed at all on motor cars in this country, is yet very largely used on boats, particularly in America.

The engine tested was a single-cylinder Day engine intended for marine use, rated by the makers at 2.5 h.p., at 900 revolutions per minute, the cylinders having a bore and stroke of $3\frac{1}{4}$ins. From the sectional elevation of the engine (Fig. I.) it will be seen that crank-case compression is employed, and that the engine is of the three-port type—i.e., the admission of the fresh charge to the crank-case is governed by the piston, the lower edge of which uncovers a port when near the top of the stroke. The upper edge of the piston on the downward, or working, stroke first uncovers the exhaust port, and then the inlet port, which is in communication with the crank-case.

The valve diagram (Fig. IIA) gives the position of the crank at the opening and closing of each of the three ports, and enables a determination to be made of the time during which each port remains open.

A deflector plate cast on the top of the piston is intended to assist in preventing the incoming charge from passing direct from the admission to the exhaust port, and so escaping from the cylinder with the exhaust products.

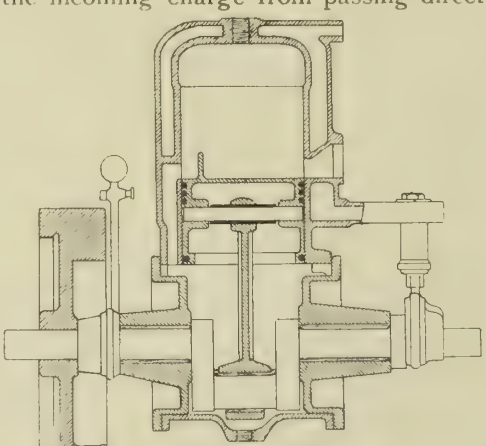


Fig. I.

The ignition is by means of a high-tension trembler coil and battery, the timing being adjustable. It is of interest to note that the maximum spark advance which could be obtained as the engine was sent out by the makers was as shown in Fig. III., where the speed is 900 revolutions per minute. By filing away the stop which limited the travel of the ignition lever, so as to admit of much more advance, the diagram took the form shown by the dotted line in Fig. III.; an increase of power of 15.6 per cent. being thus obtained at this speed, while at higher speeds the increase was even more marked.

The carburettor is of the single jet, float-feed type, fitted with a throttle valve and a hand-operated extra air valve (Brown and Barlow bicycle type). During the tests the throttle and extra air valves were always kept full open (except in a few cases), so as to reduce the throttling of the incoming air to a minimum. The strength of mixture was adjusted by means of a needle concentric with the jet, and by means of which the effective area of the jet could be adjusted. Both air inlets to the carburettor were connected by a pipe to a large box, used in measuring the amount of air taken by the engine.

The exhaust pipe is $1\frac{1}{4}$in. internal diameter, and about 12 feet in length, and is provided with a silencer at a distance of 22in. from the engine. A small pipe, used when taking the exhaust gas samples, is connected to the exhaust pipe between the exhaust port and the silencer.

The cylinder is water cooled, the circulation being maintained by a plunger type

*Proceedings Inst. A. E., vol. 3, p. 387. 1908-9.

pump which is driven by the engine.

The lubrication is by means of grease cups to the main bearings, and by drip to the inlet pipe, the oil being swept into the crank-case by the fresh charge as it enters.

Method of Conducting Tests.

A dynamo, driven by means of a belt, was used to absorb the power; it was separately excited, and the current gener-

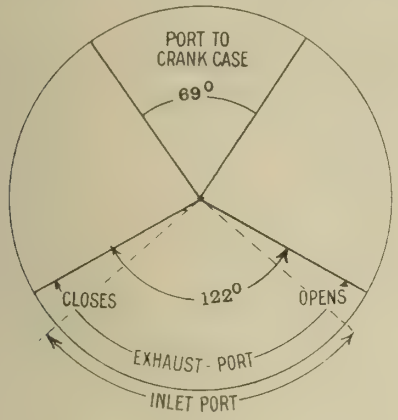


Fig. IIa.

ated was led through adjustable resistances, by means of which the output of the machine could be regulated and the speed of the engine controlled. Variations in the speed during a test were detected by means of a Weston electro-magnetic speed indicator, but the values for the speed given for the different tests were obtained by means of a revolution counter and a stop watch, both being started (and stopped) at the commencement (and end) of each test. No direct measurements of b.h.p. were attempted, since it would have been difficult to allow for losses in the belt, and for those due to the strain put on the bearings owing to the large amount of belt tension found necessary to prevent slipping. The i.h.p. was obtained by means of the diaphragm optical indicator described in the paper on the four-cycle engine. In general four diagrams were taken (by photography) during each test, and the mean effective pres-

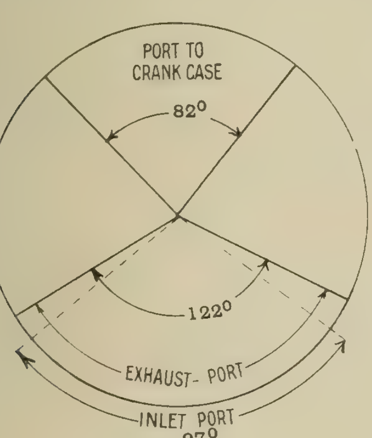


Fig. IIb.

sure was deduced by calibrating the indicator and determining the area of the diagrams by means of a planimeter.

The arrangements for measuring the petrol consumption were similar to those employed in the previous tests, i.e., provision was made so that the engine could be fed either from a tank or a graduated vessel, a three-way cock being used to change from one to the other. Each test occupied the time during which 100 cc. of petrol were consumed, the average duration being about four minutes. The temperature of the petrol was measured, and throughout the tests the same sample was employed, the calorific value being 18,600 B. Th. U. per pound.

The weight of air supplied to the engine was determined by measuring the drop of pressure between the two sides of a circular aperture in a thin metal plate forming the inlet to a box from which the engine drew its air. The volume of the box was about 22 cub. ft., and it was provided with an indiarubber diaphragm to assist in reducing the pressure fluctuations in the box. The difference in pressure was measured by a water gauge graduated in hundredths of an inch, and the size of the orifice was so chosen that the pressure difference was between 1in. and 1½in.

The exhaust gases were collected in glass sampling vessels, having a capacity of 300 cc. The tube from the exhaust pipe was fitted with a three-way glass cock, one branch leading to a non-return valve, and the other to the top of the sampling vessel. During the preliminary adjustments before a test, the exhaust gases were allowed to blow through the non-return valve, and thus cleared out all the air from the connecting tube; the sampling vessel was also filled with mercury. When the test commenced, the cock was so turned that one end of the sampling vessel was connected with the exhaust pipe, and the mercury was allowed to flow away slowly from the other end, so that the collection of the sample continued throughout the time the test lasted. In this way an average sample of the exhaust gases, uncontaminated by atmospheric air, was obtained. The method of performing the analysis was the same as that described in the previous paper; the carbon dioxide, free oxygen, and carbon monoxide being directly determined, while the hydrogen and marsh gas were calculated by means of Mr. H. Ballantyne's relation,* namely: percentage of hydrogen is 0.36, and percentage of marsh gas 0.12, of that of the carbon monoxide.

The trials made range themselves under four speeds, namely, 600, 900, 1,200, and 1,500 revolutions per minute, the speed being kept as nearly as possible constant at one of these values during any one test.

To study the way in which the charge supply to the cylinder varies at different speeds, indicator diagrams were taken from the crank-case, in which the charge is compressed before being delivered to the working cylinder. Diagrams were also obtained from the working cylinder at the various speeds, using a thin diaphragm in the indicator, so as to obtain an open scale. The crank-case diagrams were averaged in the ordinary way by means of a planimeter, and the power consumed in doing the pumping work was calculated. Specimen diagrams taken from the crank-case are shown in Figs. V., VI., VII., and VIII.

From the construction of the engine it is evident that there is every probability of some of the incoming charge escaping through the exhaust port with the exhaust products. Since any of the fresh charge which escapes in this way will carry with it a certain amount of petrol vapour, such a loss is a distinct disadvantage, and it is important to determine its amount at various speeds.

In the case of this engine, when the mixture was richer than one of petrol to fourteen of air (by weight) the exhaust gas contained both carbon monoxide and free oxygen. Now in the experiments on the four-cycle engine it was shown that whenever there was carbon monoxide in the exhaust there was no free oxygen. In the four-cycle engine the exhaust and inlet valves were never open at the same time, and hence it was impossible for the fresh charge to find its way into the exhaust, unless one of the cylinders was not

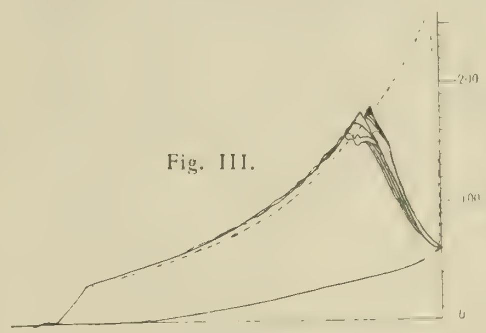


Fig. III.

firing. The explanation of the presence of the oxygen in the case of the two-cycle engine evidently is that some of the charge has escaped into the exhaust pipe without combustion taking place in the cylinder, and from the proportion of oxygen present we can determine by calculation the fraction of the charge supplied through the carburettor, which escapes combustion in the cylinder, and—as far as the engine is concerned—is lost.

To give an example:—

At a speed of 1,218 revolutions per minute, the analysis of the exhaust gases gave the following results, by volume:—

CO_2 =	7.4	
O_2 =	4.2	
CO =	5.6	
H_2 =	2.0	Calculated from % of CO.
CH_4 =	0.7	
N_2 =	80.1	By difference.
	100.0	

Now, in atmospheric air, 0.266 volumes of oxygen are mixed with one volume of

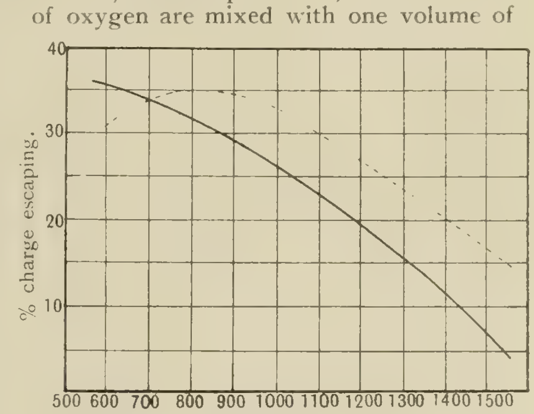


Fig. IV.

nitrogen, and therefore with the 80.1 volumes of nitrogen there must originally have been 80.1×0.266, or 21.3 volumes of oxygen. Of this 21.3 volumes of oxygen entering the engine, 21.3—4.2, i.e., 17.1, volumes took part in the combustion, and the remaining 4.2 volumes escaped through the exhaust port. Since the oxygen in the incoming charge bears the same ratio to the oxygen which escapes as does the total volume of the incoming charge to the total volume of the charge which escapes, we get:—

$$\frac{\text{Volume of escaped charge,}}{\text{Total volume of charge}} = \frac{4.2}{21.3} = .197.$$

i.e., 19.7% of the charge escapes unburnt

*Proceedings Inst. A. E., vol. 2, p. 101. 1908-9.

at this speed. To obtain satisfactory results for the percentage of escaping charge by this method, it is necessary to use a fairly rich mixture, so that there is at least one per cent. of carbon monoxide in the exhaust gases.

As a test of the accuracy of the deductions which can be made by the above method, we may calculate what would be the composition of the exhaust gases supposing we were able to eliminate the portion of the gases which escaped combustion. Returning to the example considered above, 4.2 volumes of oxygen must have been accompanied by 4.2/0.266, i.e., 15.8 volumes of nitrogen, and hence the composition of the exhaust if no unburnt charge were included would be

CO_2 =	7.4 volumes or	9.3 per cent.
CO =	5.6 ,,	7.0 ,,
H_2 =	2.0 ,,	2.5 ,,
CH_4 =	0.7 ,,	0.9 ,,
N_2 =	64.3 ,,	80.3 ,,
	80.0	100.0

Now in the above experiment the air/petrol ratio was 10.96, and from the test on the four-cycle engine the composition of the exhaust gases with this ratio was

CO_2 =	9.5
CO =	7.5
H_2 =	2.7
CH_4 =	0.9
N_2 =	79.4
	100.0

The agreement between these two results is a confirmation of the correctness of the deductions we have made with reference to the proportion of the charge which escapes in the case of the two-cycle engine. Although the agreement is not always quite as good as in the above example, yet results agree generally to within about half of 1 per cent.

The manner in which the percentage of charge escaping varies with the speed is shown by the continuous curve in Fig. IV., and varies from 35 per cent. at 600 revs. per min. down to 7 per cent. at 1,500 revs. per min.

Results of the Tests.

The results of the tests made at the four speeds, with the engine valve timing as adjusted by the makes, are summarised in Table IA.

In addition to the table given above, a few examples of indicator diagrams are given in Figs. V., VI., VII., and VIII. For each speed there are given two typical diagrams from the cylinder taken with a strong indicator spring. The top diagram is for an air/petrol ratio of about 14; i.e., one for which there is no carbon monoxide in the exhaust, and the second diagram for an air/petrol ratio of about 11. The third diagram is obtained from the crank-case, and the bottom diagram is a weak-spring diagram, by means of which the changes of pressure during exhaust and induction can be studied. The vertical lines on the diagrams represent the points of the stroke at which the different ports are uncovered.

TABLE I.—(A. TESTS).

Test Number.	Speed R.P.M.	I.H.P.	B.H.P.	Lb. of Petrol per 1,000 revs.	Lb. of Petrol per I.H.P. hour.	Thermal Efficiency, Gross.	Thermal Efficiency, Net.	M.E.P. lb. per sq. in.	Air / Petrol by Weight.	Amount of Charge which escapes, per cent.
1	636	2·71	2·1	·0651	·917	·149	·230	62·6	11·24	35
2	596	2·51	1·9	·0660	·941	·145	226	·61·8	11·48	36
3	638	2·73	2·1	·0592	·830	·165	·252	62·9	12·45	34
4	609	2·48	1·9	·0531	·784	·174	268	59·7	13·88	–
5	601	2·35	1·7	·0493	·756	181	·279	57·5	15·12	—
6	604	2·36	1·7	·0493	·758	·180	·278	57·3	15·18	—
7	897	3·51	2·8	·0650	·997	·137		57·5	9·86	
8	903	3·60	2·8	·0612	·923	·148	·212	58·5	10·35	30
9	902	3·59	2·8	·0579	873	·157	·217	58·5	10·91	28
10	898	3·55	2·8	·0579	·881	155		58·0	11·05	—
11	938	3·68	2·9	·0520	·795	·172	·247	57·7	11·76	30
12	905	3·54	2·8	·0541	·831	·165	—	57·4	11·83	
13	900	3·57	2·8	·0493	·746	·183	—	58·2	13·02	—
14	937	3·55	2·8	·0452	·716	·191	·270	55·6	13·57	—
15	907	3·30	2·5	·0425	·702	·195	·276	53·4	14·86	
16	897	3·26	2·5	·0426	·703	·194	—	53·4	15·31	
17	1209	4·12	3·1	·0564	·992	·138	·169	50·1	9·75	18
18	1206	4·23	3·3	·0561	·960	·143	·177	51·6	9·91	19
19	1199	4·34	3·4	·0567	·945	·145	·175	52·8	9·92	17
20	1218	4·29	3·3	·0498	·847	·161	·201	51·8	10·96	20
21	1212	4·29	3·3	·0396	·671	·204	·250	52·0	13·87	
22	1210	3·77	2·8	·0333	·641	·213	·262	45·8	16·74	—
23	1218	3·79	2·8	·0331	·637	·215	·264	45·7	16·89	—
24	1504	4·85	3·6	·0449	·838	·163	·179	47·2	10·06	9
25	1514	4·71	3·4	·0428	·825	·166	·177	45·7	10·43	7
26	1510	4·75	3·5	·0424	·808	·169	—	46·2	10·68	—
27	1509	4·80	3·5	·0398	·751	·182		46·7	11·29	—
28	1502	4·94	3·7	·0392	·716	·191	·206	48·3	11·61	7
29	1500	4·80	3·5	·0375	·703	·195	·208	47·0	12·12	6
30	1509	4·77	3·5	·0369	·700	·195	—	46·5	12·18	—
31	1510	4·82	3·5	·0342	·643	·213	—	46·9	13·10	—
32	1510	4·82	3·5	·0327	·614	·223	—	47·0	13·71	—
33	1508	4·91	3·6	·0325	·598	·229	·247	47·9	13·98	—
34	1508	4·51	3·2	·0304	·610	·224	—	43·9	14·75	—
35	1501	4·52	3·3	·0307	·610	·224		44·3	14·85	—
Throttle Valve partly closed.										
36	639	2·25	—	·0486	·828	·165	·216	51·7	12·12	25
37	897	2·86	—	·0415	·781	·175	·187	46·9	11·27	6
38	896	2·89	—	·0416	·775	·177	·199	47·4	11·36	11
39	1208	3·55	-	·0339	·693	·197	·213	43·1	11·73	7
40	1510	3·65	-	·0269	·669	·204	·209	35·5	12·27	2

The following points may be noted with reference to these diagrams :—

Speed 600 revs. per minute, Fig. V. At this speed the pressure in the cylinder falls to atmospheric considerably before the end of the stroke, while the compression line starts at atmospheric pressure. In the diagram taken from the crank-case it will be seen that as the piston rises the pressure gradually falls to about 4 lbs. per square inch below the atmospheric line, when the third port is uncovered by the lower edge of the piston. The pressure then rises owing to the influx of the fresh charge, but does not reach the atmospheric line till the piston has completed about one-fifth of its downward stroke. By the time the inlet port is uncovered the pressure in the crank case has risen to about 6 lbs. per square inch, and falls nearly to the atmospheric line by the time the piston reaches the bottom of its stroke.

Speed 900 revs. per minute, Fig. VI. The pressure in the cylinder sinks to atmospheric some time before the end of the stroke, and owing to inertia of the gases in the exhaust pipe, falls a little below the atmospheric line. This effect, which is particularly noticeable at this speed, is probably due to a resonance effect depending on the particular length of exhaust pipe employed, and shows itself in some of the other measurements made at this speed. The chief peculiarity noticeable in the crank-case diagram is due to the fact that when the inlet port is uncovered the pressure in the cylinder is higher than that in the crank-case, and hence some of the exhaust blows back into the crank-case, causing a hump on the diagram.

Speed 1,200 revs. per minute, Fig. VII. In the crank-case there is not time, during the interval when communication with the carburettor is open, for sufficient fresh charge to enter and raise the pressure to atmospheric. In fact, the pressure does not reach atmospheric till the piston has completed half the downward stroke.

Speed 1,500 revs. per minute, Fig. VIII. The effects noted above are here slightly more accentuated; for example, the small amount by which the pressure in the crank-case rises due to inflow of fresh charge during the time the third port is open, and the rise in pressure due to blow back from the cylinder.

Discussion of Results.

The results given in the preceding table are best examined by means of the series of curves which follow. First, we have in Fig. IX. the relation between the volume of air drawn in through the carburettor and the volume swept out by the piston. The upper full-line curve gives the ratio of the total volume of air taken in by the engine to the stroke volume, and it will be observed that there is a fairly rapid decrease as the speed increases, so that at 1,500 revs. per minute the air taken is only 38 per cent. of the stroke volume. If in place of considering the total volume of charge taken into the engine we only take account of the volume of charge which remains in the cylinder and takes part in the combustion, i.e., if we deduct from the total air the air which escapes, we get the lower full-line curve given in Fig. IX. From this curve it will be seen that the volume of charge which remains

INDICATOR DIAGRAMS OBTAINED WITH TWO-STROKE ENGINE.

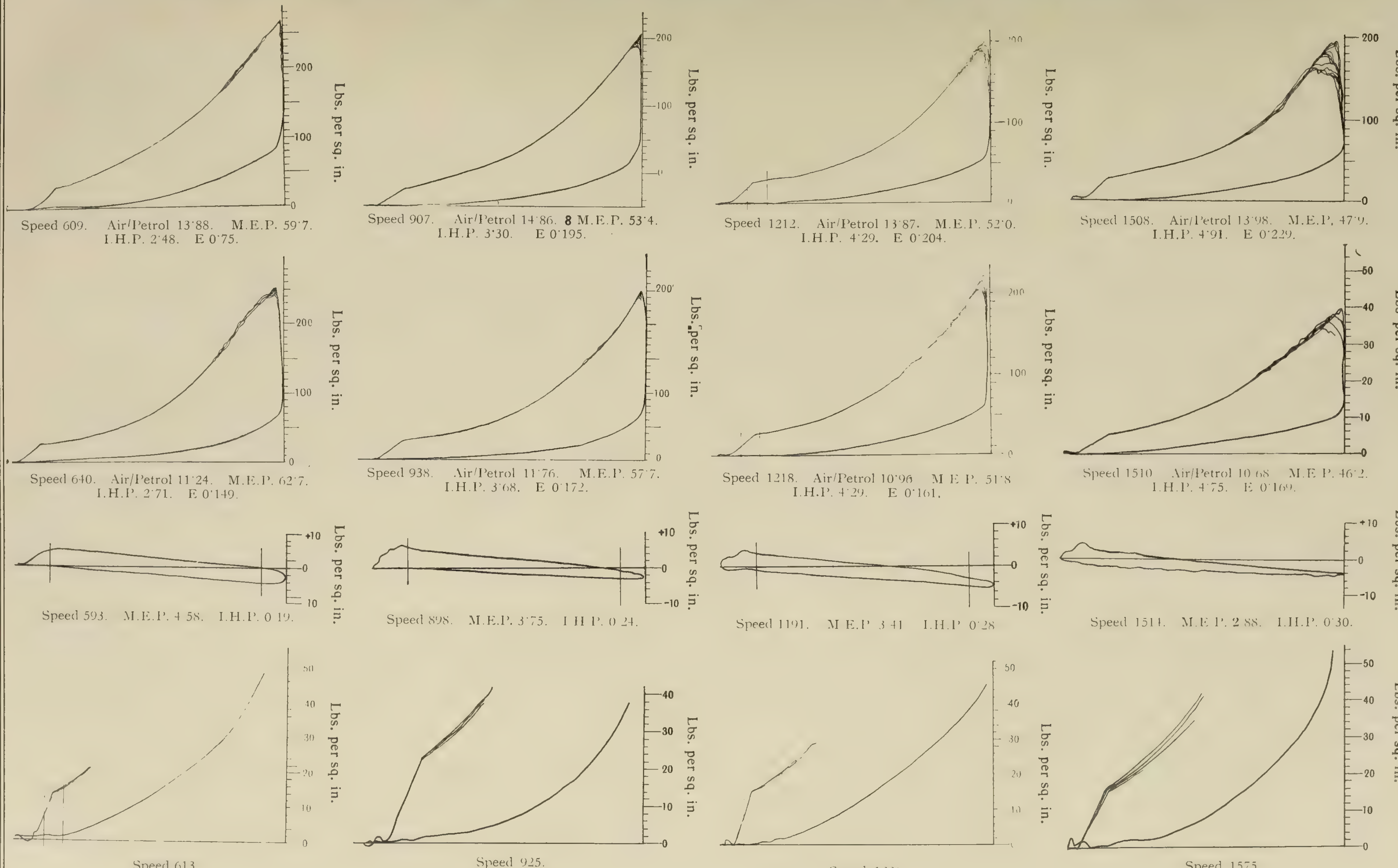


Fig. V. Fig. VI. Fig. VII. Fig. VIII.

in the cylinder, and which is actually burnt, varies much less with the speed, the decrease in the total quantity of air drawn into the engine being, to a considerable extent, counterbalanced by the decrease in the amount of charge which escapes.

Owing to the fact that the exhaust port is always open after the inlet port is closed, the pressure at the commencement

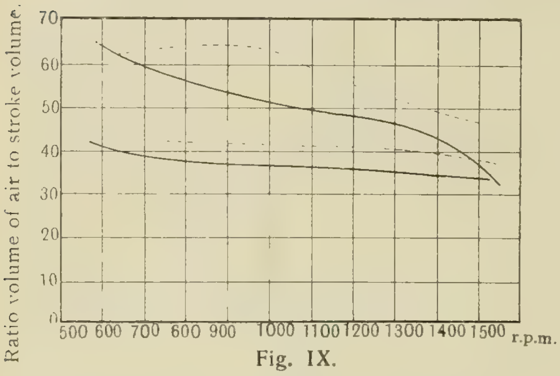


Fig. IX.

of compression is very nearly the same as the pressure in the exhaust pipe, that is, it is practically constant. In the same way the compression pressure is between 62 and 65 lbs. per square inch at all the

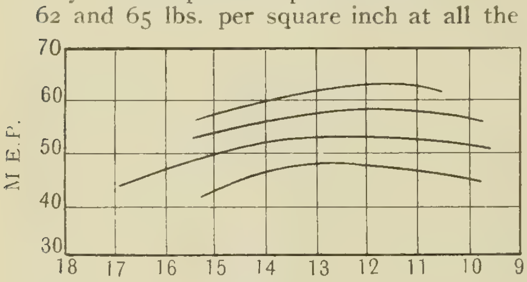


Fig. X.

speeds. It would thus appear that the heat transferred between the cylinder contents and the walls is not appreciably different over the range of speed 600 to 1,500 revs. per minute.

The change in mean effective pressure with speed and richness of mixture is shown in Fig. X. It will be observed, as in the case of the four-cycle engine previously tested, that except for weak mixtures, the M.E.P. remains very nearly

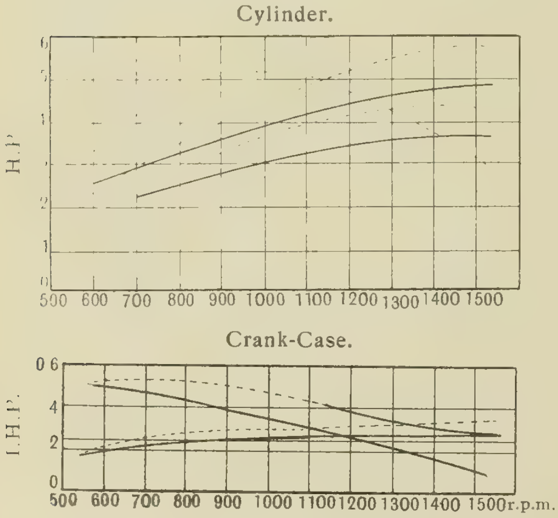


Fig. XI.

constant at any given speed whatever the richness of the mixture employed. The variation of M.E.P. with speed in this engine is, however, very much more marked than in the case of the four-cycle engine. At first sight this is exactly what would be expected, on account of the short time available for the escape of the burnt charge and the introduction of the fresh charge, so that the quantity of charge which enters the cylinder gets less and less as the speed rises. When, however, the matter is looked into more carefully, and account is taken of the amount of the fresh charge which escapes, the problem appears more complicated. As far as the change in M.E.P. is concerned, it is evident that what we have to consider is the way in which the volume of charge remaining in the cylinder (i.e., after deducting the amount which escapes) alters with the speed. Now from the lower full-line curve in Fig. IX. it will be seen that the charge remaining in the cylinder at a speed of 600 revs. per minute is to that at 1,500 revs. per minute as 40.5 is to 35.2, or 1.15 to 1, and if the thermal losses in the cylinder were independent of the speed we should expect the M.E.P.'s to bear the same ratio, while, as a matter of fact, the ratio obtained is about 1.33 to 1. It is difficult to account satisfactorily for this discrepancy. One explanation is that it is due to the comparatively slow burning of the charge in the engine, owing to the large proportion of products of combustion which is always present.*

The effect of this slow burning will be more pronounced as the speed increases. Another explanation is that the proportion of charge which escapes has been over-estimated at slow speeds, or under-estimated at high speeds.

The manner in which the i.h.p. for a constant richness of mixture (air/petrol = 12) varies with the speed is shown by the full-line curve in Fig. XI. The i.h.p. given in this figure, as also in column 3 of the third table is obtained from the area of the diagram given by the working cylinder only, no deduction having been made for the work done in compressing the charge in the crank-case. The amount of power expended in doing the pumping work in the crank-case is shown by the lower series of full-line curves in Fig. XI. It will be observed that it does not vary very much with the speed, and amounts to about 0.25 h.p.

Although the investigation of the b.h.p. of the engine was not seriously considered, yet the i.h.p. when the engine was run light was determined, and from these values approximate values for the b.h.p. at different speeds have been computed, and are entered in column 4 of the table, and are plotted in Fig. XI. It will be noticed that the b.h.p. does not increase much when the speed is raised above 1,200.

Thermal Efficiency.

The thermal efficiency has been calculated on the i.h.p. as defined above, taking into account the heat value of all the petrol supplied to the engine. The results are shown in the table and in Fig. XII. The efficiency increases as the richness of mixture is reduced below the point at which complete combustion takes place—i.e., for mixtures containing less than one of petrol to fourteen of air. This result,

*The proportion of exhaust products to new charge is about twice as great in the case of the two-cycle engine as it is in the four-cycle engine. By determining the crank angle at which contact was made on the commutator, and the interval which elapses with the particular coil employed between the closing of the primary circuit and the passage of a spark, it was found that to obtain the best-shaped diagram at 1,500 revs. per minute the spark had to pass, and hence the charge was fired at a crank angle of 30° before the top of the stroke.

while it agrees with those previously obtained in the case of the four-cycle engine, does not support Prof. Hopkinson's

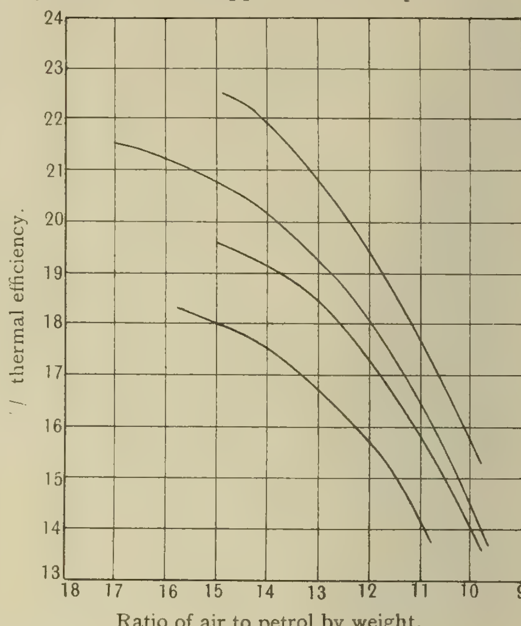


Fig. XII.

values* from the engines he has tested, for he found that the efficiency was a *maximum* when there was just enough oxygen to give complete combustion (i.e., air/petrol = 14).

The change in the efficiency, when the thermal value of all the fuel supplied is taken into account, or what for brevity may be called the gross thermal efficiency, with speed, is largely governed by the effect of loss of charge, and hence it is of interest to calculate what the efficiency would be supposing no loss of charge took place—i.e., to obtain the net efficiency. This can at once be obtained from the numbers for the percentage loss at different speeds. The values for the net thermal efficiencies at the different speeds are given in Fig. XIII., and it will be observed that the change with speed

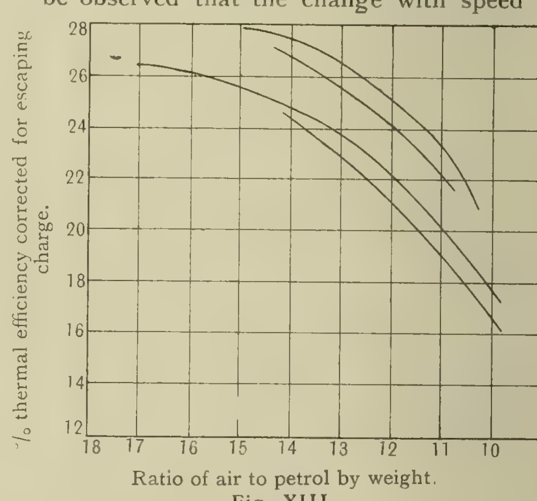


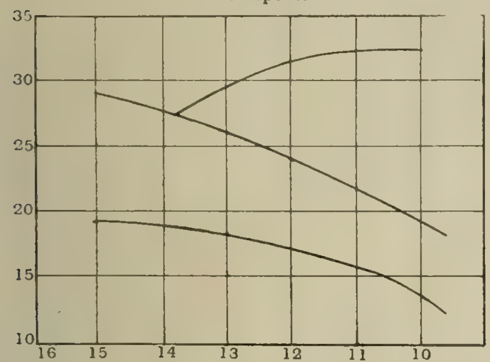
Fig. XIII.

is not very great. The efficiency is higher at a speed of 900 revs. per minute than at the other speeds, the lowest efficiency of all being at a speed of 1,500 revs. per minute. It is of interest to note that the speed recommended by the makers for this engine is 900 revs. per minute. The decrease of the efficiency at the higher speeds may be due to the slow combustion already considered, or to a wrong estimate of the amount of charge escaping. Owing to the agreement between the values for the composition of the exhaust gases, when corrected for the proportion

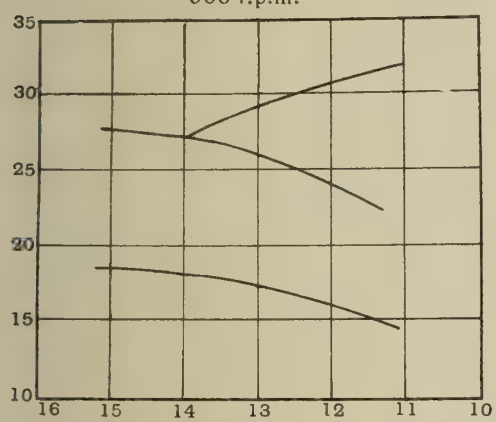
*Proceedings Inst. A. E., vol. iii., p. 220. 1908-9.

of the charge escaping, with the values obtained in the case of the four-cycle engine, the authors do not think that errors in estimating the amount which escapes can account for the decrease of M.E.P. and net efficiency at high speeds.

600 r.p.m.



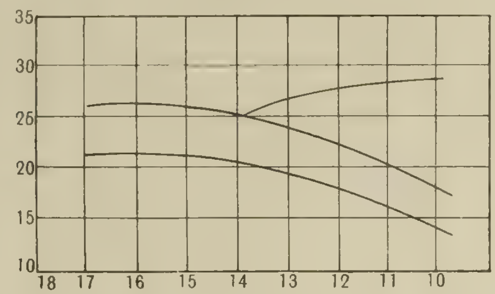
900 r.p.m.



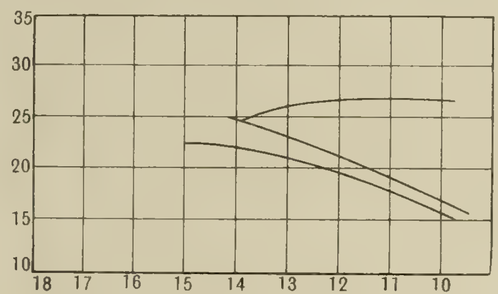
The difference between the gross and net thermal efficiencies at different speeds is brought out more clearly in Fig. XIV. The bottom curve gives the gross efficiency for different air/petrol ratios, the middle curve net efficiency, while the top curve shows the values of the net efficiency if allowance is made not only for the escaping charge, but also for the thermal value of the combustible gases, carbon monoxide, hydrogen, and methane, in the exhaust. The probable explanation of the rise in the top curve as the mixture gets richer has been given in the paper on the four-cycle engine. The correctness of the views there expressed receives support from the calculation of the theoretical efficiency of an engine in which there are no thermal losses due to heat communicated to the cylinder walls, for in such a case it is found that taking into account the change in the specific heat of the gases, the efficiency rises as the mixture gets both richer and weaker than that for which there is complete combustion. Measurements have also shown that the temperature of the exhaust is a maximum for mixtures corresponding to complete combustion.

Since, in the case of the two-cycle engine, the combustion chamber is entirely free from valve pockets, and, in fact, its surface is almost the minimum possible for the given volume, it is of considerable interest to compare the actual efficiency attained with that which might be expected in the case of an ideal engine in which there were no thermal losses due to communication of heat to the walls of the cylinder. When making such a comparison the question as to the compression ratio to be adopted as that of the engine is rather doubtful. We have assumed that the actual compression ratio is the ratio of the volume of the cylinder up to the point at which the exhaust port is first uncovered, to the volume of the combustion space. The value obtained in this way is 3.92, while if the whole cylinder volume up to the out-centre position of

1200 r.p.m.



1500 r.p.m.



Thermal Efficiences.
Lowest curves = Gross Thermal Efficiency.
Middle curves = Net Thermal Efficiency.
Top curves = Net Thermal Efficiency corrected for unburnt Petrol.
Ordinates—Ratio air to petrol by weight.
Abcissae % thermal efficiency.

Fig. XIV.

the piston were taken the value would be 4.67. The reason for adopting the value 3.92 when obtaining the efficiency of the ideal engine is that in the actual engine the work done on the piston during the time it has over-run the exhaust port is very small, and the compression portion of the cycle cannot start till this port is closed.

Two ideal efficiencies have been calculated. In one, the air cycle efficiency, the specific heat of the gases in the cylinder has been assumed to be constant and equal to the specific heat of air at ordinary temperatures. In the other, the specific heat of the gases has been taken as variable, the values being calculated from the tables given in the report of the British Association Committee on explosion temperatures, in conjunction with the results of the exhaust gas analyses. The heat supply assumed in obtaining the ideal diagrams given in Fig. XV. is 453 ft. lb. per cycle, which corresponds to the heat actually liberated in the cylinder when the air/petrol ratio is 13.9 and the speed is 1,212 revs. per minute. The larger of the dotted diagrams corresponds to the "air cycle," and the smaller to the ideal cycle with variable specific heat. The full-line diagram is an actual indicator diagram from the engine running under the given conditions.

The results are summarised in the second table, in which the values given for the actual indicator diagram under (a) are obtained when allowance has been made for the heat contained in the escaping charge, while under (b) no such allowance has been made, but the engine is debited with the thermal value of all the petrol supplied to the carburettor.

Allowing for the escaping charge, the relative efficiency attained on the assumption made as to the compression ratio amounts to 0.76. In the case of the four-cycle engine, having a bore of 85 mm.—the bore of the two-cycle engine being 82.5 mm.—the relative efficiency was 0.77, so that apparently the smaller surface exposed to the hot gases in the two-cycle engine has not improved the efficiency, or, if it has done so, the effect has been masked by some other cause. This result supports the view expressed in the account of the previous experiments, that the thermal loss in these small engines does not vary to any great extent with the ratio of surface to volume of the combustion chamber.

The diagrams given in Fig. XV. and in table II. show the comparatively small increase in efficiency which it is pos-

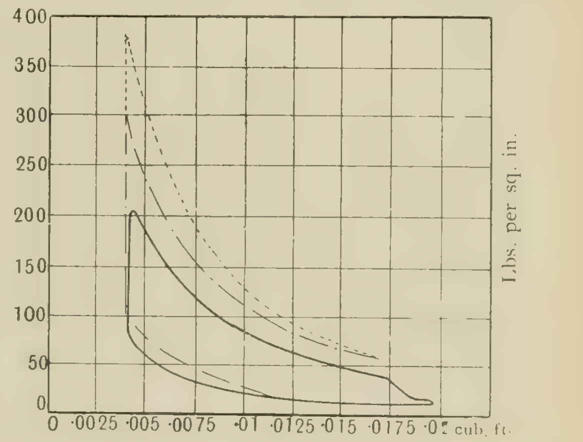


Fig. XV.

sible to obtain, even if we entirely suppress all thermal losses due to communication of heat to the cylinder walls.

A consideration of the crank-case dia-

Table II.

Table relating to Indicator Diagrams, Fig. XV.
Speed, 1,212 revs. per minute.
Ratio of Air to Petrol (by weight), 13·87.

	Thermal Efficiency.	M.E.P. in lb. per sq. in.	Heat Supply per Cycle in ft. lb.	Work in ft. lb.	Heat to Exhaust and Jacket in ft. lb.	Heat in Unconsumed Fuel.	Relative Efficiency.
Ideal Diagram. Constant Specific Heat.	42·1	106·9	453	191	262		
Ideal Diagram.—Variable Specific Heat.	33·9	86·2	453	154	299	–	1·00
Actual Engine (a) ...	25·6	52·0	453	116	337		0·76
Do. Do. (b) · ...	20·3	52·0	573	116	337	120	0·60

grams showed that at high speeds there was a very large amount of wire-drawing in the admission port to the crank-case and in the carburettor. To see what improvement in the power developed at high speeds would be obtained by lengthening the time the port was open, ⅛in. was cut off the bottom edge of the piston opposite the port. In this way the port was open for 80 degrees of crank angle as against 70 degrees. The new valve diagram is shown in Fig. II. (b).

The results obtained under the new conditions are given in table III., and are shown on the figures by means of dotted curves.

Owing to the increased length of time during which the crank-case is in communication with the carburettor, it fills up better at all speeds except 600 revs. per minute. At this low speed it completely filled up with the previous setting, and hence the effect of keeping it open longer on the down stroke of the piston is to lessen the volume of charge retained in the crank-case. At the high speeds the increased volume of charge taken is quite marked, as is evident from a study of Fig. IX., while the proportion of charge escaping is also increased, see Fig. IV.

The effect on the i.h.p. and b.h.p. is indicated by the dotted curves in Fig. XI. The effect of the change has been to decrease the i.h.p. at 600 revs. per minute by 8 per cent., and to increase the i.h.p. at 900 revs. per minute by 12 per cent., at 1,200 revs. per minute by 21 per cent., and at 1,500 revs. per minute by 19 per cent. The gross efficiency at the higher speeds is decreased owing to the increased loss of charge.

Comparison of Two and Four-cycle Engines.

When comparing the working of this two-cycle engine with an ordinary four-cycle engine it is to be noted that the range of mixture richness which it is possible to use is considerably smaller with the two-cycle than with the four-cycle, due to the very much larger admixture of exhaust products with the fresh charge. Unless the richness of mixture is adjusted within comparatively narrow limits, particularly at the high speeds, the engine refuses to work on the two-cycle, and only fires on every other out-stroke, the intermediate stroke acting as a scavenging stroke. The result of this peculiarity is that unless the carburettor provides a mixture of uniform richness at different speeds and for different throttle openings, satisfactory working cannot be obtained. In the case of the engine under test the effective use of the carburettor jet was hand-adjusted in every case, but even then, at a speed of 1,500 revs. per minute it was often difficult to exactly hit off the correct mixture.

The easiest way of comparing the power developed with that of a four-cycle engine of the same size is to take double the M.E.P., and compare this quantity with the M.E.P. given by a four-cycle engine. In this way we get :—

Speed (revs. per minute).	Piston Speed (feet per minute).	M.E.P. (lb. per sq. inch).	Twice M.E.P. (lb. per sq. inch).
600	324	57·4	115
900	487	65·5	131
1,200	648	63·4	127
1,500	810	55·4	111

It is thus evident that at the low piston speed at which this engine works the power developed compares quite favourably with that of a four-cycle engine working at approximately the same piston speed. Thus, in the case of the Siddeley engine tested by Prof. Hopkinson and the Clement-Talbot engine tested by one of the authors we have :—

TABLE III.—(B. TESTS).

Test Number.	Speed R.P.M.	I.H.P.	B.H.P.	Lb. of Petrol per 1,000 revs.	Lb. of Petrol per I.H.P. hour.	Thermal Efficiency, Gross.	Thermal Efficiency, Net.	M.E.P. lb. per sq in	Air/Petrol by Weight.	Amount of Charge which escapes, per cent.
41	601	2·34	1·71	0710	1·093	·125	·184	57 2	10.28	32
42	606	2·46	1·83	·0723	1·078	·128	·191	59·7	10·33	33
43	602	2·30	1·67	0679	1·065	·128	·190	56·1	10·81	32
44	604	2·34	1·71	·0586	0·910	·150	·214	56·8	12·37	30
45	922	4·11	3·34	·0698	0·940	·146	·221	65·5	10 75	34
46	1,200	5·18	4·20	·0639	0·888	·154	·210	63·4	10·28	27
47	1,502	5·74	4·64	0537	0·841	·163	—	56·2	10·35	—
48	1,506	5·44	4·16	·0457	0·760	·180	·218	53·0	12·13	17

Engine.	Piston Speed (feet per minute).	M.E.P. (lb. per sq. inch).
Siddeley	450	89
,, ...	790	88
Clement-Talbot	552	87
,, ...	868	87
,, ...	1,024	83

A further advantage of the two-cycle engine is that the maximum pressures attained are much lower. Thus, at a speed of 600 revs. per minute the mean maximum pressure attained was 260 lbs. per sq. inch gauge pressure. This fact is quite worth while keeping in mind in cases where extreme lightness is of the utmost importance, since the weight of an engine depends in a great measure on the maximum pressure which occurs in the cylinder. Of course, if the two-cycle engine fails to fire the charge on any stroke, the pressure reached on the next stroke is very much greater than the average. Thus, in the engine tested, after a misfire the pressure rose to over 360 lbs. per sq. inch. Hence, if an engine on the two-cycle were designed for the low pressures which obtain when working on that principle, and if, for the sake of lightness, the parts were cut very fine, it would probably be necessary to provide a relief valve on the cylinder head to allow for the high pressure reached after an accidental misfire.

Discussion.

Mr. T. W. Browne hoped Dr. Watson would pursue his investigations further and deal with more recent types of two-cycle motors, such as the Lamplough, the New Engine Co.'s, and the Lucas engines. Dr. Watson, he said, was the first to show what really happens in the inlet and exhaust pipes of a petrol motor. The two-stroke motor had hardly been used on motor cars at all, and the reasons were obvious. The two-stroke motor of this simple type could not be varied much in its speed. It did not run well at low speeds and only ran then like a four-cycle motor. On motor boats the conditions were different, as regular speeds were obtained for long periods, and the motor could run for a long time without being touched. The carburettor could be adjusted so that it could run at an absolutely even speed. The advantages of this type of engine were its great simplicity and the fact that a three-cylinder two-stroke engine would give the same evenness of torque as a six-cylinder motor of the four-cycle type. The motor used by Dr. Watson had a very short stroke, and it would be interesting to compare those experiments with others conducted upon a long stroke engine. The disadvantages of this simple type of engine were crank case compression, the deflector plate and tendency to produce pre-ignition where there was carbon deposit. Worst of all, there was imperfect valve setting—no lead was possible, as any alteration of the time of opening meant a corresponding alteration in the closing. At the same time there were no big poppet valves, and likewise no reciprocating sleeves. It was also well known that the two-stroke motor was very wasteful at low speeds. One conclusion to be drawn from the paper was that a different valve setting was needed for different speeds in this type of motor, at all events as regards the port from the crank case. In this connection he suggested the use of a mechanical arrangement by which the driver could vary the size of the port while the engine was running. As to getting an absolutely even carburation at all speeds there ought not to be much difficulty about this with modern carburettors, within limits, though perhaps

there would be practical difficulties in doing this with crank case compression. It appeared from the tables given at the end of the paper that a three-cylinder motor of this type compared very favourably with the same size of engine running on the four-cycle system.

Mr. R. W. A. Brewer said he had been experimenting for two months past with a Lamplough two-cycle engine, and gave it as his experience that this was one of the most difficult problems he had ever tackled. He was sorry that the engine Dr. Watson had experimented with was so old-fashioned, as many of the faulty points inherent in its design had been overcome. It was curious to note, however, that in many respects the results he had obtained agreed with Dr. Watson's. Although the engines were so different there was a very remarkable agreement in regard to their thermal efficiencies. After referring in detail to the results he had obtained he said the valves of the Lamplough engine were so arranged that under ordinary conditions it was impossible for any mixture to blow back into the cylinder. This system of timing was brought about by a particular arrangement of curved connecting rod. He had abtained a minimum speed of 278 r.p.m., and a maximum speed of 3,000 r.p.m. simply by opening and shutting the throttle and not making any other alterations at all.

Mr. W. A. Tookey asked whether the authors would supplement the figures given for the mixture strengths, so as to give an indication as to the number of British thermal units of heat per cubic foot or per lb. The table would then be most useful to other engineers who are interested in internal combustion engines generally, and not petrol engines in particular.

The speaker thought that if the authors had taken into account, not the richness of the mixture entering the cylinder—even when corrected for the leakage referred to—but the number of British thermal units retained after the exhaust ports had been covered per cubic foot of "stuff" (air, petrol vapour and residuals), they would find that the inexplicable difference to be noted in connection with the 900 r.p.m. series with regard to net thermal efficiency would be non-existent. From a series of four lantern slides, Mr. Tookey illustrated his remarks upon this point. The first showed that the air/petrol ratios adopted by the authors differed from the B.Th.U. basis proposed by the speaker. The second showed that the net thermal efficiencies when compared upon the new basis were regular and rational. The third showed that the relation of mean pressure and mixture strength was but slightly altered from those shown by the authors, and had the same characteristic of lessened pressures with weakened mixtures. The fourth slide indicated, however, that such reduction affected only the power of the engine—the ratio between mixture strength and M.E.P. being uniformly increased with decreasing richness.

Mr. Tookey believed that the method of comparison he proposed had the merit of novelty, and he was of the opinion that by its adoption some useful ratios were to be obtained which enabled the performances of engines of different compression pressure ratios and scavenging arrangements to be better appreciated.

Mr. Day, the inventor of the engine upon which Dr. Watson had experimented, explained that the particular engine which had been dealt with was one ordered by Dr. Watson, but at the time it was ordered he (Mr. Day) had not the slightest idea what was in store for it, or he would have put it upon the best diet he could, and have given it some special treatment, when no doubt it would have shown better results. Still, he thought, from a practical point of view the results were eminently encouraging. The engine used was one of a new type which were being made in batches of fifty, and this was the second of the first batch, so that it would readily be seen that it was capable of improvement, and as a matter of fact improvements had been made. He (Mr. Day) realised its faults as quickly as anyone, before they were pointed out by Dr. Watson. He questioned Dr. Watson's statement as to the amount of charge that escaped. He could not think it possible from a practical point of view that as much as 35 per cent. could have been got rid of through the exhaust port. If the engine was losing this enormous portion of its charge, how did it come about that the thermal efficiency was so good?

Mr. Hounsfield said he gathered from the paper that the chief weaknesses of the two-cycle engine were the loss of fresh charge through the exhaust port, the narrow range of ignition possible, and the narrow range of mixture that would give ignition with this type of engine. He asked whether any other weaknesses had been found, such as excessive quantity of lubricating oil being necessary, or oil being impaired by being mixed with petrol, or firing in the base chamber to any extent. He asked Dr. Watson to state what he considered the temperature of the cooling water was.

Mr. Wood suggested that in future experiments Dr. Watson might deal with some of the conditions of general interest to motorists as apart from the merely gas engine point of view, as about 90 per cent. of the running of a car was done at between one-third and one-half load. In regard to the difficulty of firing incorrect mixtures at high speeds, he asked whether that had anything to do with the difference between the coil spark and the magneto spark. It would be interesting if Dr. Watson would try this engine on magneto ignition at the higher speeds.

The President explained that the use of the exhaust pipe for scavenging purposes was the invention of Mr. Atkinson. He asked the author whether the consumption of lubricating oil was excessive, as the cost of oil, added to the cost of fuel, would make a material difference in the total cost of running.

Dr. Watson, in his reply, said the two-cycle engine would not run well at very low speeds. He disagreed with Mr. Browne's suggestion for getting equal strength of mixture at all speeds. He was much interested in Mr. Brewer's experiments, as also in the remarks of Mr. Tookey, but deferred his reply. In regard to Mr. Day's scepticism as to the loss of charge, he was afraid he could not alter his views. He would include a torque diagram, as requested by Mr. Legros. In reply to the President, he said he gave the engine plenty of lubrication, but did not measure the quantity of oil; he was too much occupied in taking indicator diagrams. The temperature of the cooling water was about 58 degrees Cent., but it did not seem to affect the running of the engine. In regard to the question of spark-coil *versus* magneto, he said he had always found that the efficacy of the spark had nothing to do with its strength. It was the timing and things like that which made the magneto better than the coil for ignition purposes. In all cases he found that the charge burned just as quickly whether it was ignited by a small spark from a coil or a large spark from a magneto.

CASTINGS TO WITHSTAND HIGH PRESSURES.

ALTHOUGH the recent paper on castings to withstand high pressures, read by Professor Carpenter before the Institution of Mechanical Engineers, does not contain very much which is of interest to those connected with the design and construction of automobiles, nevertheless there are some points which are of great interest, and from which much can be learned.

The paper dealt with gun-metal and aluminium-bronze castings in particular, and was an account of a series of tests to determine the suitability or otherwise, of these materials for constructing hydraulic engines in which the pressures may be extremely high.

The troubles which are so often met with in foundries, due to characteristics of the metals, or small errors in the moulds, were dealt with in a clear, comprehensive manner, and this same thoroughness is characteristic of the whole paper, so that it is not surprising that the resultant castings, being cylinders, about an inch in diameter and three-quarters of an inch in thickness, withstood an average pressure of 18 tons per sq. in.

In connection with the casting of these cylinders there occurred an incident which again indicates the effect of a sharp un-radiused corner when bearing a heavy stress, as when the tops of the cylinders had not been carefully machined the latter burst at very much lower pressure than those which were provided with a suitable radius at the junction of cylinder wall and head. Another point in connection with these particular castings which may be of interest, was the inability of those cast in either green or dry sands, to stand up to the work they were intended for. While those cast in chills withstood an average of 18 tons per square inch, sand castings would but rarely hold 7 tons, the authors accounting for this fact by reason of the segregation of tin-rich areas and the trapping of gas bubbles between the crystals during the large freezing interval of the alloy.

Further experiments were conducted in aluminium-bronze, a material but little used in connection with automobile work but which appears to have certain

properties which may be of use, while a very interesting point is brought out concerning the alumina which, forming on the walls of the mould, militates against the production of a sound casting. A considerable number of experiments were made before this difficulty was overcome, but information was gathered from these failures which the authors condensed as follows:—"The pure aluminium-bronzes are very viscous even at the highest temperatures, and pour in a thick and sluggish manner. The sluggishness is not directly due to the metal itself, but to a tenacious film of alumina which invariably covers the surface of the alloy. This can be demonstrated by stirring a crucible full of alloy and noting the surface. It will be seen that the surface-film comes to rest while the metal below is still moving rapidly. Accordingly it may be supposed that the liquid metal offers no unusual resistance to the natural tendency of the dross to rise to the surface. Consequently, as in all cases great care was taken to pour the metal absolutely clean, the authors were convinced that the dross was forming inside the actual mould."

After this discovery means were adopted whereby the metal could be watched, and it was found that a skin formed rapidly on the clean surface of the alloy if the metal entered rapidly. Considerable agitation of its surface caused the exposure of a much larger area than was the fact when a steady slow feed was adopted, while a considerably greater amount of alumina was formed. Accordingly, to quote the words of the paper, "It follows that aluminium-bronzes should be poured as quietly as possible and every care should be taken to avoid agitation of the metal." The final part of the paper dealt with castings made with this rule in mind, which castings stood the 18-ton pressure with ease, while the percentage of scrap was greatly reduced.

The introduction of small amounts of manganese caused the castings to have a spongy nature and to give way under test at a considerably smaller load than without the admixture, while another point, brought out during the final part of the tests, was the, not always recognized, necessity for keeping the time occupied in melting alloys as constant as possible, and the need for watching the exact composition of the furnace charge carefully, it being found that the varied nature of the castings produced, their liability to sponginess and to airholes, was caused by using gates and old castings as well as charges consisting of pure copper and pure aluminium, both taking different times to melt.

When greater care was taken in the composition of the charge, together with a careful watch on the time taken in melting, a much higher quality of casting was produced, while the percentage of scrap fell to a minimum. Again it was found advisable to keep the molten alloy in its crucible as long as possible before pouring, since the solubility of the contained gas falls as the heat, and an air temperature of 50 deg. C. was the lowest at which it was found desirable to pour.

There was very little attempt at a serious discussion upon the conclusion of the paper, those criticisms which were made being almost entirely on the commercial side and concerned more with the prices of the metal and general cost of manufacture than on the properties of the alloy itself.

The large gate which was used in obtaining all these castings was the subject of a great deal of cricitism, but as the joint author, Mr. C. A. Edwards, remarked, it was better to obtain a sound casting every time with a large gate than an uncertain one with a small gate, moreover, since the gates may be remelted and used for purposes where such great strength is not needed they cannot be regarded as scrap-metal.

A remark was made concerning the difficulty in persuading buyers to use certain alloys since they were almost invariably tied down to specified materials. This is really a greater difficulty than would appear at first sight, not only with castings of this nature but with all, as the buyer of a large firm sets down certain compositions which must be supplied to his firm, and which must not be varied in any way, since this particular specification has been found satisfactory and the firm are exceedingly unwilling to alter it, or to adopt an entirely unknown one, even in the face of numerous tests which the vendors may bring before their notice.

Of course, this is an old complaint, and one that all inventors or makers of new things have to contend against, but conservatism of this kind is, luckily, decreasing.

SIMPLE TESTS FOR PETROL VAPOUR.

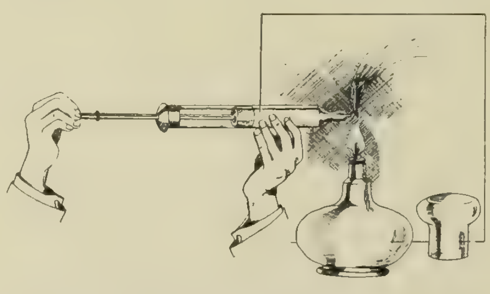
The detection of petroleum vapour or gas is a subject of considerable interest to those who store or handle petrol, and a simple test for the presence of vapour is a very useful thing to know. No automobile engineer is the worse for knowing the exact conditions under which a mixture of vapour and air becomes inflammable or explosive, and the paper recently read by Mr. John Heck before the Institution of Engineers and Shipbuilders in Scotland dealt with a simple method of testing which exactly meets practical requirements.

The apparatus required is simple, and consists merely of an ordinary glass medical syringe, of about 4 ounces capacity, preferably with a fairly long nozzle, a glass spirit lamp burning methylated spirit, and a blackened sheet of cardboard, wood or tin.

A sample of the air to be tested is drawn into the syringe, which in the case of very weak mixtures of air and vapour should be filled and emptied several times with water by a few strokes of the plunger.

Heating the vapour also makes the test flame more luminous, and so more easily observed, hence the nozzle of the syringe should be kept in the testing flame for a short time, until it reaches a dull red heat. The end of the nozzle being finally brought quite close, and just below the tip of the flame from the spirit lamp with the syringe in a horizontal position, the contents are then expelled by a series of gentle jerks or pushes of the piston, so as to cause the mixture to issue in puffs, the observer's eye being in front, and a short distance above the top of the flame.

If the atmosphere or mixture tested is inflammable or explosive, the mixture, as it issues from the nozzle, will at once ignite, and burn at each puff with a somewhat blue colour for poor, and with a yellowish blue for richer mixtures. With weaker mixtures the stream of particles issuing from the nozzle are carried upward above the tip of the test flame, and appear at each puff as a bright and incandescent streak or cap, and with very weak

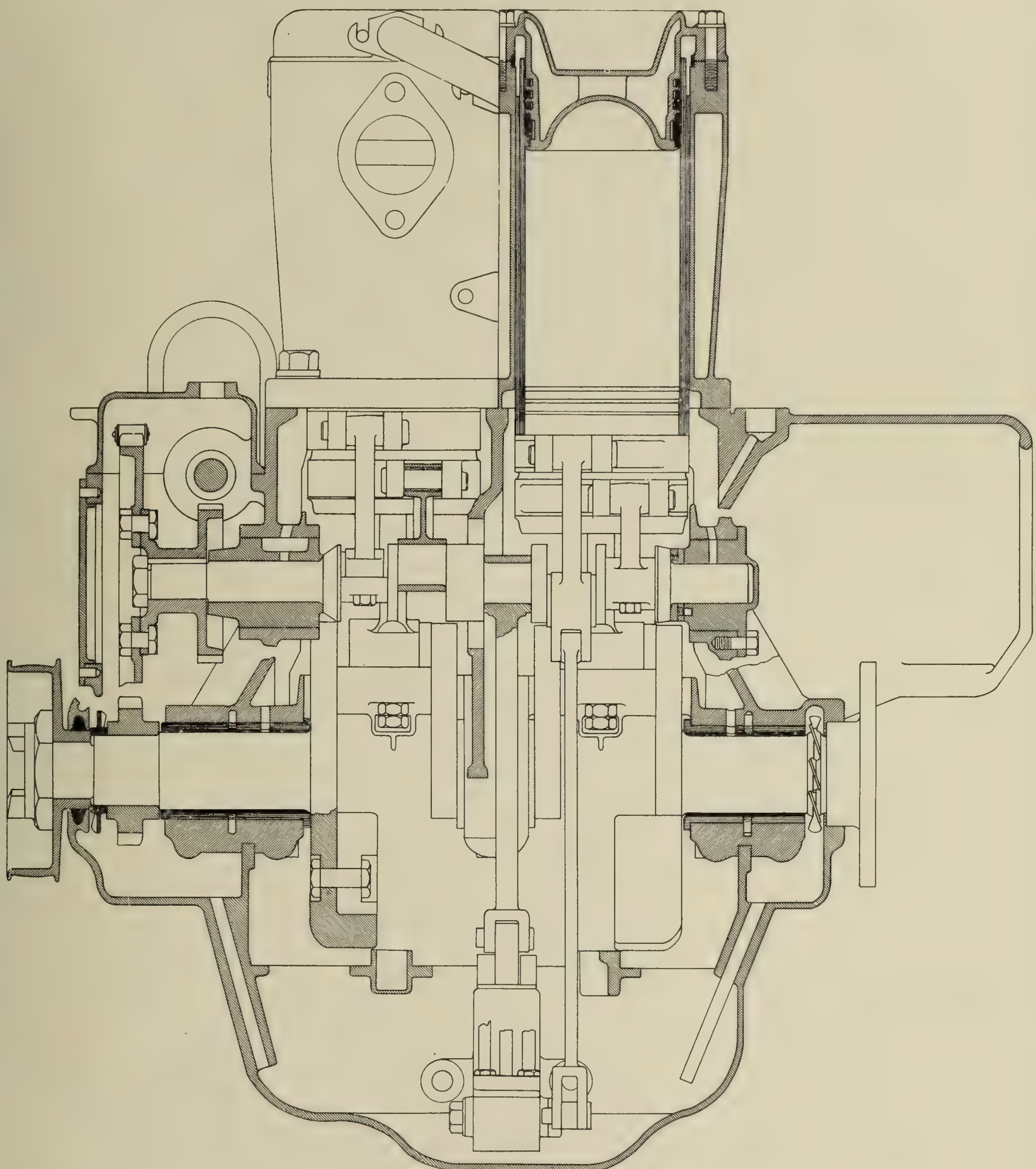


mixtures the puffs present a fainter and less vivid appearance. When the atmosphere tested is free from vapour, no lining or streaks can be detected, the only result being a slight flattening or shrinkage of the tip, in which case it may be taken as absolutely certain that there is no trace of vapour present.

The heating of the syringe contents is important, as it renders the test so very much more sensitive, the considerable increase in brightness in the appearance of the whiffs or streaks being probably due, among other causes, to the non-chilling of the flame, and the separation or partial separation of the carbon contained in the gas. Many experiments have been made with various mixtures of air and petroleum vapour, and when the samples were well heated before testing it was found possible to detect the presence of 0.25 per cent. of vapour in an atmosphere —an amount far below that necessary to form an inflammable mixture. Spaces were also rendered inflammable and explosive, and afterwards gradually purified, until the hot test indicated freedom from vapour, which was then verified by the insertion of a naked flame. The results were found to be much the same, whether the flame was from methylated spirit or from hydrogen, but candles, petroleum oils, or benzoline were found to be of little use. As regards the syringe, glass was found to give better results than metal, the latter probably giving a greater cooling effect, as well as being more difficult to clean.

In testing an atmosphere it should be remembered that petrol vapour is comparatively heavy, and hence the sample should be taken from near the floor, or at the bottom of the receptacle.

Many instances will occur to readers in which a simple and easily applied test such as this would make for safety. There have been many mishaps in effecting repairs to tanks which have held spirit, and have not been free from vapour at the time of brazing or soldering.



THE 12 H.P. TWO-CYLINDER ROVER—KNIGHT ENGINE.

[This engine was described briefly in our annual issue published last month. The camshaft drive is, of course, by chain, and it should be noticed that there is a slide valve to control the plunger oil pump. A peculiar point in the design is the small paddle wheels fitted to the crankshaft in place of the usual oil thrower rings.]

CORRESPONDENCE.

No responsibility is accepted by the Editor for the opinions of correspondents, and the right is reserved to omit any portion of a letter if it is considered desirable to do so. Letters to be inserted with a nom de plume must be accompanied by the name and address of the writer. If publication in the next issue is desired letters should reach the Editor in no case later than the 20th of the preceding month.

CARBURETTOR ACTION.

Sir,—I have just been reading in your December number Professor Morgan's paper on carburettor action, and as one who has dabbled and experimented a lot with carburettors I would like to ask Professor Morgan a question.

If he is right in his summing up that "there appears to be no necessity in any carburettor for extra air devices or the multitudinous contraptions employed" how is it that the addition of these multitudinous contraptions have so improved the running of engines from what they used to be with the original elementary devices employed?

It is not for want of any constant or inconsant extra supply of petrol, for I have tried carburettors with large jets and small jets, with large choke tubes and small choke tubes, and it was invariably the case that giving more air at the higher speeds increased the engine power.

Professor Morgan's remarks are, however, qualified by the words "there appears to be," and I make bold to say the more Professor Morgan works with carburettors the more he will find appearances to be deceptive, for the fact of the matter is that there is nothing more tricky than the action of small petrol jets which will appear to follow one law to-day and a different one to-morrow, also they vary with every twist and turn of the air passages as well as with the weather and temperature conditions.

Mr. Brewer and Mr. Bickford were both right in some of their remarks, but I am quite convinced that their combined wisdom coupled with that of Professor Morgan will fail to find any law which a carburettor jet will always follow.

J. JOHNSTON.

BIG END DESIGN.

Sir,—In reference to the design of connecting rod big ends, as shown in the assembly drawing, Fig. I. of the Armstrong-Whitworth engine in your December issue, I feel sure that it would interest your readers to know the opinions of engineers regarding this bearing arrangement. The point to which I should like to draw attention is the one-sided placing of the big end bearing in such a way that the centres of pressure and area are not coincident.

I should be very interested to know whether this is theoretically permissible. I rather think that the centre of pressure should fall within the "middle-third" of area of the bearing, as is assumed in masonry arches, etc., to prevent tension in the mortar joints. Of course, I am well aware that this asymmetric design has been in use for many years by several well-known makers, but in cases where the practice has been carried to excess—and there is a great temptation to make use of the increased space afforded by the adoption of the three-bearing crankshaft in four-cylinder engines—the big ends have invariably shown a decided tendency to wear "bell-mouthed."

"BIG END."

ESTIMATING HORSE POWER.

Sir,—In my letter on the above subject published in your issue for December, I notice several errors, the chief being in the formulæ, in which the index of r, the ratio of stroke to bore, has been printed as a factor instead of as index. The following amplification may also prove interesting to some readers.

If we assume that the inertia stresses in the connecting rod limit the piston speed, and also assume that the masses of the reciprocating parts are influenced by the diameter alone and not by the stroke (being in consequence proportional to D^3), then the piston speed which gives equal intensity of inertia stresses is proportional to $\sqrt{r}$, and the horse power per cylinder is $0.4\,D^2 \sqrt{r}$, at a piston speed of $1{,}000 \sqrt{r}$, feet per minute. But, as Lanchester pointed out, the mass of the piston shell (i.e., the thin portion which acts as guide) and the mass of the connecting rod shank are influenced by the stroke, varying as D^2S (i.e., D^3r), instead of as D^3; and in a typical design he found the reciprocating masses proportional to $D^3\,(1+0.3\,r)$. The piston speed for equal inertia stresses then becomes $1{,}140/\sqrt{\frac{1}{r}+0.3}$ feet per minute. The S.M.M.T. committee showed graphically that this expresssion is approximately represented by $1{,}000 \sqrt[3]{r}$. Hence Lanchester's formula is $0.4D^2 \times \sqrt[3]{r}$ at a piston speed of $1{,}000 \sqrt[3]{r}$ feet per minute, and Callendar's is $0.2D\,(D-1)\,(r+2)$ at a piston speed of $1{,}000\,\frac{r+2}{3}$ feet per minute, $\frac{r+2}{3}$ being an approximation for the cube root of r.

The table of horse powers published with Mr. Brantsen's letter proves nothing more than that, if the definition of the horse power is varied, its numerical value varies accordingly. No advance in horse power formulæ is possible until it is understood that in a modern petrol engine the horse power is simply proportional to the speed within the limits of safe running, hence for purposes of rating the piston speed must be defined, and every formula for horse power has a certain piston speed, or revolution frequency, explicitly or implicitly assumed in its derivation.

J. B. HENDERSON.

BALL v. ROLLER BEARINGS.

Sir,—Replying to the letter in your December issue, written by your correspondent, Messrs. The Auto Machinery Co., Ltd.

I notice there are one or two points upon which they disagree with my letter in your November issue.

A considerable portion of the Auto Co.'s letter is taken upon the assumption that I am in error in stating that it is common practice to strike the radius of the bearing surface from 7/10 of the ball's diameter.

I notice that it is said, they do not know any firm who is *now* making bearings using this proportion. I was not aware any change had been made. But let us (for the sake of argument) admit that I am wrong in making this broad and general statement; surely, it would have been wiser to have challenged the figures and not the statement.

In Table No. 1 the area given for the annular bearing C is, .01108. Now, before saying that all my figures are wrong, it would have been safer to have ascertained whether the figure given (.01108) was in exact conformity with the statement, and if this reasonable precaution had been taken, the Auto Co. would have found that they were basing their remarks upon something that did not exist, and if they will be good enough to measure up their own area of their own ball bearings, as well as those made under the common practice (which they say is not what I state), they will see that the area given in my figures is quite liberal, and will cover the assumed inaccuracy.

It would also have made matters a little more clear to your readers, if, in making a correction of this kind, the Auto Co. had shown *to what extent* my statement was wrong.

I have checked bearings in my possession, and I find that they are struck as stated by me in my letter—7/10 of the balls' diameter. According to the Auto Co., this radius has been reduced, but such reduction does not seriously affect the figures that have been given. I am quite open to any correction the Auto Co. are good enough to show, but I respectfully beg to submit that they should show their correction in actual figures, and not in general statements.

The same remarks apply when the Auto Co. refer to Professor Stribeck's equation. There is really very little in the point raised—it is another way of saying it. As regards the thrust or axial load; it was distinctly stated in my letter that this was taken as a simple axial load, so that no complication was introduced in these figures, which are taken on the hypothesis that every ball and every roller are engaged under working load.

I admit that in this case, any variation of the curvatures of the ball race will give different results, but before I can admit the Auto Co.'s correction, I should want to know what is the difference in the curvatures they refer to, and what percentage of difference will it make to the load. I respectfully venture to suggest that the difference is so infinitesimal that the alteration in the figures is not worth mentioning.

As regards adjustability, here again, the Auto Co.'s line of argument is unreasonable. They take a case that I illustrated in my letter, of a stationary ball race showing signs of abnormal wear upon 25% of its circumference, and they then build up upon this abnormal condition, the argument that no adjustment would compensate for such a condition. Obviously it would not, and nobody ever suggested that it would, and to build up a case of this description, is liable to mislead one from the true facts.

Surely, the illustration in question was for the purpose of showing that the point to point surface of the ball was insufficient in this case to carry the load, and, therefore, the ball had worn itself a larger bearing surface, and it is well known that immediately a ball commences to do this, a spinning motion is imparted to it while it is under its load, causing both the ball and the bearing surface to destructively wear each other.

With a roller bearing, giving an area of .17000, or a percentage of 2,680% against 175% for the ball, or an excess of area in favour of the roller of 2,505%; it is quite clear that no such local wear could take place; therefore, the imaginary difficulty in reference to the adjustability of the bearing never could commence to exist; therefore, the Auto Co.'s argument has no existence.

I think I have now shown that there is no basis for the Auto Co.'s letter. As regards the Auto Co. having manufactured ball bearings which have successfully travelled 45,000 miles; I do not question this statement for a moment. I can only say that it is a well-known fact to the agents and motor car repairers throughout the country, that very many cars, of both English and foreign manufacture, frequently have as many as four sets of bearings fitted in twelve months upon their front wheels, and I make this statement with bona fide letters before me from persons who are desirous of trying roller bearings.

R. F. HALL.

Sir,—In reply to my critics. Mr. Sinclair objects to the single row bearing for a back wheel on the ground that a shaft 1½in. diameter will not stand side slips into kerbs. The fact that such axles are in practice made too small is no argument against the correctness of the theory. Given a material with an elastic limit of 40 tons per square inch, the shaft to be stronger than the strongest of wire wheels would have to be 2¼in. in diameter. If the sleeve type of axle is used practice insists on a solid axle of 1½in., at least, and the outer diameter of the hollow sleeve axle would have to be 2.6in. *with the same* elastic limit.

Of course, neither system would require such section throughout the whole length, but it is clear that so designed each system would be equal against extreme shocks, and that the single row bearing type would have a slight advantage in the matter of weight, and a very great advantage in cost of production.

To make comparison easy I have calculated the size of bearings required on the not unwarrantable assumptions that the *volume* of the

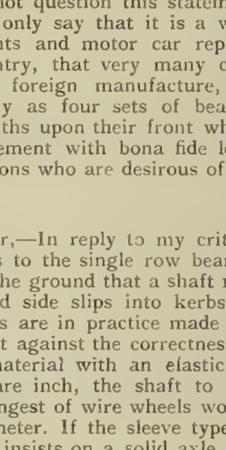


Fig. I.

bearings is a satisfactory index of their load carrying powers, and that a ball bearing 72 mm. external diameter and 20 mm. internal diameter is satisfactory for the outer bearing for a front wheel. My own experience and the general practice is sufficient to warrant the second assumption.

I do not know, and there is no way of finding

out, short of prolonged trial on the public, whether jolts when travelling straight, side slips and kerbs, or the stresses turning corners are the conditions that make bearings give out, so I have designed the hub shown in Fig. I. on the basis that gives the outer front bearing the least work to do in comparison with that to which the

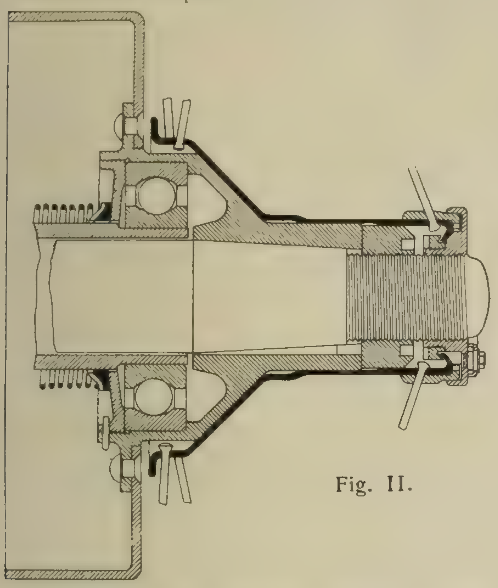


Fig. II.

other bearings are subjected, namely, during shocks.

Fig. II. is a design for a rear hub, in which it has been assumed that turning corners is the essential condition; this is not altogether improbable, in view of the fact that the force on the outer bearings of a rear wheel under a side shock of 8,000 lbs., is equal to that on the front outer bearing, viz., 24,000 lbs., while that due to jolt is only 4,650 lbs. (The jolt effect on an outer front bearing is only 800 lbs.)

I do not know which assumption is correct, but if jolts are the ruling condition, then a larger hub is required, with the sleeve type, and this surely is a serious handicap, especially where detachable wheels are used, as the same large hubs must be used in front as well and carried as a spare, too.

Now as to types of bearings. I make no claim to be an authority on these, though my experience with cup and cone bearings, covering as it does the production of over 600,000 bicycles, each having twelve bearings, has naturally given me many opportunities of studying them, and I feel quite content that under equal conditions the cup and cone bearing would show a slight superiority to the journal type of bearing, quite apart from the fact that the cup and cone type can be adjusted.

There are many difficulties to be solved though, and when The Auto Machinery Co. suggest I am not serious, I agree, if by that they mean that I am not to-day prepared to put money and energy into a scheme for making cup and cone bearings for automobiles.

The obvious thing to do would be to slightly shift the spinning axis of the balls as in Fig. III., and by so doing make it an adjustable cup and cone bearing, but I do not know how to prevent people from screwing them up too tight, or from putting them in the wrong way round.

A bearing of this type was once put on the market, and to my knowledge a racing car turned up in the Isle of Man, and on the driver complaining of looseness in the hub, an examination revealed that this putting the bearings in wrong way round had actually happened.

This is not perhaps strictly a technical defect, but it is one that would kill a scheme, and engineers have to design devices that are suitable for the multitude, which contains a proportion of persons who will not know how a particular device should be assembled, or how to take care of it.

As to the spinning of balls in races, I do not quite know what Mr. Rodway means by this. There is an appreciable amount of sliding of one surface over another in all bearings, this sliding is least when the axis on which each ball rotates is parallel to the axis of the bearing, but though appreciable it is negligible, or else thrust bearings where the slipping is greatest would not stay up.

It is sometimes urged that in a cup and cone bearing of the two point type there is a tendency for the balls to travel outwards, and thus very materially increase the pressures and break up the bearing. I do not think this is correct, because such good results have been obtained with them. Also I doubt if the tendency exists at all. It is difficult to try, but an analogous experiment —a board revolving on billiard balls on a table discloses no spreading whatever.

By the way, events are likely to prove Mr. Hall is wrong in saying that the inadequate proportions of the cup and cone bearings employed in

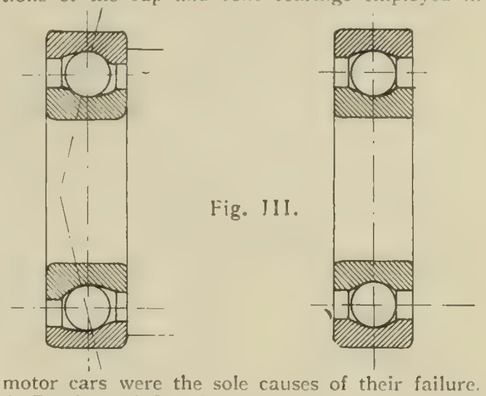


Fig. III.

motor cars were the sole causes of their failure. A Continental firm is beginning to use cup and cone bearings, the dimensions of which are smaller than some types of cup and cone bearings that to my knowledge failed. These new ones are being better protected from water and better lubricated, so that my contention is at least shared by this Continental firm, viz., that keeping the bearings absolutely full of a suitable lubricant and keeping the water out are of more importance with any reasonable design than all the other considerations put together.

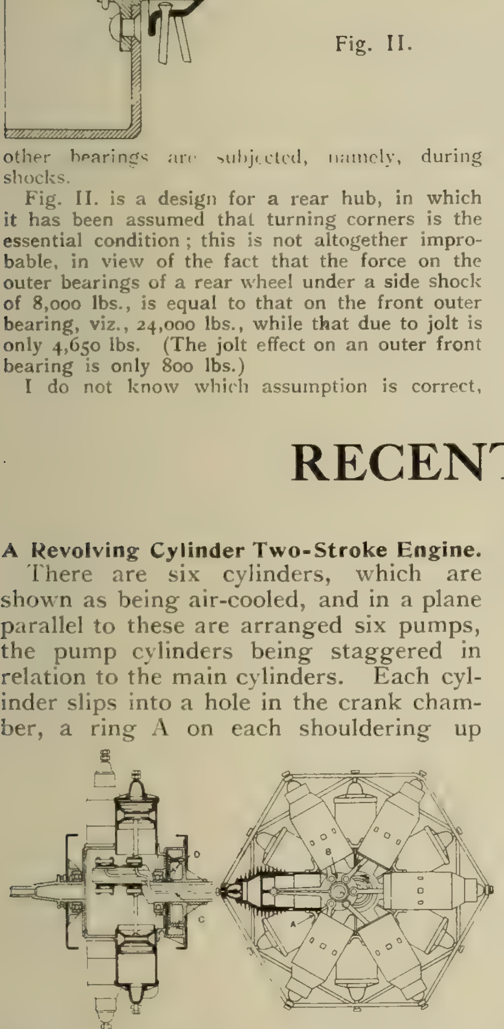
JOHN V. PUGH.

RECENT AUTOMOBILE PATENTS.

By Eric W. Walford, F.C.I.P.A.

A Revolving Cylinder Two-Stroke Engine.

There are six cylinders, which are shown as being air-cooled, and in a plane parallel to these are arranged six pumps, the pump cylinders being staggered in relation to the main cylinders. Each cylinder slips into a hole in the crank chamber, a ring A on each shouldering up

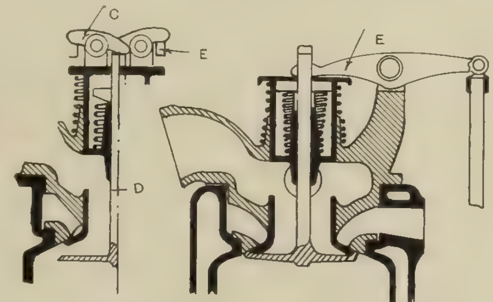


against a suitable face. Around both the working cylinders and the pump cylinders is a shrouding ring carrying set pins screwing down on the cylinder heads, or rather, on the inlet valve domes. These domes merely slip into place, and are held down by the pressure of the set pins, which also retain the cylinder in the crank chamber. Thus, by undoing a set pin the dome and cylinder can be removed. The engine is provided with a master connecting rod, to which the other connecting rods are attached, as in the Gnome engine. The big ends are lubricated by a forced lubrication supply, by means of the tube C. Mixture enters a supply chamber D, which has a spring-controlled plate valve working up against the rotating crank chamber face, which is provided with ports adapted to put the pump cylinders into communication with the supply chamber D. These are provided with a large automatic valve, through which the gas is supplied to the working cylinders.

H. R. Ricardo. No. 26,563/09.

A Concentric Valve System.

The direction of flow of the gases is here perfectly obvious, and it will be understood that the central valve is the inlet, and the outer one the exhaust. The feature lies in the method of operation by a single rocker and cam. The valve rod is held down on a peculiarly shaped cam, by means of a spring, which is stiffer than the springs mounted on the valves. The valve rocker is provided with a forked end E, which lies beneath short rocking levers C, the other ends of which act on the inlet valve stem D. As the valve rod is raised the forked lever end E descends, pushing down the exhaust valve, which incidentally carries with it the inlet valve, as is usual. Movement of

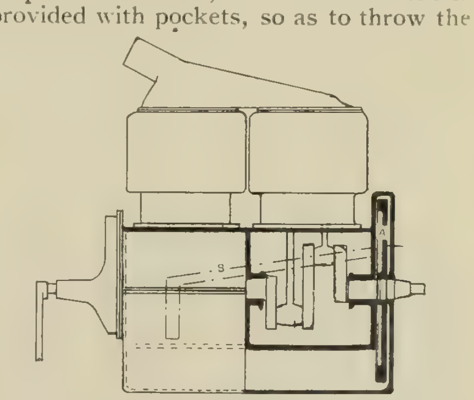


the rocking lever in the opposite direction raises the ends E, allowing the exhaust valve to rise under the action of its spring. Further movement brings the rocking lever C into operation, pushing down the inlet valve.

Société Anonyme des Anciens Etablissements Panhard and Levassor. No. 14,278/10.

Engine Lubrication.

The crankshaft is provided with a disc or paddle wheel A, the rim of which is provided with pockets, so as to throw the



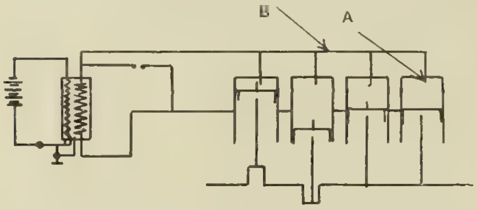
oil lifted up from the sump into collectors, from which it flows down inclined channels B into the crank troughs or other parts. This constitutes an ingenious form of pump having no delicate moving parts, and it is difficult to see how it could fail in its operation as long as there is any oil in the oil sump.

W. Hall, and D. W. Illius. No. 19,614/10.

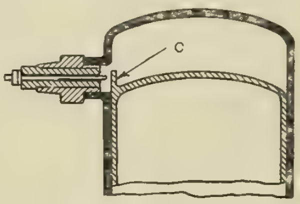
A Simple Ignition System.

By this invention the use of a high tension distributor, or a number of coils can be dispensed with in engines in which no two cranks occupy at the same time the same rotative position. The construction illustrated applies therefore to a four-cylinder two-stroke engine. In the primary

system a contact breaker is arranged, and the single secondary system is connected to four electrodes A, which project into the cylinder, and lie in proximity to earthing electrodes B, mounted upon the piston.



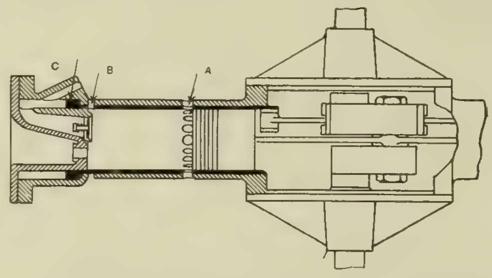
As only one crank can be at its highest position at any one time, it follows that on completion of the primary circuit, sparking will take place between the earthing electrode on the corresponding highest piston, and the fixed electrode A. The sectional drawing shows a practical construction for carrying out the invention, the piston being provided with a lip, C, which passes across the front of an earthed high-tension terminal or sparking plug. It will be gathered that the lip comes into the sparking position before the contact breaker is closed, and, as the lip is of fair length, it provides a certain latitude for variation of timing by the contact breaker. Although this is quite an unusual type of ignition gear, it seems that it might be useful for some purposes.



D. Maggiora. No. 28,378/09.

A Revolving Cylinder Engine.

In this construction the movement of the cylinders through the air is utilised to obtain complete scavenging of the exhaust gases. Located between the piston and the cylinder is a sleeve having exhaust ports at A registering with cylinder ports and additional ports at B, so that on the exhaust stroke not only are the high pressure gases free to exhaust by the outlets A, but air is forced in through the openings B, owing to the movement of the cylinders around the crankshaft, this additional air serving to scavenge the cylinders. The latter are provided with scoops or funnels, facing in the right direction, and adapted to enhance the scavenging draught of air. The sleeve is controlled by an eccentric from the crankshaft, and is formed with a thickened terminal ring portion C, constructed so as to slide in an

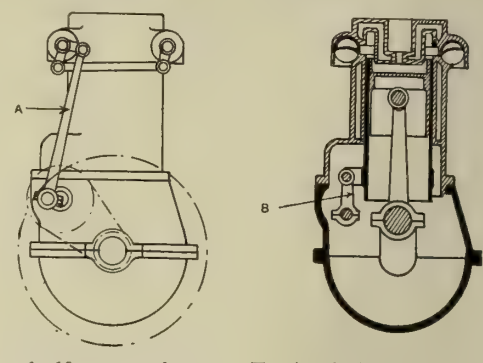


annular chamber and operate as a force pump, drawing in carburetted air through the inlet passage, and forcing it through the automatic inlet valve shown into the cylinders.

F. E. Roberts, H. Morten, and H. E. Ripley. No. 25,925/09.

A Valve System.

This invention refers to a means of cutting off the actual valves from the cylinder during the periods of high pressure. The actual valves used are oscillating ones, of



a half-moon shape. Each of these is free to oscillate in its own chamber, the two being connected together by a link, so that both move in unison under the control of a connecting rod A, driven from a crank on the half-time shaft. It will be gathered that when the connecting link moves to the right, the left-hand valve is opened, and the right-hand one cut off, and vice versa. Sliding between the piston and the cylinder is a portless sleeve, which is controlled by a connecting rod B on the half time shaft. This is adapted, during the compression and firing strokes, to be raised, and so cut off the valve chambers, and also complete the shape of the combustion chamber. This engine is interesting by comparison with the Henriod, mentioned on page 231.

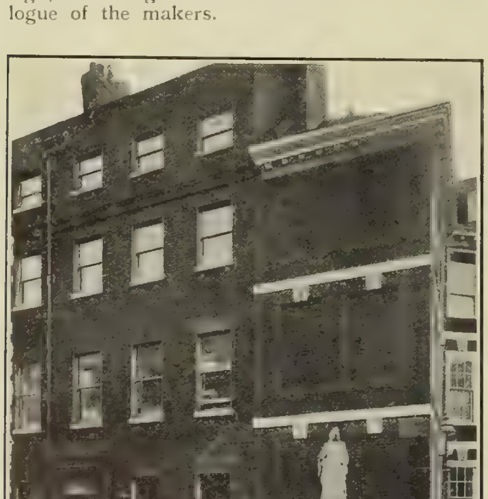
D. J. McKinnon. No. 26,576/09.

MISCELLANEOUS.

CATALOGUES RECEIVED.

TIME RECORDERS.—The International Time Recording Co. have recently issued a small booklet descriptive of their time recording apparatus for works equipment. It is thoroughly well illustrated, and shows many different applications of the devices.

BALL CLUTCHES.—A new type of ball clutch is being made by Ballclutch, Ltd., and is obtainable in many different sizes. The clutches are generally of small dimensions by comparison with other types of couplings and are of simple design, according to the information in the catalogue of the makers.



The recently-acquired Headquarters of the Institution of Automobile Engineers, Queen Anne's Gate, Westminster.

STEEL WHEELS.—A new edition of their catalogue has just been published by the Société Anonyme Atlas, and it shows some special light types for cab work.

MAGNETO.—The Mea magneto—a compact machine for which a high efficiency is claimed—is fully described in a leaflet recently published by United Motor Industries, Ltd.

CRANKSHAFTS.—A most extensively illustrated list of automobile crankshafts has just been issued by Ambrose Shardlow and Co., Ltd., through their agents, Messrs. Barron and Bithell. Other shafts, axles and various parts are also included in what should be a useful catalogue.

COMPRESSED AIR TANKS.—Mr. M. Scott Robinson has sent us a descriptive leaflet dealing with his reservoirs for compressed air. These tanks are being used for supplying tyres in some cab garages, and for compressed air for many other purposes. Small compressors of a simple and inexpensive type are also listed.

GRINDING MACHINERY.—One of the most notable firms now manufacturing tools for precision grinding is undoubtedly the Norton Grinding Co., who have recently appointed Alfred Herbert, Ltd., as their agents in this country. From the latter firm we have received a complete catalogue of the Norton machines, showing their application to many different varieties of work. As a whole the book is an unusually interesting production, and contains a considerable amount of truly useful information.

CHASSIS PARTS.—Malicet et Blin have issued a catalogue for next year which is exceptionally large, and so well illustrated that it shows the details of the majority of the very numerous parts which are manufactured by this firm. At the end of the book there is a large folded sheet bearing curves for finding gear proportions. Thus if a particular diameter of gear wheel and a particular pitch are desired a knowledge of the horse power to be transmitted and the speed of revolution enables the stress on the material and the ideal width of face for any given steel to be found instantly. The curves can be used in any way for the discovery of any proportion where the remainder are known, and it should be very useful to designers.

REVIEWS.

Automobile Law is the title of a handbook recently published, and the aim of the authors has obviously been to provide as nearly as is possible a plain and direct statement of the law affecting motor vehicles. It is almost entirely free from legal language, and the numerous ambiguous points in the law are explained on a basis of probability. With motor law given to us as at present, in instalments, any attempt at unifying it is well worth attention, and this the present volume has on the whole succeeded in doing. A useful portion of the volume is a table giving the amount of the tax payable on any size of car, from the smallest single-cylinder to the largest six-cylinder. (Iliffe and Sons, Ltd., 6d.)

The Care and Management of Accumulators, by Harold H. O. Cross, is an elementary handbook which nevertheless contains numerous useful hints for those who, with but small knowledge of electricity, have occasion to employ small storage batteries for lighting or other purposes. The expression is nearly always clear and the explanations simple, so the book should answer its purpose quite successfully. (E. and F. Spon, Ltd., 1s. 6d.)

CHAS. CHURCHILL AND CO., LTD., have been appointed sole agents in the United Kingdom for the "Alundum" grinding wheels, made by the Norton Co.

WEBSTER AND BENNET, LTD., have appointed the Judson-Jackson Co. to represent them in the South-Eastern counties and in London.

A NEW RADIAL drilling machine has recently been put on the market by Alfred Herbert, Ltd. The gear-box gives eight speeds and an additional two-speed countershaft can also be fitted if desired. Ball bearings are used to support the arm, which is not adjustable for height, an extra stiff spindle being employed. Three speed changes are provided by the feed box, giving sixty, ninety or a hundred and twenty cuts per inch.

THE SUNBEAM ENGINE described in our Annual issue was wrongly stated to have a water circulating pump of the gear type, whereas it is actually centrifugal, this being a more suitable pattern for the purpose.

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CONTENTS.

THE REDUCTION OF WEIGHT.

It is proverbial that the British engineer should consider all American machinery to be too light, and that his transatlantic colleague should think that "British" and "weighty" are interchangeable adjectives. In many ways there is reason on both sides, though no doubt it would be easy to quote many special instances where matters are the other way about. Up to the present European automobile manufacturers have found but little in American design that was noticeably an improvement on their own practice, especially when the difference in the conditions of use are taken into account, but there is certainty that the time is near at hand when Europe must look to her laurels, and the direction which the first conspicuous American lead will most probably take, if indeed, it has not done so already, is in the reduction of chassis weight. Weight is of far greater importance in America than it is in this country or in France or Germany, but there are large tracts of the Continent where the road conditions are quite as bad as those which prevail in the United States and Canada. In every country, on every sort of road, light weight is valuable, for it means low fuel consumption and light tyre wear, but on rough roads the work which has to be performed in the absorption of shocks is almost directly proportional to the inertia of the chassis, and the stresses set up by road shocks also depend upon the weight. Thus, where bad roads are common, the elimination of all unnecessary weight is of particular importance. Weight must, of course, always be comparative only with engine power and carrying capacity, and so American chassis may be either called powerful and large for their weight, or light for their size and power, and it is perhaps interesting to reflect upon the reasons which have caused the difference between American and European practice.

Throughout Europe there are supposed to be good roads (the roads of England, France and Germany are really good), but throughout America and most of the British colonies the roads are extremely poor, except in the urban areas. For this reason, in England, France and Germany, the average speed of cars travelling on the main roads is much higher than it is anywhere else, and it is essential that a chassis should be able to develop its full horse-power continuously for long periods of time. This means that there must be ample bearing surface in the engine and transmission, and that the springing must be adapted to absorb the type of shock caused by a small obstacle struck at high speed. The American car needs high power to enable it to overcome such difficult road conditions as ten-inch mud, fords, loose boulders and heavy gradients, but owing to these road conditions the American car is very seldom driven at high speed for more than a few moments. This means that to give equal durability in America to a European car in Europe, there is not the same need for large bearings, wide gears, etc., while absence of chassis *rigidity* is not only an advantage, but almost an essential. It is common knowledge that in recent American trials and races for stock cars—or standard chassis—there were many failures owing to bearings seizing or giving way altogether. Still, if the "working" parts of American cars do err on the light side for European use, there is no doubt that European cars err on the heavy side in the weight of parts other than bearing surfaces, though it must be remembered that American chassis are seldom overloaded with the heavy bodies which are common here.

It should be possible to obtain the advantages of both schools of design, the American ought to gain by a slight increase of solidity, and the European by a slight decrease of precisely the same quality, but neither have so far taken much advantage of their numerous opportunities for detail weight saving. Cylinder bore, or cylinder capacity has so long been the only form of rating basis for all kinds of competition that the possibility of cutting down weight has not received much consideration—indeed, it has received much actual discouragement. The removal of redundant parts, and the simplification of those remaining has been in progress for some years, but it has been very slow and, to an extent, accidental. Only a very few makers have expended much thought upon weight. The parts which should be attacked first are not the main castings, the main frame or the shafts, but fittings, such as carburettors, silencers, piping, tanks, control levers, some frame parts, step irons, wing irons, bonnets and, of course, accessories. By saving a pound or two here and an ounce or two there the total weight might well be rendered less by a hundredweight or more, without any sacrifice of useful strength. Sheet metal might be used much more extensively; many pipes which are now often cast would be quite as serviceable made from drawn tube, or even direct from sheet. Spun sheet brass, or even aluminium, might well replace cast metal for such parts as the float chamber, the oil filler lid and the radiator cap, while stamped sheet would form equally effective material for clutches, crankcase inspection hole covers, valve cover plates, gearbox lids and numerous small brackets and connections on the control, even

if it could not be employed usefully for larger parts, such as the lower half of the crankcase.

In very many instances more or less standard chassis have been lightened for racing purposes by the drilling of holes in almost every part. It seems that in very few cases was the weight reduced too much, and if a chassis will withstand racing stresses it should be quite sufficiently strong for ordinary work. Obviously, if numerous holes can be drilled in a piece without making it too weak, that part could normally be made less weighty by being thinner and free from drilling; yet this fact seems seldom to have received the amount of attention which it deserves. However, as has already been suggested, it is not the parts which require strength that are usually too heavy, so much as those which require little or none, such as lubricators, lamp brackets, pedal plates, silencer parts, and covers of all kinds, while even brake or change speed levers, gates, spring pads, dumb irons and back axle brake supporting arms are often quite needlessly solid.

SPIRAL GEARING CALCULATIONS.

By J. L. Milligan.

It is common knowledge that the use of spiral and worm gears in automobile work is increasing, for these are now largely used for fan, pump, magneto, and camshaft drives, as well as for the final drive on the back axle. Yet there are few text books dealing with the subject, and none of the usual pocket books give any help to the designer in laying out spiral gears. One of the chief difficulties encountered is the calculation of correct gears to work in between shafts, the relative positions of which are fixed. And the following method has been used by the writer in laying out such gears.

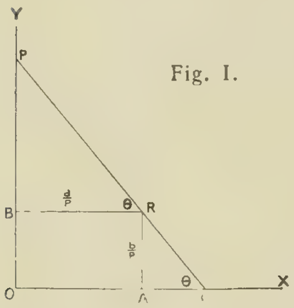


Fig. I.

If a and b are the numbers of teeth in a pair of spiral gears, P the diametral normal pitch, θ the angle of the teeth of "a" to the axis; ϕ the angle of the teeth of "b" to the axis:

Then by the usual formula.

$$\text{Diameter of pitch line of a} = \frac{a}{P\cos\theta}.$$

$$\text{Diameter of pitch line of b} = \frac{b}{P\cos\phi}.$$

If the shafts are at right angles, $\cos\phi = \sin\theta$ and the distance between the shafts is half the sum of the pitch diameters.

$$D = \frac{1}{2}\left\{\frac{a}{P\cos\theta} + \frac{b}{P\sin\theta}\right\} \quad \text{......................(1)}$$

for any given value of a, b, and P. D will vary from infinity, when $\theta = 0°$, through a minimum value and up to infinity, when $\theta = 90°$. By differentiating we get

$$\frac{dD}{d\theta} = \frac{1}{2P}\left(\frac{a\sin\theta}{\cos^2\theta} - \frac{b\cos\theta}{\sin^2\theta}\right)$$

and equating this to zero we find

$$a\sin^3\theta = b\cos^3\theta.$$

$$\tan\theta = \sqrt[3]{\frac{b}{a}} \quad \text{.............................. (2)}$$

This gives us the angle which will give the minimum distance between centres for any given ratio, if the pitch is constant. Or conversely, will give the largest pitch, if the ratio and number of teeth is constant.

As the distance D passes from infinity to a minimum and back to infinity again it follows that if D is not the minimum value there are two values of θ for each value of D, so that every spiral gear has a complimentary gear with the same numbers of teeth, the same pitch, and the same shaft centres, but with a different angle of teeth and diameter of gears. The two values of lie on opposite sides of the angle for minimum distance (2).

There is no simple solution to the equation (1), except the graphic method which is founded on the following theorem, in which the case is at first confined to instances where the axes of the gears are at right angles.

OX and OY in Fig. I. are drawn at right angles, and points A and B are marked, so that $OA = \frac{a}{P}$ and $OB = \frac{b}{P}$

If a straight line be drawn through R to meet OX and OY in Q and P then $PR = \frac{a}{P\cos\theta}$ and $RQ = \frac{b}{P\sin\theta}$

Whence

$$PQ = \frac{a}{P\cos\theta} + \frac{b}{P\sin\theta} = 2D.$$

The application is as follows:—Suppose a gear ratio of two to one is required between two shafts, three inches apart, using 8P diametral normal pitch. Then, referring to Fig. II., draw OX and OY at right angles, and draw OC, $\tan COX = \frac{b}{a} = \frac{1}{2}$. Along OY set off spaces $= \frac{1}{P} = \frac{1}{8}$in. apart. Draw lines at right angles to OY to cut OC. Draw a line = 2D = 6in. long on a strip of celluloid or marked off on a straight edge, and slide the ends of the line on OX and OY until the line passes through the intersection of OC and one of the parallel lines. There will be two lines UV through each intersection from O up to the limiting number of teeth, which in this case is twelve. The distance QV is the diameter of wheel b, the distance QU is the diameter of wheel a, and θ is the angle of teeth on wheel a.

The number of teeth in b is the number of pitches up to the point of intersection. The most suitable gear for the circumstances can be selected by inspection of its angle and diameters without any calculation.

If the diagram is made to a sufficiently large scale, the angles will be within 20′ of the correct size, thus reducing the field

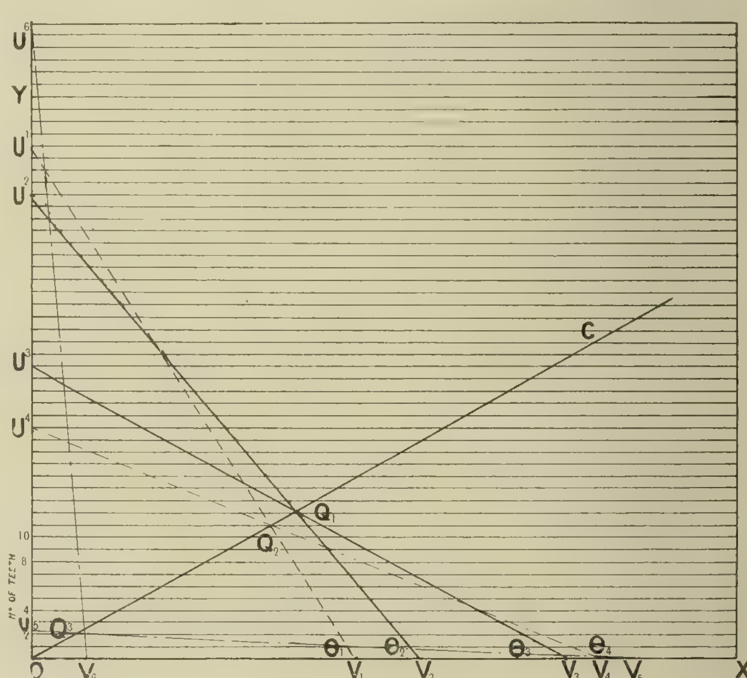


Fig. II.

when the final calculation is required for detailing the gears.

If the module is used instead of $\frac{1}{P}$ the results will be in millimetres.

If the worm is to be cut in a lathe and has a pitch of L inches, then the line UV must be marked off with distances $= \frac{L}{\pi}$ measured from V, if a is the worm, and from U if b is the worm, the lines being drawn when U and V are on the lines OX, OY, and one of the divisions of UV is on OC.

GYROSCOPIC ACTION IN RELATION TO MOTOR VEHICLES.

By H. C. Harrison.

UNDER ordinary running conditions of a motor vehicle there are certain forces called into action which do not originate in the development of power and its transmission to the road wheels. Among them are the forces arising from gyroscopic action. In view of the uncertainty with which this action is not infrequently regarded, the subject is approached in the following treatment from a consideration of the more elementary dynamical principles.

The simplest form of motion a body can possess is motion in a straight line at a perfectly uniform velocity. It then either has no forces acting upon it, or else all the forces acting counterbalance one another, so that their resultant is zero. If the motion is not uniform there must be an unbalanced force producing acceleration or retardation, and, as expressed by Newton's third law, the rate of change of momentum is a measure of the force. Thus, if m is the mass of a body, and in one second its velocity changes from v_1 to v_2 the force F acting is given by $F = mv_1 - mv_2$, or, since change of velocity per second is acceleration, a,

$$F = m\ a. \qquad (1)$$

A precisely similar condition holds in the case of a body rotating about a fixed axis. If I is the moment of inertia of a rotating wheel and the speed of rotation changes, in one second, from ω_1 radians per second to ω_2 radians per second, the torque T producing the change is given by $T = I\omega_1 - I\omega_2$, or calling the angular acceleration, α,

$$T = I.\ \alpha. \qquad (2)$$

In (1) it was assumed that F acts in the line of motion of the body, and in (2) that T acts about the axis of rotation, but treating force and torque as vector quantities, these equations are true if F is inclined to the direction of motion, and if T acts about an axis other than the axis of rotation.

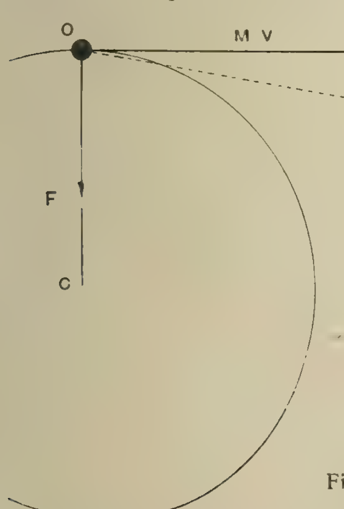


Fig. I.

The gyroscopic effect of a rotating body may best be considered by comparison with the somewhat simpler motion of a body rotating in a circle under the action of centrifugal force.

In Fig. I. a mass m at O is rotating uniformly in a circle with angular velocity a about C. Let v be the linear velocity, and let OA represent to scale its momentum mv. At the end of an indefinitely small increment of time δt sec. the velocity will have changed in direction, and the momentum is now mv in direction OB, making an indefinitely small angle δa with OA. Hence the momentum has changed by amount AB in the time δt. Since δa is indefinitely small, we may write—

AB = OA × (angle AOB in radians),

or, change in momentum in time δt, $= mv \times \delta a$, or, change in momentum per second $= mv \times \dfrac{\delta a}{\delta t}$ and from (1)

$$F = mv\ a^* \qquad (3)$$

Also since AB is indefinitely small, it may be taken as perpendicular to OA, so that F acts along OC on the mass at O. Observe that the change in direction of OA is such that it *tends* to become parallel to the direction of F.

The precession of a spinning body may be considered in a similar manner:—

Let I be the moment of inertia of a spinning wheel, ω radians per second its speed of rotation, and ϕ radians per second its speed of precession. In Fig. II. OA is the direction of the axis of spin, and its length represents to scale the angular momentum $I\omega$ possessed by the wheel. As in the case of centrifugal force, let OA precess to the position OB, making an indefinitely small angle $\delta\phi$ with OA, in the indefinitely small time δt secs. Then AB is the change in angular momentum in the time δt, but

AB = OA × (angle AOB in radians),

or, change in angular momentum in time $\delta t = I\omega \times \delta\phi$, or, change in angular momentum per sec. $= I\omega \dfrac{\delta\phi}{\delta t}$, and from (2)

$$T = I\omega\ \phi \qquad (4)$$

This torque acts along AB, which is perpendicular to OA when the increments are indefinitely small. Hence, if the axis OA of a spinning wheel precesses as shown, it must be acted upon by a torque T, whose axis OC is perpendicular to OA. Again observe that the direction of precession is such that OA tends to set itself parallel to OC.

It is to be noticed that the direction of spin may be clockwise or counter-clockwise; suppose that looking from O along OA the rotation is clockwise, then looking from O along OT, the direction of the torque causing precession is also clockwise. The wheel reacting against this imposed torque gives out a counter-clockwise torque. The diagram, Fig. II., together with (4),

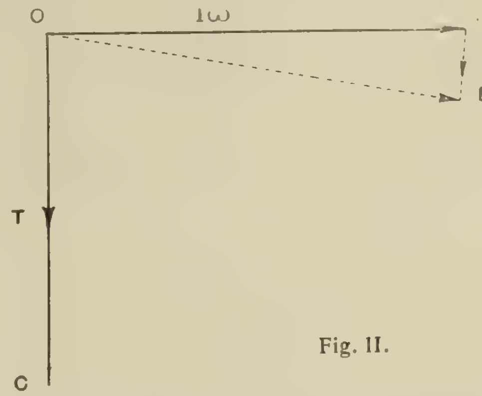


Fig. II.

enables the performance of a gyroscope to be completely stated, both as regards the magnitude and direction of the torque and the speed and direction of the ensuing precession.

Proceeding to consider the normal type of motor vehicle, it is found that gyroscopic action is exhibited to a very limited extent. It is obvious that the only parts which can give rise to such action are those possessed of rapid rotation, and of considerable inertia. The further condition must also hold, that the axis of rotation shall be subject to precession. The flywheel and road wheels, more particularly, comply with these conditions. Precession occurs in both instances when negotiating curves and a corresponding torque is produced.

Taking first the case of the flywheel, Fig. III. is a diagrammatic plan of a car. OA is the axis of the flywheel. Suppose the direction of rotation to be clockwise looking from O to A. Then if the car be steering to the right OA is undergoing precession, as indicated by OB. Since OA always tends to set itself parallel to the axis of the torque causing precession, this must be in direction OC, and be clockwise looking from O to C. The reactionary torque exerted by the flywheel on the car is opposite to this or counter-clockwise. The effect is obviously to decrease the load on the front springs, and increase that on the rear springs. It is of interest to estimate the possible variation in loading in a particular case.

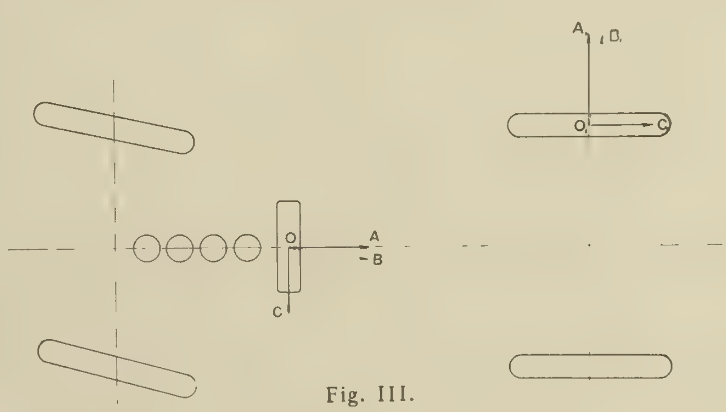


Fig. III.

*The more usual expression for centrifugal force is $F = \dfrac{mv^2}{r}$, but the above form is more suitable in drawing comparison with gyroscopic action.

Assume the weight of the flywheel to be 110 lbs., and its effective radius 1ft., so that its moment of inertia is $\frac{110}{32} \times 1^2 = 3.44$ lb. ft.^{2}; also assume a curve of 100ft. radius to be negotiated at a speed of 30 miles per hour (44ft. per sec.) with engine speed 20 revs. per sec. or $20 \times 2\pi = 125.6$ radians per sec. The angular velocity of the car about the centre of the curve is then $\frac{44}{100} = .44$ radians per sec., so that (4) gives for the value of the torque

$$T = 3.44 \times 125.6 \times .44 = 190.1 \text{ lb. ft.}$$

With a wheelbase of 10ft. the force decreasing the load on the front springs and increasing that on the rear springs is $\frac{190}{10} = 19$ lbs.

Treating the road wheels in the same way, we see that as the car negotiates a right-hand curve the axis of rotation OA precesses as indicated by OB, and hence the wheel must be acted upon by the torque, whose axis is OC, and direction counter-clockwise looking from O_1 to C_1. The reactionary torque exerted by the wheel is therefore clockwise, and acts in conjunction with centrifugal force in its tendency to upset the car.

Taking the weight of wheel and tyre as 45 lbs. at an effective radius of 1.25ft. (other data as before), the moment of inertia is $\frac{45}{32} \times 1.25^2 = 2.2$ lb. ft.^{2} The circumference of the tyre will be approximately 9ft., and the speed of rotation $\frac{44}{9} = 4.9$ revs. per sec. or $4.9 \times 2\pi = 30.7$ radians per sec. Hence the torque exerted by the four wheels is $4 \times 2.2 \times 30.7 \times .44 = 119$ lb. ft. (The inclination of the front wheels to the longitudinal axis of the car is neglected.) The torque produced by centrifugal force under these conditions on a car weighing one ton and having its centre of gravity 2.5ft. above the road surface is $\frac{2240}{32} \times \frac{44^2}{100} \times 2.5 = 3,387$ lb. ft.

These two examples serve to illustrate the extent to which gyroscopic forces may exist in a car. It is seen that they may be of quite appreciable magnitude, although their effect is masked by forces of a higher order. In the case of an engine set with its crankshaft transversely to the longitudinal axis of the car, the flywheel exerts a torque in the same direction as, or opposite to, that produced by the road wheels, according as the rotation is in the same or opposite direction.

AN INTERESTING AGRICULTURAL TRACTOR.

THE design and construction of an agricultural tractor is one of the hardest problems connected with petrol-engined vehicles, as a machine has to be evolved which will have to accomplish so many tasks, and must perform each of these several duties in a perfectly satisfactory manner. Not only does this difficulty present itself, but the tractor must be prepared to work in various climates, each with its own peculiarities and under conditions of the utmost difficulty. Moreover, it will always be in the hands of men who are not skilled mechanics, and who are likely to go on running the machine as long as there is the slightest possibility of doing so, whether the machine is in a condition to run or no. Then, again, there is a certain convention in design, even in colouring, which must, in this country at any rate, be fulfilled, in order to make the machine a commercial success, whilst, above all things, it must be cheap to run. Therefore it will be seen that for a machine to be successful it must overcome difficulties which at first sight are not apparent. For this reason the 50 h.p. Saunderson tractor is interesting. It is an attempt to design a high powered machine which can be handled by unskilled drivers, and which shall at the same time be exceedingly cheap both to run and to repair. With this end in view the designers decided upon a paraffin engine, of such a power that gear changing should be rendered practically unnecessary, and fitted with as few bearings as possible which would be likely to need adjustment. The machine is provided with attachments which render its uses manifold, having a drum placed at the rear of the gearbox for driving stationary machinery, while the spring draw-bar at the back is fitted with attachment holes corresponding to the various implements likely to be used, and with due regard to their relative position to the track. Accessibility has also been studied, there being no mechanism which cannot be got at with ease.

The engine has four cylinders of 6in. bore and 8in. stroke, with a normal speed of revolution of 800, and these cylinders are cast in pairs, having very large water jackets. Both halves of the crankcase are of cast iron, the bottom half having a gutter which serves to catch and retain anything which may be dropped into the case, and also serves as a guide to the oil drain taps. The top half is provided with separate chambers at each side for the valve gear, and these have inspection covers running their entire length, the camshaft being so designed that it can be removed easily by the unscrewing of six nuts, allowing adjustments to be made to the big end bearings. Each camshaft has three phosphor bronze bearings and is driven by steel spur gearing from the forward end of the crankshaft, which has three gun-metal bearings and is cut out from a solid forging. Splash lubrication is relied on entirely, and is effected by a small hand pump of motor-cycle type, which fills the base-chamber from an oil tank placed, with the pump, in a convenient situation for the driver. Although an arrangement which tends towards a perpetually smoky exhaust, it is the simplest form possible, while in actual practice it has been found easier to instruct a man to give one charge of the pump per half hour than to get him to attend to lubricators of the sight feed or mechanical pump variety; moreover, the engine speed being low renders this form more likely to succeed. Another important part of the engine is the governor, which operates the throttle direct to an extent which is controlled by the setting of a small milled disc. This governor is driven by bevel gearing off the camshaft at half the engine speed as the contact maker for battery ignition is placed above it on the same shaft. A magneto is fitted, which can be controlled, together with the throttle and battery ignition, by a lever placed on top of the cylinders at the driver's left hand. The valves are of the ordinary mushroom type, but an unusual feature in connection therewith is the fitting of a metal plate between the exhaust valve springs and the exhaust pipe in order to prevent the heat of the latter destroying the tension of the springs.

The paraffin carburettor seems to be the weakest point in the design, being perfectly satisfactory on petrol, but giving an unsatisfactory mixture when fed with the heavier fuel, and it is far too dependent on jet adjustment to be ideal for the work it is designed to perform. This jet adjustment, being effected by a milled disc and lock-nut, is rather a doubtful advantage in the hands of the unskilled but enthusiastic driver. Moreover, very little movement seems to affect the mixture considerably, and the proper setting is a very delicate operation. Much of the fuel vaporisation takes place in the inlet pipe, round which a portion of the exhaust gases are carried on their way to the open air, thus giving the engine, in reality, two exhaust pipes.

It seems that the running could be improved greatly by some alteration to the vaporiser, because, judging by the recent R.A.S.E. trials, the engine does not seem able to run reliably at slow, or even moderate, speeds except on petrol, supplied from an auxiliary tank, for starting. Petrol is cut off and paraffin admitted, after some ten or fifteen minutes' running, by means of a two-way tap. A large paddle pump driven from the camshaft delivers water through iron pipes to the cylinders, care being taken to enlarge the inlets successively, as it has been found that without this feature circulation was confined to the forward pair of cylinders, leaving the rear pair to act as boilers in miniature. From the cylinders water is returned to the radiator by a rather small steel pipe, the radiator being a Reliance type, mounted on a cast iron tank of large size. Behind the radiator on a massive bracket is a belt-driven fan having its spindle so mounted in a groove that adjustment is very readily effected. Altogether the cooling is most effectively carried out and should give no trouble, although it would seem probable that a considerably larger inlet pipe would be a genuine advantage when tropical conditions, and the fact that the machine will be used as a stationary engine, have to be considered. Certainly, for use in this country the present system seems efficient, as at no time does the engine show signs of undue heating. The main exhaust, as distinct from that let out through the inlet pipe, is taken to a large

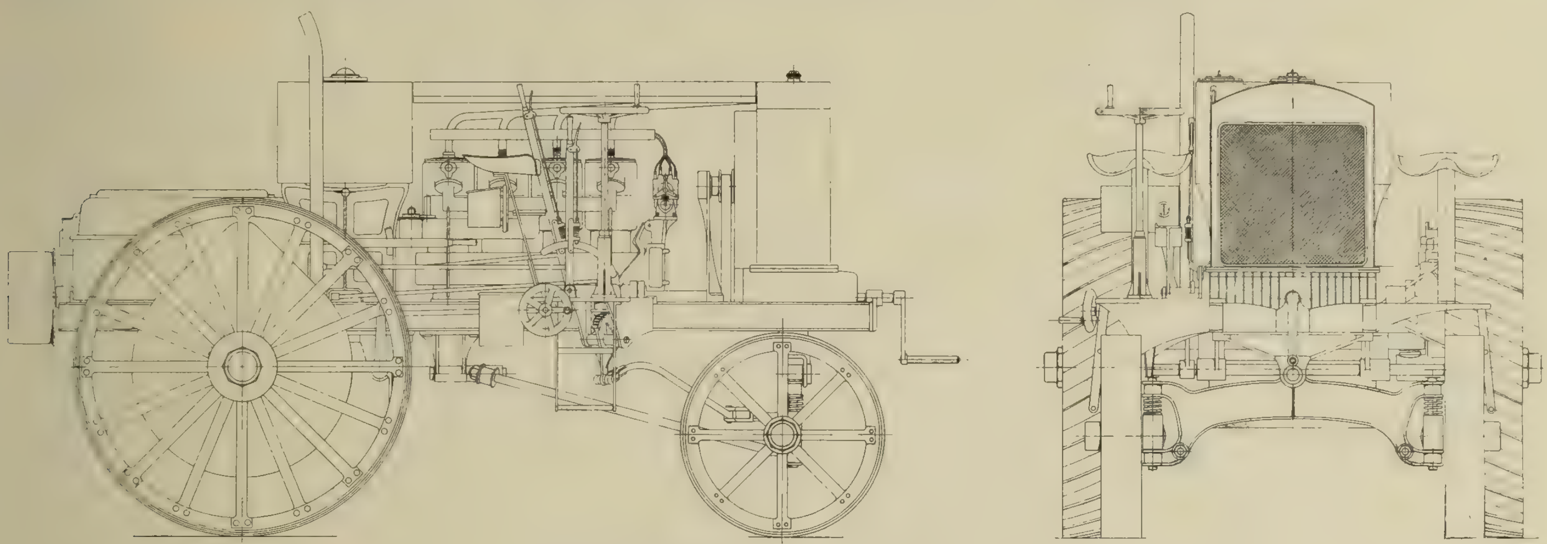
silencer placed transversely underneath the car, and from there to the air up a long vertical pipe, ending some distance above the driver's head. It has been found necessary to fit the silencer with an oil guard of metal to prevent fumes being caused by leaking oil coming from the engine, or careless oiling of the tail bearing.

From the engine the drive is taken by a large leather-to-metal clutch running on a spigot from the crankshaft, which spigot is provided with a very noticeable lubricator, thus minimising the risk of under-lubrication. Otherwise the clutch possesses no special features of interest. The drive is taken to the cast-iron gearbox (which latter is placed above and behind the rear axle), and passes through sliding spur gears of large diameter and coarse pitch, giving speeds of four, six and eight miles an hour at normal engine speed. No particular trouble is taken to render these gears silent, as this would be unnecessary. The gear change is made by a lever on the driver's right, moving over a plain notched quadrant. The bevel is at the forward end of the box above the primary shaft and on the differential box, whence the drive is taken by two countershafts ending in spur pinions meshing with gears on the road wheels, these gears being encased in a metal cover provided with sliding panels for lubricating. The ball thrust bearing for the bevels is the only ball bearing on the machine. The differential can be locked at will by means of a pin, from the driver's seat, an absolutely necessary fitment when the probable conditions under which the vehicle will work are considered. The gear shafts are long and provided with three bearings, with the exception of which the box is grease-lubricated, outside lubrication being provided for each end bearing, while the centre bearings are provided with a box lubricator for oil. The box inside the gear-case seems a weak point, as it can be very easily overlooked, and it would appear to be easy to provide an outside lubricator similar to those fitted to the other gear-box bearings and those used for the crankshaft bearings. This seems more necessary because the heating up of this bearing can quite easily occur without the driver noticing it. There are two brakes, the chief of which is foot-operated acting on the differential, and is of very simple design, the other being hand-operated and consisting of two small iron shoes which are applied to the inner rims of the steel rear wheels.

The wheels are of the type usually connected with traction engine practice, having flat spokes riveted in pockets and the outside circumference provided with diagonal strips of metal fixed by well countersunk. rivets. Very deep channels form the frame and cross members, and the rear cross member is fitted with a draw-bar, through which the pull is taken by means of long coil springs. To this frame the front axle is attached in a very notable and effective manner, being hinged at the centre and provided with long radius rods which, while preventing backward motion, allow the axle a very long upward movement on its pivots : this is an excellent feature, as the machine will surmount extremely rough or even rocky ground. Each steering pivot pin is mounted on a small coil spring and provided with mud shields where sliding motion takes place, the same shields being placed over the top of each pin to prevent dirt falling from the front wheels causing undue wear. The rear axle is mounted on a similar type of spring.

The driver, who sits at the right-hand side of the engine, is mounted on a seat bolted to a long flexible leaf spring. Steering is by wheel connected up to a worm and segment having connecting rods to the swivels in the ordinary touring car manner; this is a great improvement on the chain winding mechanism provided on some of the tractors. The tank for petrol and paraffin is provided with very large filling orifices and caps, but it is situated in a somewhat unhappy position directly over the exhaust pipe and silencer, increasing the probability of a fire when filling up, especially as the exhaust pipe is sometimes extremely hot. Although it is hard to suggest an alternative position for the tank, yet some alteration should be made when one considers that the machine is for the careless type of labourer liable to underrate the inflammable properties possessed by either petrol or paraffin, and who would probably allow the entire vehicle to be destroyed before he would attempt to extinguish it. One would be inclined to suggest the removal of the exhaust pipe and silencer to the side of the engine or frame, so that leakage might not come into actual contact with the hot parts.

In actual work a six-furrow plough can be towed on the second speed at a rate of from seven to eight miles an hour, the furrow cut averaging, if necessary, about six inches and the machine having plenty of reserve power, as it can easily start with the blades at their greatest depth, and gradients make but little difference. As already mentioned, the powers possessed of climbing obstacles are extraordinary, and it is possible to tilt the machine over to a very great angle without danger. This is brought home by watching the tractor surmount with ease a large heap consisting of scrap and small rocks of slag, the differential being locked beforehand.



The Saunderson and Gifkins Agricultural Tractor.

Thus taken altogether the design seems to be quite sound except as regards the details, while the success of the machine in service must, of course, depend upon the material and workmanship to a very great extent, and there are scarcely a sufficient number of these tractors in use here as yet to enable the quality of construction to be judged. The method of lubricating the gearbox bearing is weak, and the shafts are on the lengthy side, both points of easy remedy. Possibly it would be better to use sliding dog-clutches instead of sliding gears as this would render a change of gear more easy, and the liability to damage is in favour of the dog-clutch, while the additional expense would be small. Another point which suggests itself, although much akin to a luxury, is the addition of some simple means whereby the take-up of the clutch could be rendered softer and a dash-pot attachment to the clutch pedal would be cheaply and easily fitted. The effect would be to lessen the liability to damage the gears through starting shock caused by unskilful handling. Another smaller point is the fitting of a starting handle of considerably greater length, the present one taking two men to operate, and after a long day's work, starting the following morning would not be easy, paraffin injection notwithstanding. It seems an easy matter to fit a starting handle of a similar type to that used on the Renard road train.

Taken as a whole, the design is a commendable attempt to satisfy the exacting conditions under which a tractor must work, and, with good manufacture, should be most satisfactory.

COSTING AND THE BONUS SYSTEM.

A convenient job costing and recording system for an automobile factory.

UNDOUBTEDLY one of the most important, and most frequently neglected, duties in an automobile manufacturing concern is the installation and maintenance of a proper costing system, whereby an accurate estimate of the labour costs can be arrived at and a permanent record kept for future reference. Many firms do not hesitate to instal high-class tools and machinery in their works, and yet begrudge every penny spent on costing, with the result that they are never sure what their goods cost to manufacture. A good costing system, if properly organised, and supervised by a thoroughly competent man, is bound to be of the utmost benefit to any firm wishing to run their business on modern lines, as not only will it expose leakages which could not be detected by older methods, but it will, by comparisons and analysis, show which departments are actually paying and which are losing money. Also, it is absolutely imperative to know accurately the cost of all the component parts of an engine, gearbox, or any other part likely to want replacing, so as to be able to invoice it to a customer at the correct price and at short notice, otherwise, if guesswork is resorted to, as is often the case, the spare part might actually be sold at a loss when a good profit could have been made had the actual cost been known to base the charges on.

The first item in any costing system is the checking and recording of the workmen's time from the moment they enter the works until they leave at night. As a rule the costing staff receive no assistance from the workmen in this matter, and often very little from the foremen, who usually do not like a strict check to be kept on their men. However, in the majority of modern works nowadays the booking of the time is no longer left to the working operators, nor even foremen, but is designated to a special staff trained for this purpose.

Upon entering the works it is now most usual for men to record their time by mechanical means, and the moment the men enter their respective departments it is the duty of the management to keep them fully employed, and to know how their time is spent. The best method of ascertaining this is by the job-card system, which has superseded the old-fashioned chalked time boards, and has come largely into use during the last few years. Its great advantages are that it is a permanent record of each and every job done by every employee in the works, and thus affords an excellent means of comparison, by analysis and perusal of the various cards as the completed jobs come through the shops. As the majority of shops engaged in the automobile industry offer the workmen a bonus on the time in which a given job is completed, they work on what is commonly called the "Bonus System of Payment," for which the job card, shown in Fig. I., with written entries in italic, was designed. This card has been found to be as nearly "fool proof" as possible, while it contains all the information necessary for accurate costing on any given job. Every man gets one of these job cards on each and every job he completes, and he is booked on by the department job clerk—or by mechanical means in some cases—the minute he starts, and booked off when he has finished. Thus, if a man has ten jobs in one day, he has ten job cards, and those ten cards must contain the whole of the day's working time exactly, which means that there must be no lost time between

10TH SERIES. 4061

THE AUTOMOBILE Co.

JOB CARD. BONUS WORK.

Order No. *151.* Quantity *12.*

NAME *Jones.* Check No. *257.*

OPERATION *Bore*

Connecting Rods

Drawing No. *20.* Piece No. *a.15.*

Machine No. *45.* Wages Rate *35/-*

TIME ON JOB				Daily Quantities.
DATE	ON	OFF	TIME	
15/1/10	*6*	*5.30*	*10*	*3*
16/1/10	*6*	*5.30*	*10*	*3*
17/1/10	*6*	*5.30*	*10*	*4*
18/1/10	*6*	*12.30*	*6*	*2*
TOTAL			*36*	

Good *11*

Scrap *1*

Foreman *J. H*

Inspector *T. J.*

Time Ea. *3 hrs.*

Price Ea. *2/6*

Checked *J. F.*

Wages Recd. *22/-*

Bonus Due *8/-*

Passed *H. J.*

Other Ref. Cards *15/4162 15/4161*

RETURN TO COST DEPT.

Fig. I.

[Italic type represents hand entries.]

booking off one job and on to the next one, or else that time will be lost to the firm. It is advisable to have an independent check on this by deputing a clerk to spend two or three hours each day to check it. Suppose a man is on a job four days, he is simply booked on each morning and off each night until the job is completed, and, as shown in Fig. I., the operation is written on the top together with the quantity, drawing number, piece number, and also the machine number, the latter being very important, as it can then be seen by comparison of various workmen's cards which lathe or machine is the quickest for certain classes of work. In fact, the cards become a record of the efficiency of the machines as well as that of the men. All the information on the top for the man's instructions are filled in by the job clerk, except the wages rate, which is filled up in the time or cost office, where the cards arrive after they have passed through the various shop departments, and have been signed off by the foreman and inspector. The bottom right-hand corner refers to any job on which more than one man has been employed, such as erecting a chassis where a gang of men are often on, and in that case each man's card is cross referenced to the rest, as shown at Fig. I., so as to allow the cost department to collect all the cards together, in order to ascertain the total cost of the job. Each card is numbered from 1 up to 20,000, and they run in series, 20,000 cards making a complete set. When the work shown on these cards is paid for the cost office should file them away numerically in bundles of 1,000, for future reference, as they form an excellent guide for comparing the same job the next time it occurs, and are also useful for estimating purposes.

For work which is not for sale, such as tool and jig making, and other tool-room work, a slightly different form of job card is sometimes used, the wording varying slightly, and the colour of the card being different, so that it may be easily distinguished. These cards also, of course, run in series, and are marked "day work" instead of "bonus work." The number of these cards is entered in an "additions to plant" register, with a description of the job by the tool room foreman, and the book is sent to the cost office each month, whence the total cost of tools completed is easily obtained.

Fig. II. shows a specification sheet which the job clerk has before him when writing out the job cards, a set of these specification sheets containing every item used in every shop order. Thus if an order is given to the shops for a hundred 25 h.p. cars, a set of specification sheets contains every item used, and the clerk enters the number of every job card opposite each item, as shown, so that he can tell at a glance which work is completed and which still requires completing. By this means he can often tell much better than the foreman when there is going to be a shortage of work for the men in his department. When the specification sheets are completed they are sent to the cost office, the junior clerks go down the list, get the job card numbers, and get them out (they have come into the office previously as the jobs have been completed and paid for), they enter the figures in the £ s. d. column, which should be properly ruled, and thus the total cost is obtained. The accuracy depends on the job clerk to a large extent, and he should be a competent man well up to his work, and not too much under the thumb of the shop foreman. He should receive his instructions and be responsible to the shop manager only.

The rate-fixing department is a section that should be in close touch with the cost office, although, unfortunately, they do not very often work harmoniously together, each department not wishing to receive instructions from the other. This common state of affairs is to be deplored, as the two sections can be of real assistance to each other in many instances, the rate-fixing staff being in much closer touch with each individual shop operation and price than the cost office officials. The best method of keeping a permanent record of bonus, or other rates paid on the various operations involved, is the card-index system as shown in Figs. III. to V., and this scheme has almost completely superseded the older methods, such as piecework books, as it is so elastic that it can be adopted to suit any requirement. Fig. III. shows the finger cards or indicators which are marked alphabetically with the names of the various component parts, immediately behind which come the various cards, as shown in front, with the article and operation entries. This is a very concise

THE AUTOMOBILE CO.

SHOP ORDER No. *151.* DESCRIPTION *100 25 H.P.* CARS. DATE *8/7/10.*

MATERIAL							LABOUR			
NAME OF PART	MATERIAL	PIECE NO.	DRAWING NO.	QUANTITY WANTED	PARTICULARS	PRICE EACH	OPERATION	JOB CARD NO.	COST EACH	REMARKS
Connecting Rods	Steel Drop Forgings	A2	G15	400		4/-	*Bore and Face*	*10/2150*	*1/6*	
							Drill	*11/3480*	*-/8*	
							Mill	*11/5861*	*-/3*	
							Assemble	*11/6890*	*2/6*	
Cylinders	Cast Iron	B1	G20	200	Cast in Pairs	45/-	*Bore and Face*	*11/3140*	*2/3*	
							Drill	*12/1216*	*1/2*	
							Tap	*12/1520*	*-/8*	
							Test	*12/2170*	*1/2*	
Crank Case	Aluminium			100		75/-	*Mill*	*11/850*	*3/9*	
							Drill	*12/1307*	*1/4*	
							Bore	*12/780*	*5/6*	
							Fit Caps	*12/5870*	*2/8*	
Valves	Steel Drop Forgings 3 per cent. Nickel			800		-/8	*Turn*	*11/7805*	*-/4*	
							Mill Slot Hole	*11/9430*	*-/1*	
							Saw Slit	*11/4180*	*-/1*	
							Grind	*12/2114*	*-/3*	
Pistons	Cast Iron			400		1/6	*Turn and Face*	*12/215*	*1/3*	
							Drill Pin Hole	*12/750*	*-/7*	

Fig. II.

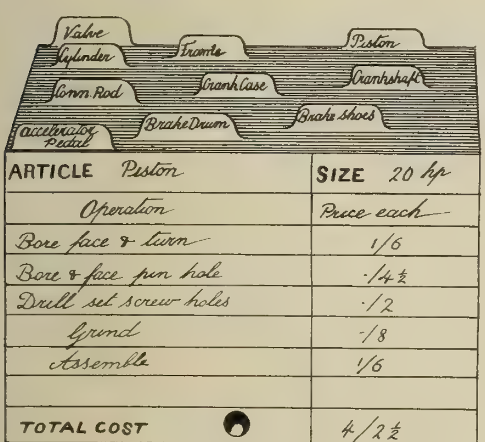


Fig. III.

and complete arrangement, and—where there is not much likelihood of there being any change in methods and consequently in prices—it takes up little room, but where much changing of design or operations is taking place the type of card and method shown in Figs. IV. and V. are preferable, as they form a more elastic record, and when any alteration in methods take place, thereby reducing the cost of production, the old records are simply cancelled by the new entry of the latest data on the next line underneath.

Upon reference to Fig. IV. the key or index to operations is shown on the front finger card. These cards run both alphabetically and numerically, as shown, each operation being numbered; for instance, should it be required to see the cost of drilling the accelerator pedals we turn up the index cards marked 4, on the indicator tab behind "accelerator pedals," as on this batch of cards is all the information required recorded in exactly the same manner as shown on Fig. V. In the same manner, should it be required to turn up the data for cost of grinding out cylinders we look for "cylinders" on the master cards, and then turn up cards marked 6 on the indicator, which is the number of the operation required. Of course, only one size of cylinder is entered on each card, and the sizes run up in rotation, the smaller size starting at the front and gradually rising until the largest power is at the back. All old or obsolete times and prices are cancelled by the new records, which are constantly coming through, and which generally show a considerable reduction as new and improved methods are introduced. With the cards shown at Figs. IV. and V., and with the operations numbered, it is much quicker to locate any particular operation required, especially where a large number of operations are involved or carried out on small items.

Any cost system, to be of any real benefit to the firm installing it, must be thorough. It must be properly controlled and supervised, or otherwise it cannot pay, and, in the writer's opinion, this is where most cost systems fail. Although installed in good faith, they never turn out satisfactorily, for the simple reason that they are very rarely worked by an adequate staff, as most employers look upon the cost department as a kind of necessary evil. Costing clerks being styled non-producers are usually very poorly paid, and have inadequate assistance. There can be no doubt, however, that in

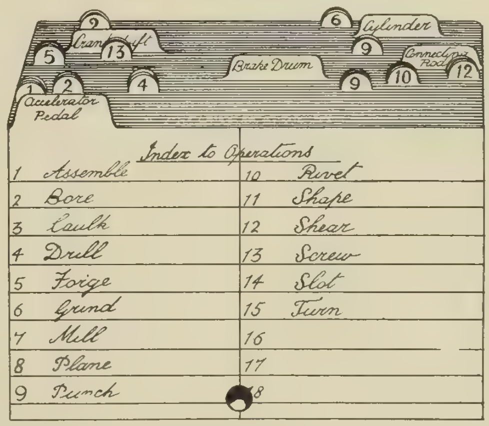


Fig. IV.

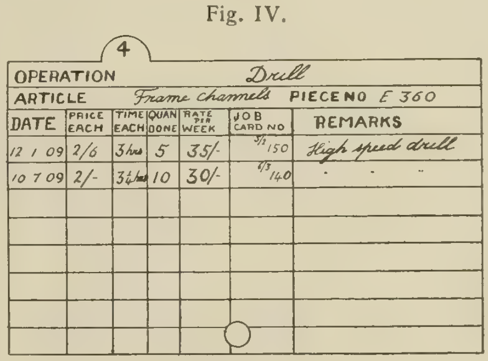


Fig. V.

any modern factory "system" does pay if properly carried out, and the costing of the finished product is no exception.

D. WALTERS.

THE 15 H.P. THAMES CHASSIS.

One of the most interesting cars in the 80 mm. by 120 mm. class.

At the recent Olympia Show there was a very large class of cars having small but powerful engines of 80 mm. by 120 mm., and among these cars there was a very great similarity in design, few having any striking deviations from what might be called the "class standard" as far as external arrangement is concerned. Among the few which, on casual inspection, seemed to exhibit a tendency to depart from the general lines was the 15 h.p. Thames, and this impression is strengthened by a closer examination of the chassis, as it reveals several ingenious features introduced in a manner which gives evidence of careful forethought on the part of its designers, especially as regards important detail.

It will be seen from the description which follows that such important units as the change gear control, lubrication and front axle details have been altered in a radical manner. Moreover, these alterations have been entirely for purposes of improvement. The engine follows the practice of its class in having a one-piece cylinder casting, but an unusual feature is the spacing of each cylinder from its neighbour to allow the fitting of a five-bearing crankshaft. The inlet and exhaust valve pockets are both on the near side of the engine and are amply water-jacketed, while care has been taken to keep down the amount of metal in the valve caps so that the heat should not accumulate therein. A single casting with four branches is used for the exhaust pipe, small baffles being cast in the pipe to minimize the interference of one cylinder's exhaust with that of another, but these baffles have been kept so small that it seems improbable they can have much effect. Cooling being by thermo-syphon, the jackets are large, while the outlet pipe is a tapering aluminium casting easily seen in Fig. I., the water inlet pipe being on the off side of the forward cylinder, to which a copper pipe is led from the radiator.

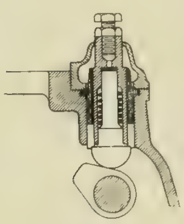


Fig. II.

The valves, manufactured from nickel steel, are provided with well radiused heads, 43 mm. in diameter, and are notable by reason of the great length of the guides, clearly shown in Fig. I. Too much stress cannot be laid on the necessity for a good bearing at this point, since wear inevitably leads to the valve seating badly—with all the attendant troubles, which are not discovered until it is desired to grind in the old valve or replace it with a new one—while the effect of a worn guide on the carburation is too well known to need comment. These guides are pressed into position, but it might be worth while to provide means whereby they could be removed and replaced easily. Turning now to the tappet gear, this is quite an unusual arrangement and one that has much to commend it, especially as regards the provision of an oil trap for each guide, which catches the surplus oil and restores it to the crankcase. The oil trap is also used for clamping the tappet guides into the crankcase in a manner made plain by Fig. II., from which it will be seen that there is a hollow gunmetal casting running over four of the valve guides, at each of which it is formed into the aforesaid oil trap, while it is secured in position by a nut between each pair of tappets. A fibre packing piece is placed between the mushroom-headed cam-striking rod and the valve tappet in order to minimize noise.

In Fig. II. it will be noticed that the cams are flat-sided, which is rather surprising, as most of the "high efficiency" engines now have cams with concave faces.

As already mentioned, the crankshaft has five bearings, it is cut from a solid forging of chrome-vanadium steel and drilled to allow forced lubrication. Each big end is white-metalled, and lubrication of the gunmetal bush in the small end takes place through a copper pipe fastened to the connecting rod side. A ring fixing is used for the gudgeon pin, but the ring is not allowed to touch the cylinder walls, and the cast iron piston shows signs of racing practice, the skirt being kept clear of the cylinder until within a few millimetres of its end, thereby reducing friction without impairing the steadiness. The flywheel is attached in an unusual manner, as it is slotted to fit a rectangular projection which crosses the face of a large flange on the flywheel, instead of being keyed on the more usual taper.

At the forward end of the crankshaft a steel pinion drives a phosphor-bronze gear keyed to the camshaft end, the camshaft and its cams being cut from the solid bar instead of being made from the cheaper (and probably equally satisfactory) stamping which is now usually employed for solid cam work. Beyond the camshaft driving wheel and meshing therewith is the steel pinion which drives the magneto through a special form of dog clutch, shown in Fig. I., this having nine teeth on the driving side and ten teeth on the driven side, thus allowing for very minute adjustment; although it is more expensive than the usual arrangement. It will be noticed that the magneto shaft runs on ball bearings and drives the fan by means of a pulley keyed to the forward end. At the rear end of the camshaft will be seen the cam-actuated piston pump which supplies air pressure to the petrol tank, and delivers through a ball valve (see Fig. I.), guides on the piston keeping this true with the cam. Oil circulation is maintained by a paddle pump, and is of interest because it is undoubtedly one of the best and most accessible low pressure systems at present in use. Turning to Fig. III., it is clearly shown that a skew gear on the camshaft drives the pump by means of a long spindle, well provided with bearing surface and suitably grooved for lubrication.

Oil is taken in on the left-hand side of the pump casting, and is delivered through the long copper pipe seen on the right, which is curled to prevent fracture from vibration. After entering the top casting the oil flows along a pipe running horizontally in the top of the base chamber to the drilled crankshaft and main bearings as shown. At the extreme end of the delivery pipe a by-pass valve will be noticed which acts in an extremely neat and effective manner, delivering through the curved pipe shown in dotted lines, and along the gutter in the crankcase top to very large copper pipes which feed extra wide troughs shown clearly in Figs. I. and III. In these troughs the big ends are allowed to splash so that should the overflow become excessive, owing to the choking of a bearing pipe, splash lubrication is set up. Now, with most systems of lubrication inspection of the pump is a troublesome job, entailing also the entire emptying of the sump, but in this case the slacking back of the butterfly nut at the top of the pump gear allows the entire pump, together with its skew gear, to be removed from the car. It appears, therefore, that the advantages of a pump at the bottom of the sump are combined with the accessibility of one placed high up outside the crankcase. Further evidence of the extreme care with which the car has been designed is afforded by the arrangement of the oil pipe lines. Most of these are external, and all have loops to prevent vibrational fracture, while inspection doors are provided for those pipes which cannot be examined otherwise, as will be made clear by reference to Fig. III.

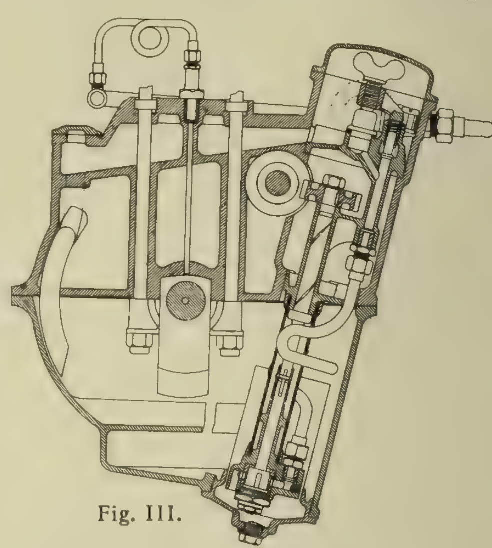


Fig. III.

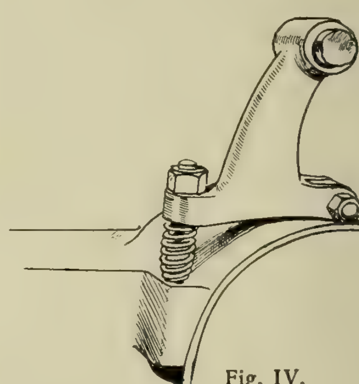


Fig. IV.

Mounted on the half speed wheel case at the forward end of the engine is the fan, the adjustable bracket of which is fully explained in Fig. IV. A peculiar feature is the adjustable ball bearing, but

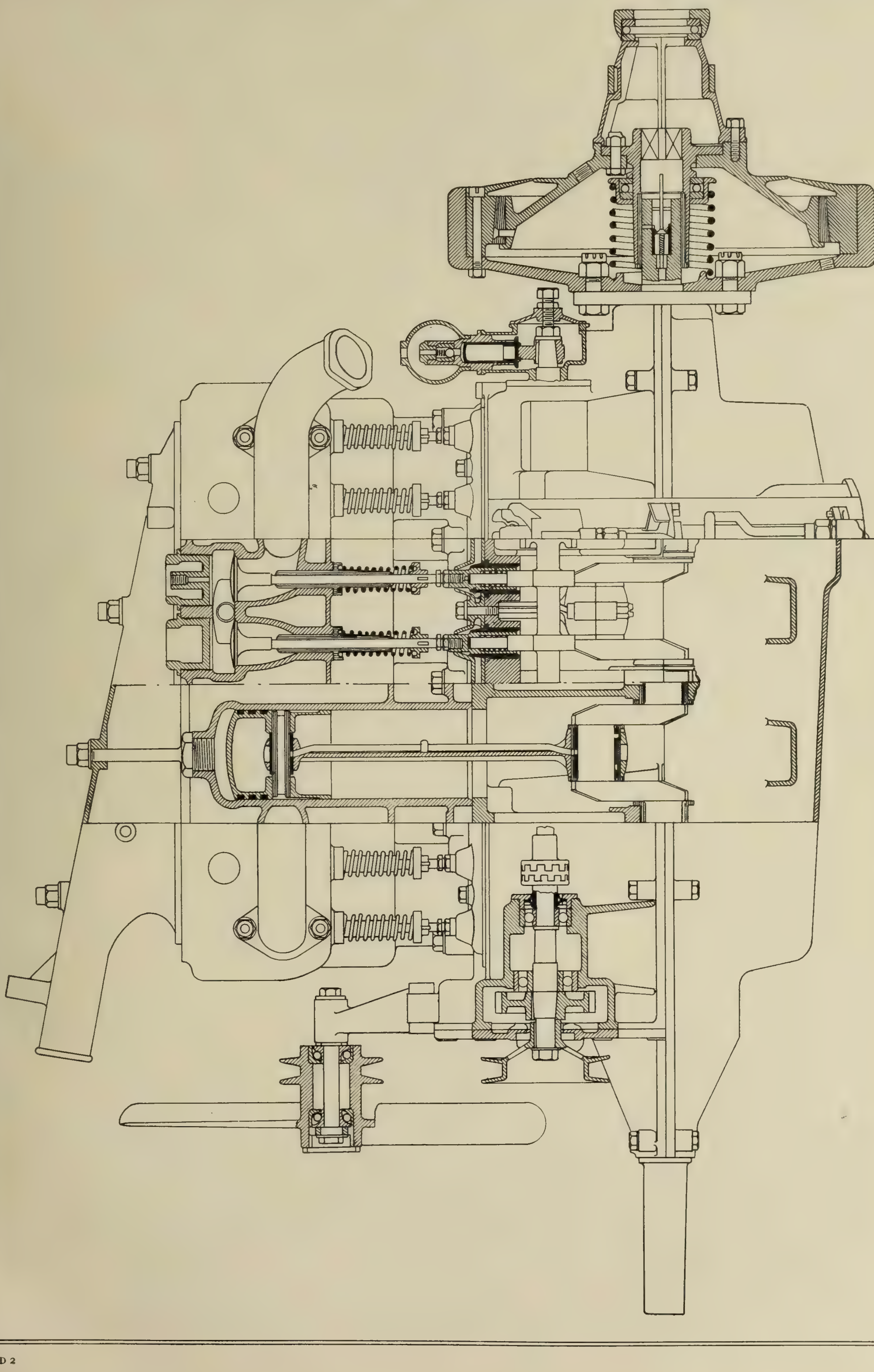


Fig. I. THE 15 H.P. THAMES ENGINE.

this appears to possess no advantage except that of cheapness, especially as it is not provided with a lubricator: indeed, the whole fan is scarcely a necessity as the engine keeps cool without its fan belt. A honeycomb radiator is used, and here again evidence is forthcoming of the thought which has been given to the various units, as a peculiar form of trun-

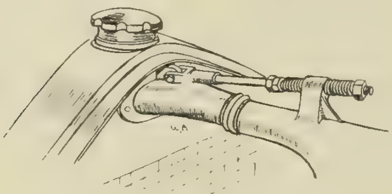


Fig. V.

nion is used to attach the radiator to the frame, while the upper part is held by a spring attachment, both fittings made clear in Figs. V. and VI.

Ignition is by magneto only and is variable, while all wiring is taken in a pipe until opposite the plugs, which are on the off side of the engine, the magneto being held in place by the usual straps and thumb-nut.

The carburettor is a Polyrhoë and, bearing in mind the fact that thermo-syphon circulation is used, the difficulty of providing efficient water heating for the carburettor becomes at once apparent. To overcome this difficulty the return pipe from the carburettor is brazed to the exhaust pipe at a point slightly in advance of its re-union with the cylinder outlet pipe, this exceedingly simple device ensuring regular water flow.

Fig. I. also shows a section of the clutch, which is of the rare internal metal cone type, having a self-contained thrust except when withdrawn. The metal cone is provided with grooves which contain the oil necessary to allow smooth engagement, and guards are provided to prevent this oil from working out of the clutch. A neat arrangement is a small spring-controlled valve which only admits oil to the spigot when the clutch is held out, thereby preventing excess of oil finding its way to the clutch faces.

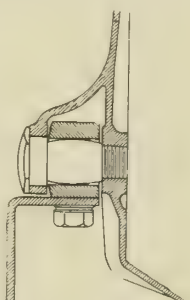


Fig. VI.

The gearbox is connected to the clutch by a shaft which is allowed sufficient play on the square ends to act as a universal joint in a small degree. This feature, which is wisely being altered, has drawbacks inasmuch as a very bad knock is produced when the surfaces wear, even ever so slightly. It is probable that the new joint will take the form of a leather disc, having one shaft bolted to its centre and the other to its circumference, after the pattern of the Isotta-Fraschini, though this change is not likely to be made until the makers are entirely satisfied as to the durability of the leather.

The thrust ring shown in Fig. I. withdraws the clutch by means of a bell crank lever operating on the thrust from the pedal, and this bell crank brings a novel form of clutch brake into engagement. It will be observed from Fig. VII. that the clutch is withdrawn before the semi-circular brake block comes into contact with its drum, and it is interesting to note that the brake block is lined with Raybestos. Altogether most effective, this is an expensive method to adopt when it is considered how simple most clutch brakes are, and also the braking effect obtained is rather too violent, as it can frequently be felt when the clutch is withdrawn too rapidly.

The gearbox is supported by a dropped cross member at the forward end and by a very deep member at the rear, to which a flange running round the gearbox is bolted. In connection with this method of support it would be interesting to add a word on the lining up of the engine, for it is accomplished in a manner open only to those firms who have a big plant at their disposal. The engine, being bolted to the frame at its neutral line, has to be supported on aluminium packing blocks, and the forward end of the gearbox rests on other packing pieces to facili-

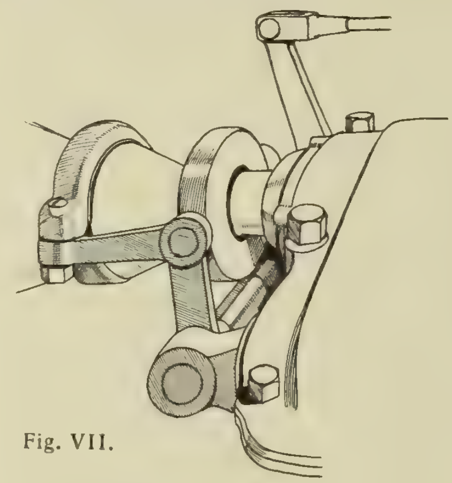


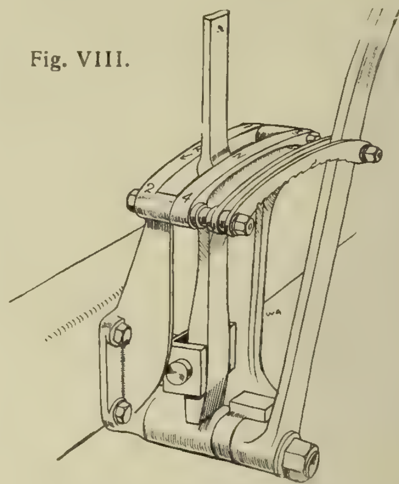
Fig. VII.

tate adjustment; these are fastened to the frame first, then the whole is placed on a planing machine while the packing pieces are faced up. As soon as this is finished the deep cross member is bored and faced up to receive the gearbox flange by a special tool attached to the bed of the planing machine. By these means very little adjustment is needed when the engine and gearbox are erected. When in position on the frame a pointer is attached to the gearbox driving shaft and is adjusted to the outside circumference of the flywheel, so indicating any error in alignment which is corrected by alterations in the packing blocks.

The gear changing mechanism differs greatly from the usual gate system by having a plain slot for the lever to slide in, although the actual motion of this lever is that of an ordinary gate change. Fig. VIII. shows the quadrant plainly, and to fully grasp the motion it must be remembered that the lever, while in neutral, is pushed outwards before the third or fourth position is reached, thus moving the actuating tube inwards, while inward movement from neutral is necessary before either first or second speed can be engaged. Still further motion inwards, allowed only when the catch is lifted, permits the reverse position to be reached.

Turning now to the means adopted for accomplishing this result it will be seen in Fig. IX. that the actuating tube has a selector arm working in a small and simple gate formed by the ends of four set pins. This selector engages the striking rods according to the particular slot which happens to be opposite it, this position being governed by the inclination of the change gear lever. Spring plungers falling into notches in the striking rods keep the latter in position. The whole arrangement has the merit of extreme simplicity, while the greater part

Fig. VIII.



of the actuating gear is covered in, thus increasing the durability without rendering it unduly inaccessible.

The gearbox is shown in Fig. X. and gives four speeds, the top being a dog clutch direct drive of 4 to 1, the third 5.96 to 1, second 9.6 to 1, and first 14.9 to 1. The gears are case-hardened after cutting, and any exceptional noise is reported after the first test run. In the event of an unfavourable tester's report the gears are run in with a grinding medium, or smeared with red lead to obtain the high spots which are afterwards rubbed down by hand. All the shafts are fitted with ball bearings of great size, housed in the aluminium. Those on the direct shaft, which are seen in the drawing, and those at the front of the direct-drive clutch have 11 mm., while that at the tail end is provided with 25 mm. balls. Although the fitting of outer ball races in the aluminium has many manufacturing advantages, it is likely to cause trouble due to wear, and it would seem a better policy to provide thin brass linings, which could be renewed, but the great size of the races may make this unnecessary. The peculiar arrangement of the gears which are always in mesh is very plainly shown on the drawing between the first pair of ball-bearings, and incidentally it may be remarked that the spacing of the two ball-races, necessitated by this arrangement, is to be commended, as it renders un-

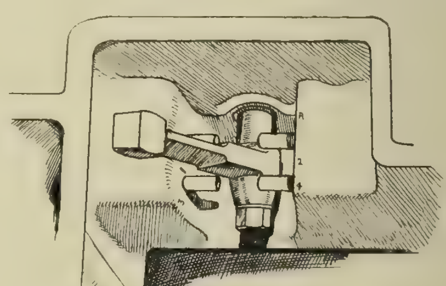


Fig. IX.

equal loading unlikely to occur. Care has also been taken to provide a large ball bearing for the spigot end of the splined shaft. At the back of the gearbox, between the first and second speed gears, the selector arm is visible and to a lesser degree the pins forming the gate. A forked lever seen immediately below the same pair of wheels is that with

which the reverse may be brought into action, the reverse wheels themselves being omitted for the sake of clearness.

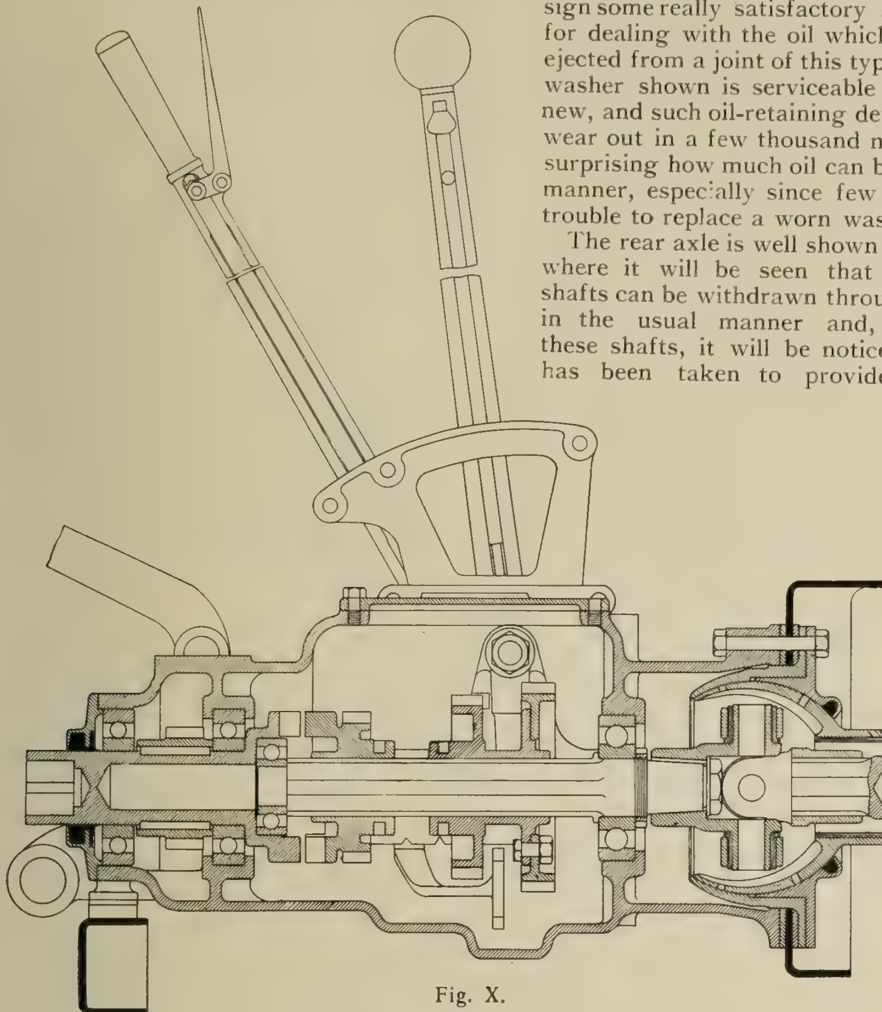


Fig. X.

An extension of the gearbox casting forms a cover for the ball-ended universal joint, over which it also forms a flange for attachment to the frame. This is shown in the drawing, and the exceptional size of the cross-member to which it is fixed is also made clear therein. Other noticeable parts are the packing pieces (supported by a dropped cross-member below the forward end) on which the gearbox rests, and with which lining up is accomplished.

Turning to the universal joint, this differs little from the ordinary, save perhaps as regards the pins which are hollow, thus saving weight without reducing the bearing surface. The gearshaft jaw is keyed to a taper, while that on the propeller shaft, however, has a splined fixing, and the ball is provided with lubricating holes. It should be worth while to design some really satisfactory arrangement for dealing with the oil which is usually ejected from a joint of this type, as the felt washer shown is serviceable only when new, and such oil-retaining devices usually wear out in a few thousand miles. It is surprising how much oil can be lost in this manner, especially since few drivers will trouble to replace a worn washer.

The rear axle is well shown in Fig. XI., where it will be seen that the driving shafts can be withdrawn through the hubs in the usual manner and, concerning these shafts, it will be noticed that care has been taken to provide an ample radius at the point where the driving jaw springs from the shaft itself.

This is one of the few axles in which thrust bearings are provided for the rear hubs and it will be observed that great care has been taken to provide very ample resistance for thrust throughout the chassis. Contrary to standard practice, a spur differential gear is used instead of the customary bevel, the spur pinions being of ample size and so arranged that the whole can be withdrawn from the rear through the large cover. In order to facilitate cleaning and to simplify the external appearance, a very neat spun copper cone is fixed by screws to the main casing and extends forward to the torque tube, thus smoothing out the otherwise awkward corners rendered necessary by the attaching bolts and strengthening ribs. A drain plug is provided at the bottom of the rear cover and might with advantage be transferred to the lowest part of the axle-case itself.

A small refinement which could be suggested is the addition of some device whereby the space reserved for the bevels and differential could be filled with oil without at the same time filling the whole of the axle case, although this is an extremely unusual and perhaps luxurious device.

Fig. XII. fully explains the front wheel brake arrangement, showing the neat single leaf spring which is used to return the brake shoes to the "off" position. By following the levers seen in dotted lines the actuation of the brake is easily understood, remembering that there is a slot in the swivel to give the short arm of the bell-crank-lever sufficient movement in all positions. A groove is cut in the face of the actuating wedge in order that this may revolve with the brake shoes when the road wheel is not parallel with the frame of the car. Both the front and rear brake shoes are interchangeable, and are faced with metallic packing rivetted to the cast iron brake shoes.

In Fig. XII. the brackets, with which the torque rods are clipped to the front axle are also apparent, and Fig. XIII. is a sketch of the socket into which the rear end of the same rods are brazed, this illustration also showing the method of attachment to a cross-member of the frame.

Care has been taken to attach the rear brake actuating rods to a bracket on the frame at points which are on a line drawn through the universal joint, so that movement of the rear axle on its springs does not alter the brake lever position. This is one of those usually neglected minor points which are responsible for the absence of rattle when the car has been in use for a considerable time.

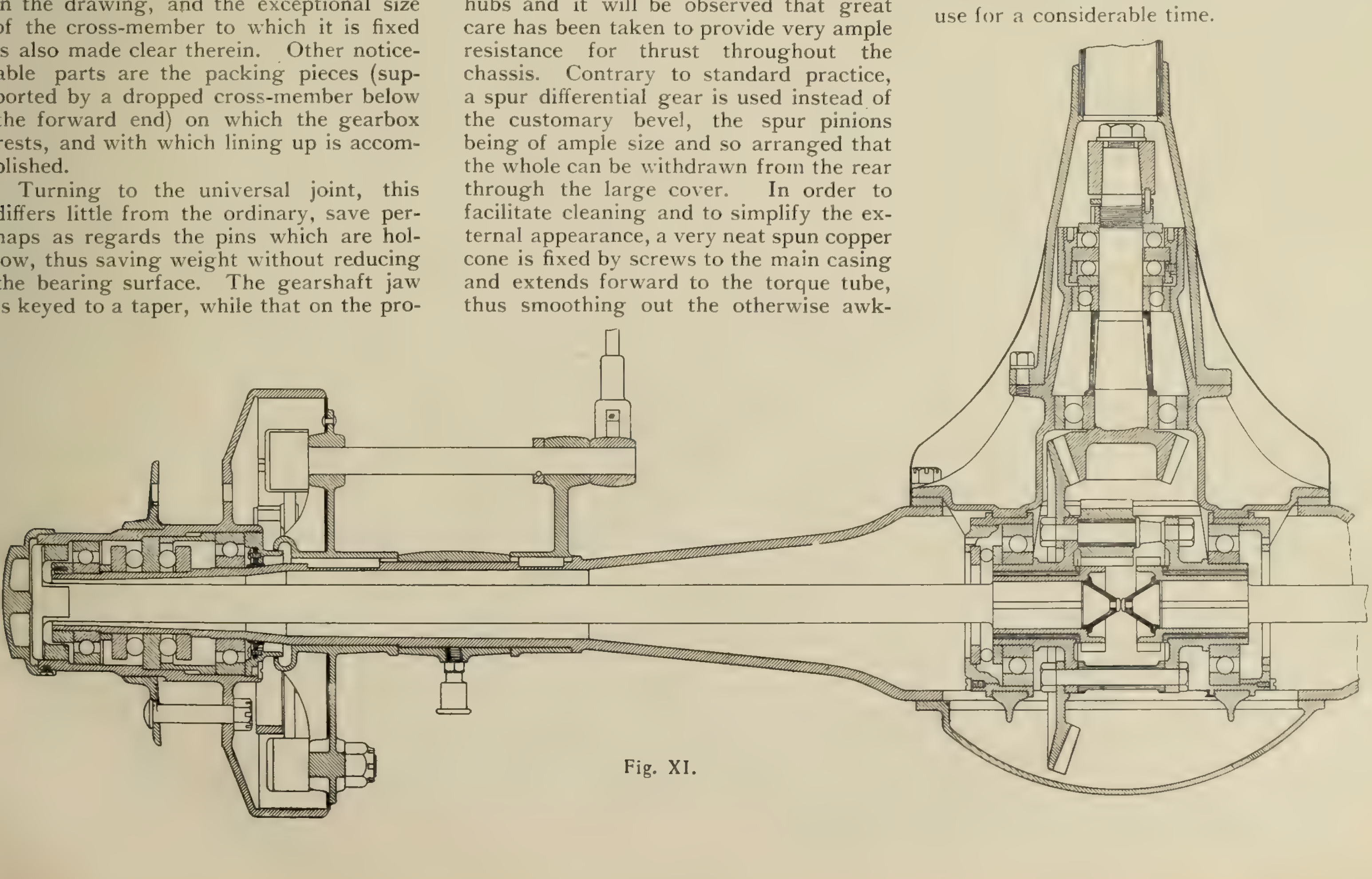


Fig. XI.

Steering is by worm and worm wheel, the steering tube having a large double ball thrust bearing placed immediately above the worm, and, contrary to usual practice, the worm and worm wheel are without play when the chassis is erected.

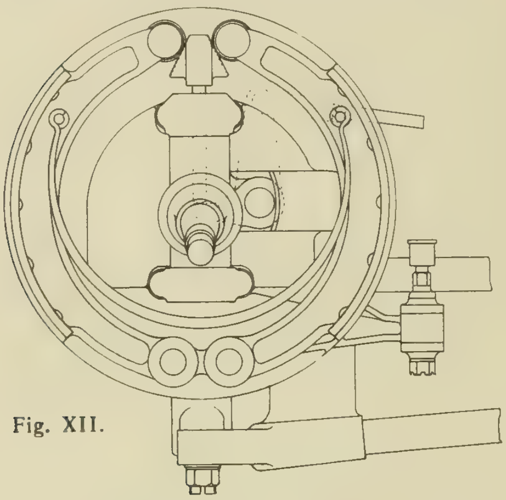


Fig. XII.

Fig. XIV. shows the outside of the worm case and the method whereby the control gear is connected to the wheel. Each lever is keyed to a bevel segment meshing with a small bevel pinion connected to the steering-wheel lever, and the control rod for the ignition is shortened by passing through the upper half of the crankcase to the magneto contact-breaker on the further side. A large grease cup is used to supply lubricant to the worm. In order to suit individual tastes three different steering column angles are obtainable, but while this is a commendable practice, it would be better if a simple form of adjustment was provided on the steering column itself.

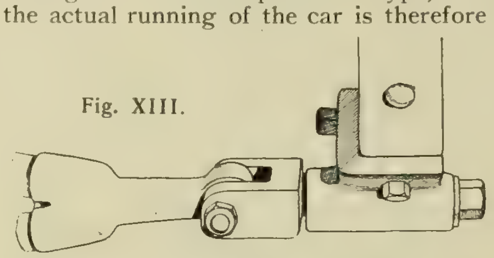
A strong gun-metal plate bolted to the frame provides a substantial support for the clutch, brake and throttle pedals, the latter being arranged to move the throttle only beyond the position set by the hand lever. Countersunk holes are drilled in the pedal bosses where movement takes place on the pivot tube and the same means of lubrication is used for the brake movements and compensating gear. Undoubtedly it would be better to fit lubricators or small greasers, for, although a minor point, their absence may result in a stiff bearing and undue wear, since dirt speedily chokes an uncovered hole.

Save for the deep cross-member behind the gearbox there is little extraordinary in the frame, but that the forward dumb iron arrangement is decidedly uncommon, consequent on the employment of front axle radius rods, for to allow spring movement each of the forward springs have slotted eyes which slide on rollers fitted on pins bolted to the dumb iron.

Semi-elliptic springs of great length are used for the rear axle instead of the common three-quarter elliptic and in support of this practice it can certainly be said that the car is unusually comfortable.

It would, perhaps, be anticipated by an examination of the engine drawings, that the engine would have the somewhat harsh feeling usual with its particular type, and the actual running of the car is therefore

Fig. XIII.



a pleasant surprise, for the engine runs extremely sweetly, even when racing hard on a low gear, and is remarkably well balanced. There is very little valve noise and but slight sound from the timing gears. An excellent feature is the flexibility and lack of jerkiness when picking up from a speed of five miles an hour on the high gear, without clutch manipulation.

The clutch is easy in action, and save for an occasional jerk from the clutch brake, stopping by either the hand or foot brakes is very smooth. There is none of the decided action which is a characteristic of the usual gearbox brake, and this may or may not be a decided advantage,

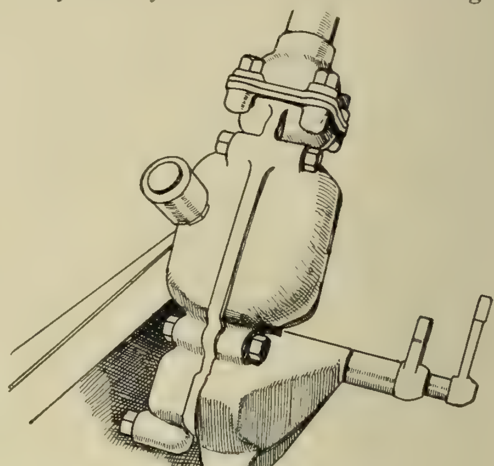


Fig. XIV.

that is to say, there are occasions on which an immediate stop is a necessity, although the sweet action is ordinarily pleasant. Probably owing to a large extent to the front axle radius rods the car holds the road in a remarkable way, especially on an extremely rough surface, when no involuntary deviation from straight can be observed, consequently, even with smooth tyres, side-slips are of rare occurrence, especially on corners or when mounting a steeply barrelled road.

A very wise policy has undoubtedly been followed in placing smooth running before abnormal power, as the resultant is a car eminently suitable for the everyday use to which the majority of cars are subjected, and the easy change speed mechanism with a reasonably quiet third gear should prevent the chassis being too much forced on top.

CARBURETTOR ACTION.

Professor Morgan replies to Mr. Brewer's criticisms of his investigations.

IN the issue for January, 1911, Mr. Brewer expands and amplifies the remarks he made in the discussion held after the paper entitled "Carburettor Action," read by Messrs. Morgan and Wood before the Institute of Automobile Engineers last November, and published by *The Automobile Engineer* in December, 1910. The object of both groups of investigators is to state a law connecting the quantities of petrol and air delivered to an engine by a plain tube carburettor. Disregarding, for the time, variations due to temperature changes, it is a surprise to find that the two results obtained are fundamentally discordant.

It is very necessary to determine whether the divergence is due to differences in conditions or due to error. In the following discussion it will be convenient to refer to Mr. Brewer's graphs connecting petrol discharge and air velocity as curves Y, and the graphs connecting petrol discharge and suction as curves Z. Mr. Brewer is not quite clear as to how the graphs Y were obtained, but apparently his special apparatus, described on page 228, was used, in which it is to be presumed that the quantity of petrol was measured directly, and the air estimated from the suction as read on the gauge. But we are met here with the difficulty that Mr. Brewer uses the gas velocities on curves Y in connection with curve D, which is the graph

$$V = 8.15\sqrt{2\,gh},$$

connecting suction and air velocity, to establish the relationship between petrol discharge and suction on curves Z. Now, if the air graph Y was estimated from observed suctions, why the necessity of this conversion? Why not plot observed suction and observed petrol. This obscurity renders it perilous to argue on these results, and modifies their value extremely. In any case, according to Mr. Brewer, these curves Y represent the type of relationship which exists between air velocity and the petrol delivery under similar conditions to those described in the original paper, and reprinted in *The Automobile Engineer*, on pages 207 and 209. (See also Mr. Brewer, page 228, col 3.)

Mr. Brewer's curves Y, neglecting the lower limbs are in the form

$$Q = k\,(a+b)^2 + c \quad\ldots\ldots\ldots\ldots\ldots\text{I.}$$

where Q is the quantity of petrol delivered per hour, a is the velocity of air and k, b, c are constants.

Now, in the original paper it was stated that the relationship between volume of petrol and volume of air is the form $Y = AX + b$. Stating this in terms of volume of petrol and velocity of air, and using the above notation this becomes

$$Q = ka + b \quad\ldots\ldots\ldots\ldots\ldots\text{II.}$$

In Fig. I. Mr. Brewer's U curve for the 0.90 mm. jet is plotted together with a curve giving petrol and air relationship, according to my own investigations. It is therefore mathematically and graphically apparent that the two results are irreconcilable. It is now to be decided whether or not one of these results must be rejected. The results given in the I.I.A.E. paper were obtained by actual measurement, and can only be subject to errors of observation which, from their description, could not be serious. Further, such a method of measuring the

actual quantities dealt with is unaffected by reactions in the engine, whether due to wire-drawing in the pipes or the compression in the cylinder. The suggestion that methods depending on estimations, as nebulously outlined by Mr. Brewer, are better than methods of direct measurement, can only be described as amazing.

In deciding between results I. and II., it is worth while to note that Mr. Brewer, page 228, col. 3, proves and confirms by reference to Unwin that the flow of petrol

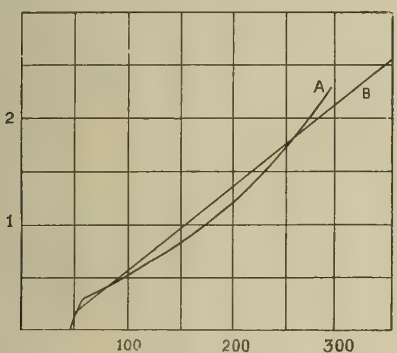


Fig. I.

is proportional to the square root of the suction; then, on page 229 graphs *Z*, states that the law connecting flow of petrol and suction is in the form :—

$$Q=aV+b.$$

To make clear the difference, curve A, Fig. II., shows the mode of petrol delivery under suction according to the above law, and curve C shows the relationship of petrol to suction according to Mr. Brewer, and confirmed by Unwin. Now, Mr. Brewer cannot have it both ways, either he must admit his graphs *Y* and *Z* are wrong, or show clearly in what respect his conditions were peculiar, and how such anomalous results were obtained. The results obtained by myself and Mr. Wood have also been plotted in Fig. II., curve B, both suction and petrol being observed directly. It will be seen how closely this graph follows the curve given by Unwin's formula. It may be noted here that in the latter experiments a jet was used which was less than 1 mm. long, and was approached by a coned aperture—hence the large coefficient of C=0.95.

In the paper on "Carburettor Action" the droop on the upper ends of the graphs (see *The Automobile Engineer*, page 208) has been stated to be due to resurgence in the induction pipe. Mr. Brewer admits the existence of such a surging, Dr. Watson proves its existence at fairly high speeds (Proc. I.I.A.E., 1908-9), and we have obtained indicator diagrams showing the same effect at high speeds.

The droop is not a velocity function, as will be seen in Fig. V., page 208, where no droop appears at forty cubic feet of air for the 900 r.p.m. curve. That the droop is due to surging with open throttle is at least a reasonable suggestion to make.

Mr. Brewer appears really to agree with our deductions, for he shows that

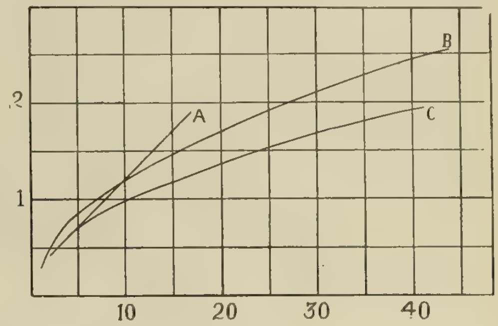


Fig. II.

steady flow of petrol is to be expected, but fails to note that the surging flow of air would lessen the effective volume of air passed, thus giving the enriched mixture effect noticed. Our suggestion as a simple solution of the carburettor difficulty was not to give "a set dribble to the jet," but to give a continuous supply limited to a constant quantity, and it may be added that this suggestion has already been tried with success in more than one carburettor.

A TRIAL OF THE BANNER ENGINE.

And an account of an improved design.

IN the December issue of *The Automobile Engineer* a short account of the Banner engine was published and Fig. I. shows the present design in diagrammatic form. The new type differs from the engine which was running in London during the Olympia Show, in that the crankshaft is now set slightly out of centre to allow the rotary valve to be driven by a straight shaft without universal joints. We recently had the opportunity of observing the performance of the engine during the course of a twelve mile run over comparatively hilly country, and it may be said at once that there is nothing in the behaviour of the car which would indicate that the engine was not an ordinary good four-cylinder. In fact, at high engine speeds the vibration appears to be unusually slight, while no bad periods were to be noticed up to the limit speed of revolution, which was between fifteen and eighteen hundred r.p.m.

Of course, so far no attempt has been made to produce an engine giving a very high power by comparison with the cylinder dimensions, so fairly smooth running was to be anticipated, but whereas a four-cylinder vertical engine of similar weight and size often creates considerable vibrational noise at speeds of revolution much in excess of a thousand per minute, the Banner engine is quite smooth at fifteen hundred, and vibrates but little, even when raced light at its utmost speed.

As a reminder of the mode of operation may be useful, it is as follows :—The rotary valve is quite a loose fit in its chamber, and is supposed not to touch the walls at all. Fresh mixture enters from the dome on top of the cylinders, and is passed to the cylinder ports in rotation, while exhaust gases pass out beneath the central division of the valve. By these means a good and free opening is provided for the gases and the timing is controlled by the rotation only. The poppet valves merely serve to protect the rotary valve during the explosion stroke, and are lifted early, remaining open dur-

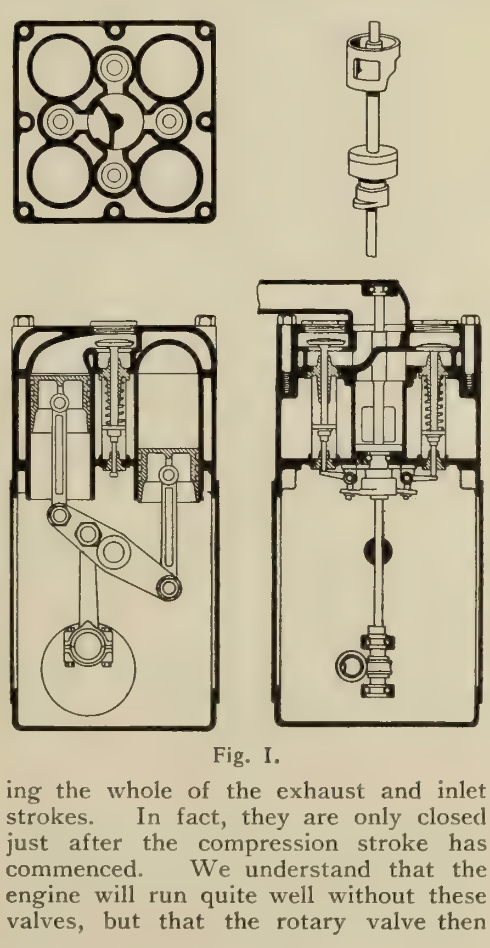


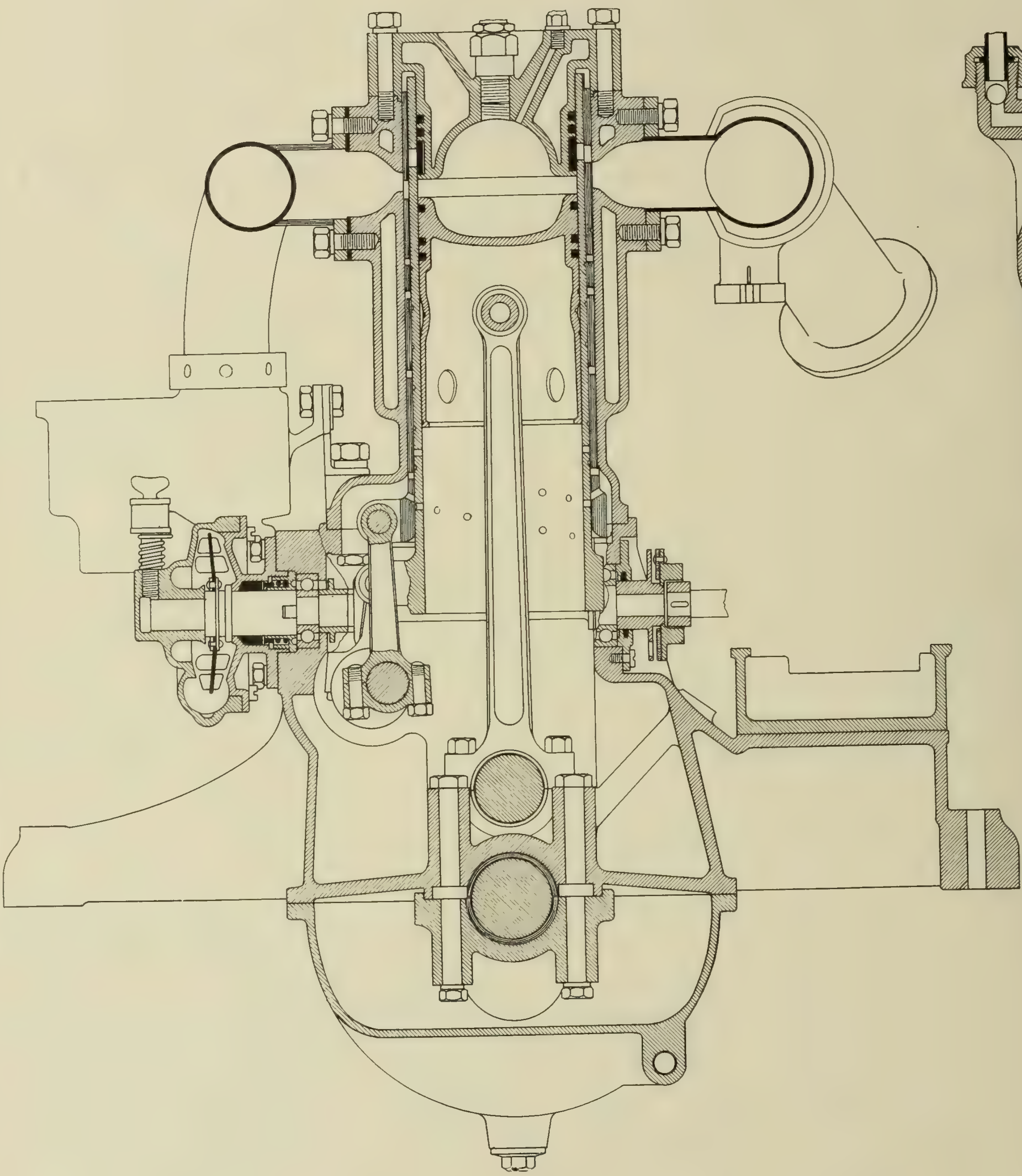
Fig. I.

ing the whole of the exhaust and inlet strokes. In fact, they are only closed just after the compression stroke has commenced. We understand that the engine will run quite well without these valves, but that the rotary valve then soon becomes dirty and clogged with carbon deposit.

Of course, only careful bench tests can show how this engine will compare with other and older types on the scores of efficiency and durability, but there seems no doubt that it is an engine which can easily be made both quiet and smooth, as the decidedly roughly made and very heavy trial model undoubtedly possesses these qualities.

The extremely short length would be very convenient for many purposes, and the increased width is no disadvantage for car work, while another point in favour of the construction is that an eight-cylinder could be made to occupy very little more bonnet space than a vertical four-cylinder of not above half the horse-power.

The obviously weak point is the introduction of the beams and concomitant extra bearings, but there is no difficulty in making the latter of ample size and the stationary shaft on which the beams oscillate serves as a most convenient oil conductor for feeding the big ends. Owing to the fact that the angularity of the connecting rods from the pistons to the beams is very slight, and that the angularity of the two lower connecting rods affects pairs of pistons instead of single reciprocating masses, the balance should be particularly good. This is borne out by the behaviour of the trial engine, which has very heavy pistons indeed. It should be possible to use very short pistons, as the side pressure is slight, and so get unusually light reciprocating pieces. A new engine made to the design of Fig. I. is now nearing completion, and will be tested fully during the next few months.



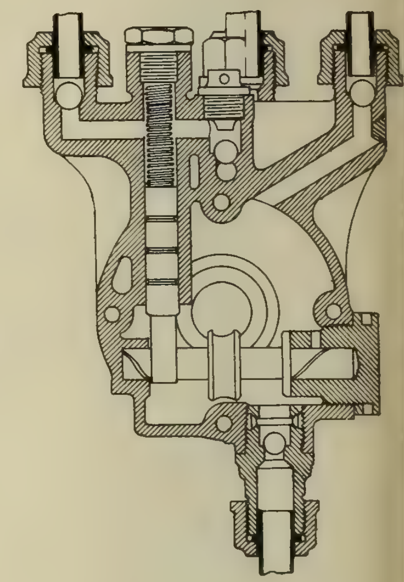


Fig. II.

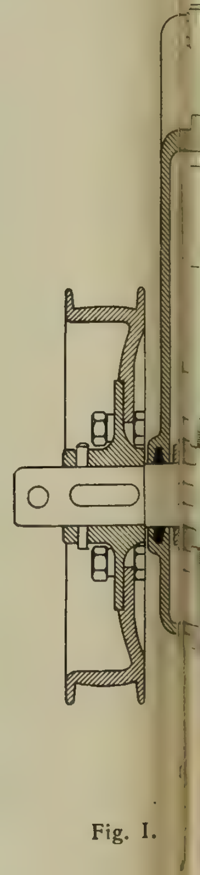
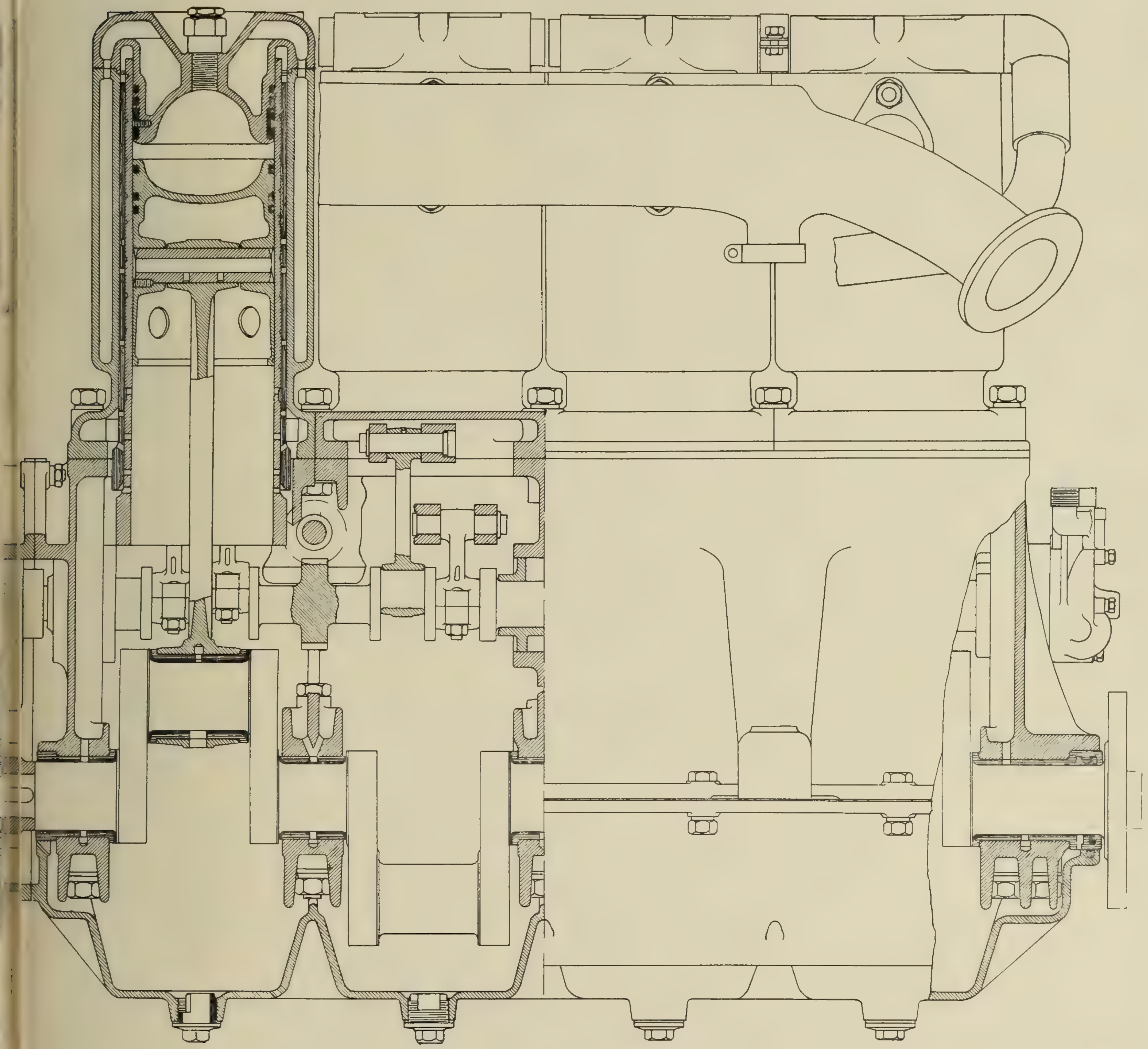


Fig. I.

THE PANHARD-KNIGHT

Inset, sectio

A description of this engine



SLEEVE VALVE ENGINE.

ofOil Pump.

pears on page 264, overleaf.

THE PANHARD-KNIGHT ENGINE.

A description of the design shown on pages 262 and 263.

IN examining the sleeve-valve engines which have been constructed by different firms under the Knight patents it is interesting to notice the individual characteristics which have been given to the original design by the various manufacturers. If a comparison is made between the Panhard engine and the similar Daimler or Minerva designs this point becomes at once conspicuous.

The engines which have been constructed by the Panhard Company have always been notable for their solidity, rendering them extremely serviceable, albeit on the heavy side, and there has been a tendency to develop an engine with as few parts as possible. Both these features are prominent in the drawings of the Panhard-Knight engine, on pages 262 and 263, being emphasized by the absence of troughs with their attendant rod control, the size of the sleeves themselves, the great length of the pistons and the use of plain splash lubrication.

The whole lubrication system is a departure from that usually employed with Knight engines. A pump driven by the tail end of the eccentric shaft supplies oil at pressure to the main crankshaft bearings by means of an external copper pipe and an aluminium lead shown in the cross-sectional view of the engine. The sleeves, gudgeon pin, piston and big ends are all lubricated by splash from the large chambers into which the lower half of the crank case is divided.

The oil pump, Fig. II., is of the double-acting plunger type, oil being sucked from a reservoir bolted to the frame below the footboards (through the lowest pipe seen in the drawing) by the upstroke of the cam-actuated plunger.

On the down stroke the ball inlet valve closes and oil is delivered by the pipe on the extreme right of the pump casing to the drip feed on the dashboard, whence it returns by the centre pipe to the upper side of the plunger. The next upward movement then forces its charge to the main crankshaft bearings by means of the left-hand pipe. Thus it will be seen that the drips serve as oil circulation indicators.

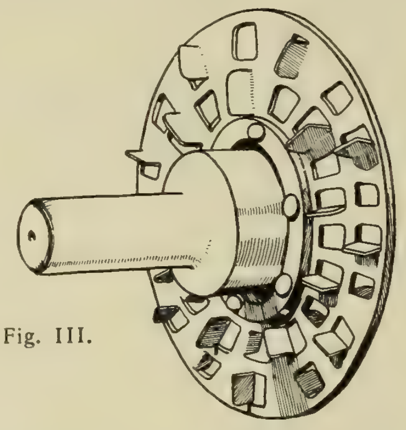


Fig. III.

To facilitate the lubrication of the outer sleeve and cylinder wall 4mm. holes are drilled in both sleeves, in addition to the spiral oil-grooves. The use of plain splash lubrication for all parts save the main bearings, is surprising on an engine of this type, since it must necessitate the provision of a larger clearance between the sleeves than would otherwise be desirable, and it is anything but economical when compared with the forced systems which have now established themselves in this country.

It is the more surprising when it is remembered that there is already a pump on the engine, and it would seem to be better to use the pump to force oil direct to the bearings than to merely pass it through drip lubricators which, if used as an adjustment for quantity of oil, must also affect the pressure.

A striking feature of the water circulation system is the unusual form of centrifugal pump employed therein. This appears clearly in the sketch Fig. III., showing the vane disc, which is made of soft brass. The vanes or paddles are formed by pressing out sections of the disc in alternate directions and in two rows, thus establishing a form of double centrifugal pump and reducing the manufacturing costs considerably. The carburettor is the type of Krebs, which has been associated with Panhard work for some years and is too well known to need description here.

In other respects the engine does not differ to any great degree from the parent design, and is rendered clear by the drawings. Everywhere, however, the stamp of "Panhard" is most obvious, and the engine appears suitable for long-continued hard work. Each detail is substantial and simple, there being practically no complicated process from the foundry or machine shop point of view. The pistons are obviously rather heavy, and the purpose of the very deeply cupped heads is not clear, except, perhaps, that the cupping enables the total depth of the engine to be reduced somewhat.

MAKING INTERCHANGEABLE CRANKSHAFT BRASSES

The object of this method of manufacture is to produce half bushes which shall be quite reliably interchangeable so as to save the trouble necessitated when it is essential to keep the two halves together throughout the manufacturing process and in the stores. This particular method also does away with the need for soldering the halves together for part of the machining, which is another point in its favour, as the soldering is a waste both of time and labour.

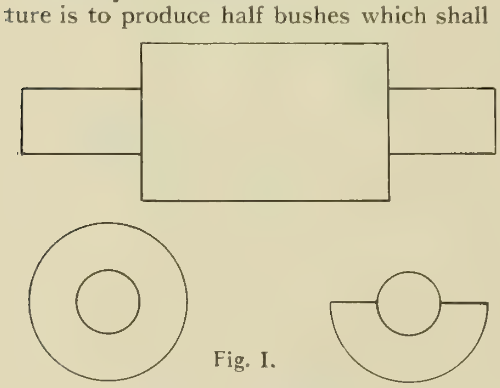


Fig. I.

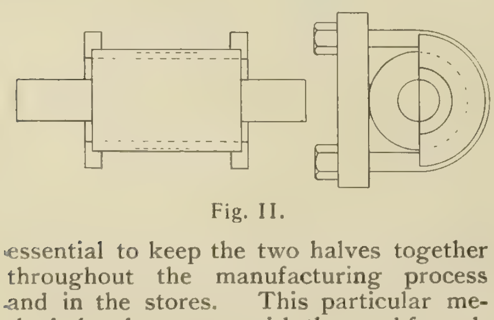


Fig. II.

There is practically no machining allowance on the joint, the first operation being to face the halves up on a disc grinder. When this is done each half is taken singly and attached to the jig shown in Fig. I. This jig consists of a piece of mild steel turned with a wide collar, which is subsequently half removed by milling and the half brass is attached by means of the clip shown in Fig. II. Eye and hand alone are relied on for centering the brass longitudinally, so the attachment is a very quick and easy job, and the flanges can be turned to gauge size with almost equal quickness. The next operation is to place a pair of brasses together in a split chuck for boring, turning the radius on the outer end, and recessing by the tool shown in Fig. III., the inside being left rough so as to provide a good surface to take the white metal. The brasses are then white metalled and are bored out to size, the radius being put on at the second end at the same time, and the outside is turned on the cup arbour shown in Fig. IV. Fig. V. shows the finished brass with an inset view of the radiused part to a larger scale, the purpose of this peculiar shape being to make it impossible for a fitter to waste time trying to make the radius on the brass fit that on the crankshaft, as an old steam hand will often do.

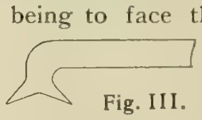


Fig. III.

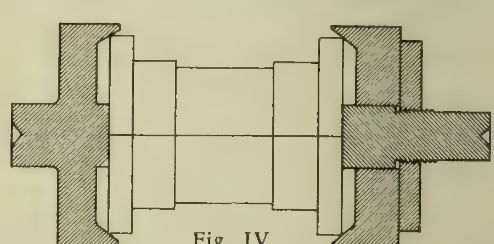


Fig. IV.

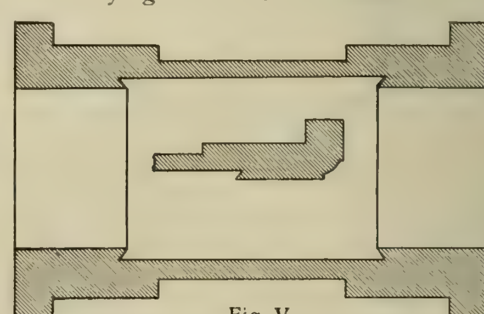


Fig. V.

O. LINLEY.

THE STRENGTH OF CASTELLATED SHAFTS.

Being a Paper read before the Institution of Automobile Engineers by Charles Edward Larard, M.I.Mech.E., A.M.Inst.C.E.

As far as the author is aware, no published information is available for the guidance of the motor car manufacturer relating to the strength, elasticity and dimensions of clutch and gear-box shafting.

The best and the latest practice, however, is to use castellated shafts similar to those tested and here reported upon by the author, or alternatively shafts of the form shown in Fig. III. In either case the grooves and key-ways in the sleeve or boss are cut out so as to leave solid projections or keys corresponding to and fitting in grooves or key-ways in the shaft. In every case where gear wheels are moved into engagement there must be no constraint with respect to axial movement along the shaft, while at the same time the fit should be good enough to prevent rotational play of the wheel on the shaft. If the first condition is to be fulfilled, it follows that the shaft should be made stiff enough, *i.e.*, its diameter should be large enough so that when the maximum torque is brought to bear upon the shaft it should not cause spring or twist sufficient to result in binding between the shaft projections and the sleeve or boss of the wheel. It is well known in general engineering practice that, if the spring is to be taken into account, the diameter found by calculation is larger than it would be if the strength or resistance of the shaft alone were considered, and the motor car manufacturer has also found that gear-box shafting should be designed from considerations of stiffness rather than from those of strength.

Fig. I.

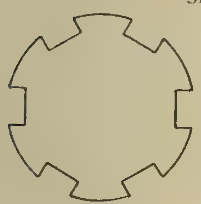


Fig. II.

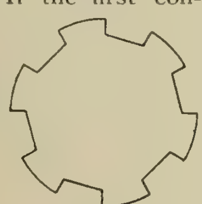
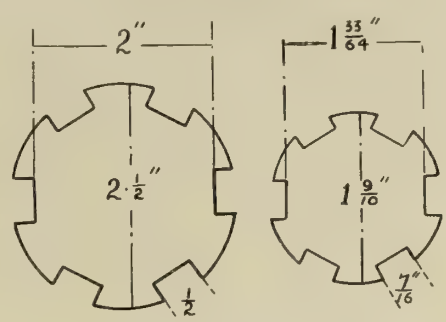


Fig. III.

Motor car manufacturers have found from experience that in some cases shafts have taken a slight set, rendering it difficult to slide the gear sleeves along without considerable friction, and so it is becoming the practice to use comparatively heavy shafts as compared with the power transmitted. The author's experiments have shown that the exact torsional limits of elasticity for shafting are much lower than the figures representing yield points, which are published in works dealing with the strength of material, and thus to some extent account for this heavier shafting, experience has shown to be necessary.

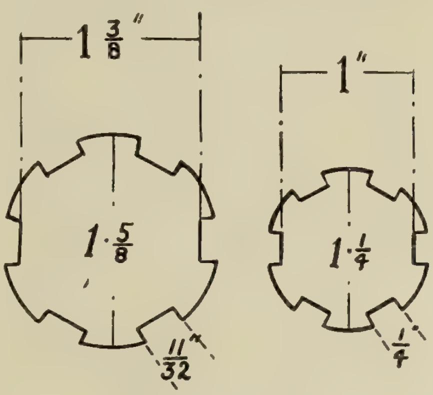
In addition to the torsion, the bending action in the case of gear-box shafting due to the load on the teeth of the wheels is another factor adding to the difficulty of arriving at suitable dimensions of shafting for a car of given power. Though in some cases the effect of the gear sleeves is to stiffen up the shafts to the extent of spreading the bending load, in any case the maximum bending moment as well as the maximum twisting moment should be considered in fixing the dimensions.

With respect to the kind of material used for gear-box and clutch shafting, soft mild steel seems hitherto to have been the favourite, and steel containing about 0.15 of one per cent. carbon is very useful for case hardening, which at the present time seems to be a practice adopted in order to prevent the development of slackness between the shaft and the boss. The various forms of the alloy steels are, however, being used in addition, notably a steel containing about three per cent. of nickel. The elastic limits of such alloy steels are considerably above those of mild steel, and in the case where the material has been oil-hardened or treated, the limit is much above that for the same kind of material when tested in the untreated condition.



Figs. IV. and V.

It will be useful to tabulate for reference the dimensions of six-grooved castellated shafts for some cars of various powers manufactured by high-class firms, at the same time pointing out a great disadvantage which a motor car manufacturer experiences in the lack of standardization of transmission shafting. The figures are given in the following table by way of suggestion only.



Figs. VI. and VII.

TABLE I.

H.P.	15.9	30	35	60	90
Outside Diameter	mm. 35	mm. 41	mm. 44	mm. 48	mm. 48
Diameter at Bottom of Key-ways.	29	35	40	40	40

The desirability of making torsion tests on shafting, both of the plain and castellated forms, before building it up into the motor cars does not yet seem to be fully realised—it is thought by some firms quite sufficient to make occasional tensile tests. First-class firms may, and perhaps do, in the case of crankshafts have a tension test made on a piece of steel cut from each shaft, and as to the wisdom of this course, failing a torsion test, there can be no doubt whatever. The author has tested a large number of specimens cut from motor car crankshafts, and has been considerably surprised at the immense difference in the behaviour of different specimens prepared from the same charge—the results showing both very ductile and very brittle conditions of material. Although comparative results of value are obtained from tension tests of material to be used as shafting, yet undoubtedly results of more practical value would be obtained from the testing of the material for that kind of stressing action to which it is to be subjected in regular work. There is, further, the strong probability that tension and torsion tests will be found in some respects to be complementary.

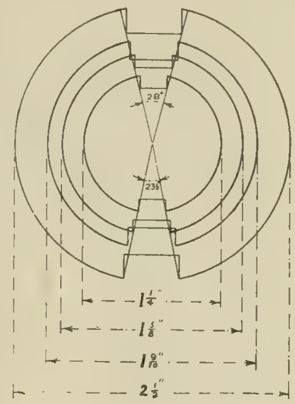


Fig. XII.

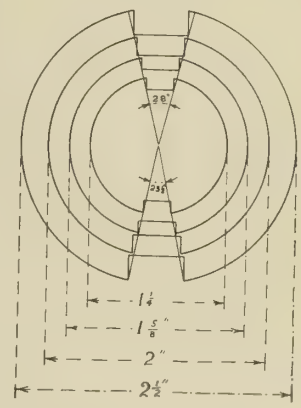


Fig. XIII.
Angle of Key-Ways.

Taking for granted then that the torsion test is more valuable that the tension test in such cases, the question arises as to what results should be given in a report, or required by a specification. The author suggests the following as covering all results of practical importance:—

(1) The torque at the elastic limit and, for cylindrical specimens, the corresponding shear stress at this limit.
(2) The ratio of the torque to the angle of twist for the elastic period together with the angle of torsion at the elastic limit.
(3) The elastic resilience or work per unit length and per unit volume, at the elastic limit of the specimen.
(4) The maximum torque.
(5) The ratio of the torque at the elastic limit to the maximum torque.
(6) The total work per unit length and per unit volume required to twist the specimen to destruction.

(7) The ratio of the total plastic work to the elastic work.
(8) The angle of torsion per unit length and the angle of the helical lines of twist.
(9) The appearance of the fracture.

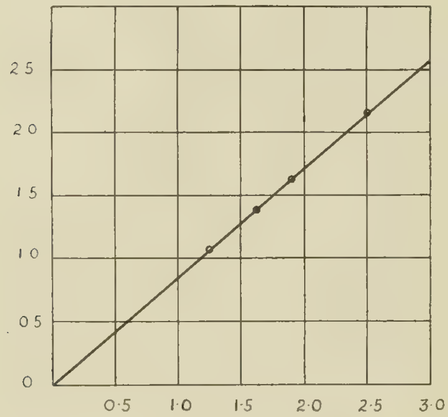


Fig. XIV. Mild Steel.

The author has the honour of presenting to the Institution the results of some torsion tests on two qualities of material, mild and nickel steel; with the former, the tests were made on solid cylindrical and castellated specimens, and with the latter on castellated specimens tested in the normal condition, and after oil-hardening. The complete numerical results for the mild steel specimens are given in Table II., and for the nickel steel specimens, in Table III.

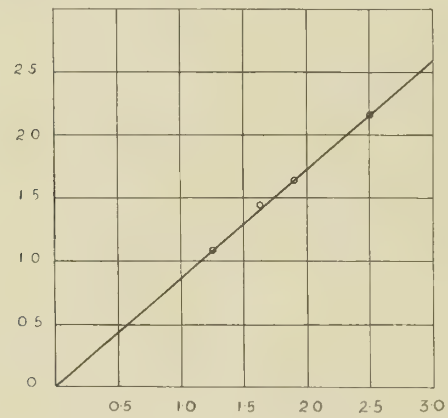


Fig. XIVa. Nickel Steel.

The experiments were carried out in the Mechanical Engineering Testing Laboratory of the Northampton Polytechnic Institute, Clerkenwell, and were made part of the regular laboratory course of work for both day and evening engineering students.

The principal objects of the experiments on the mild steel specimens were to determine (1) the effects produced on the elasticity and strength by castellating or grooving the shafts; (2) the equivalent

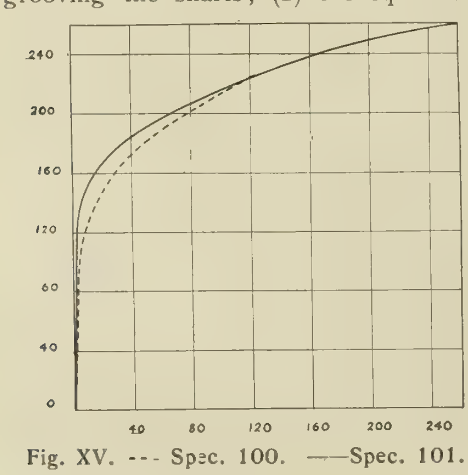


Fig. XV. --- Spec. 100. ——Spec. 101.

diameter of a solid cylindrical shaft which would have the same angle of torsion at the elastic limit as the castellated shaft, or in other words, the diameter of a shaft which would have the same value of the ratio elastic torque to twist; (3) if possible, the diameter of a solid cylindrical shaft which would have the same moment of resistance as the castellated one.

For this series of experiments four pairs of specimens were prepared, one specimen in each pair having six key-ways (see Figs. IV. to VII), while the other specimen was a plain cylinder with a diameter the same as that at the bottom of the key-ways for the castellated shaft (corresponding to Figs. IV. to VII., and hereafter referred to as Figs. VIII. to XI.). The largest castellated shaft was $2\frac{1}{2}$ in. outside diameter with the correspond-

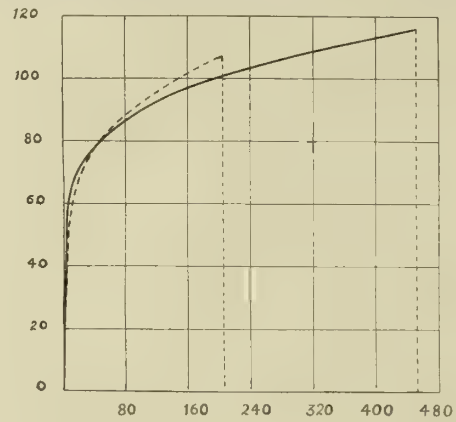


Fig. XVI. --- Spec. 102. ——Spec. 103.

ing plain specimen 2 in. diameter, and the smallest castellated shaft was $1\frac{1}{4}$ in. outside diameter, with the plain specimen 1 in. diameter. The angular extent of the key-ways is shown in Figs. XII and XIII.

As will be seen by reference to the tables, the dimensions given both for the mild steel and the nickel steel specimens fully cover those used for shafting em-

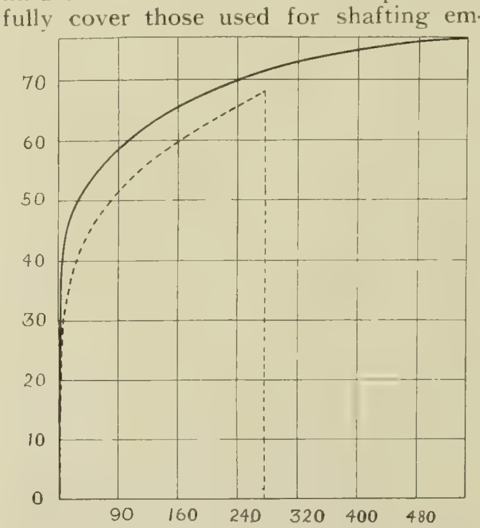


Fig. XVII. --- Spec. 104. ——Spec. 105.

bodied in the construction of a chassis, and this being so it is hoped that the test results will be of real practical value.

Mild Steel Test Results.—Comparing the figures for the several pairs of the elastic limits given in column 7, Table II., it will be seen that in each case the limit for the plain specimen, the diameter of which is the same as that at the bottom of the key-way of the castellated shaft, is somewhat less than that for the castellated specimen, showing, as might be expected, that a little additional strength is afforded to the larger shaft by the projections. This result is borne out, too, by a comparison of the torques at fracture given in column 14, where in each case the torque at fracture for the castellated shaft is greater than that for the corresponding plain shaft. When, however, we come to

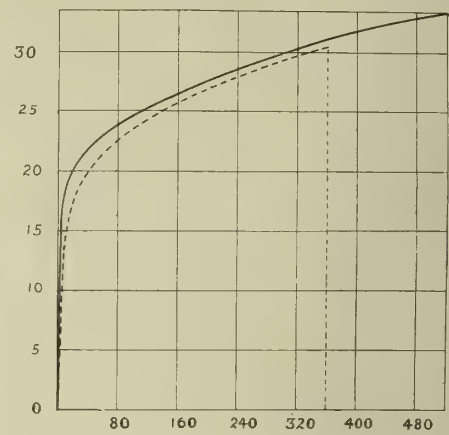


Fig. XVIII. --- Spec. 106. ——Spec. 107.

compare the work required to fracture the material we find that, with one exception, more work is required to fracture the plain specimen than the corresponding castellated specimen, from which it may be inferred that in the ultimate part of the test the castellations become a source of weakness. This is borne out by a study of the

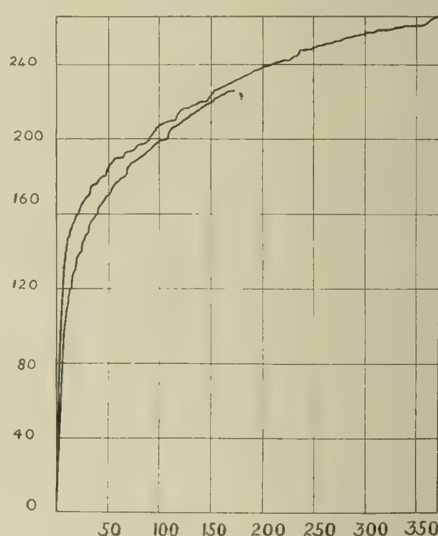


Fig. XIX. Upper line Spec. 101.
Lower line Spec. 100.

fractured specimens (see Figs. XXIX., XXX., XXXIII., and XXXIV.).

In order to find the equivalent elastic shaft or the diameter of a shaft which will give the same spring or angle of torsion as the castellated shaft for values of torque up to the elastic limit, we must take the usual formula for the rigidity modulus for a plain cylindrical shaft, and evaluate for the diameter—substituting in the formula the value of the ratio torque/twist

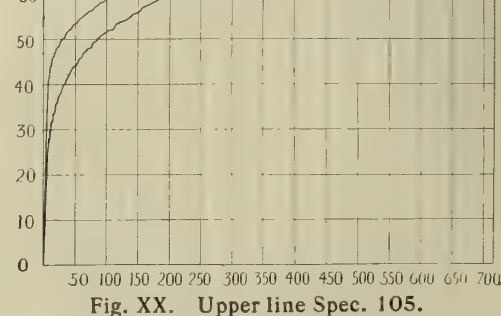


Fig. XX. Upper line Spec. 105.
Lower line Spec. 104.

found experimentally for the castellated shaft, and tabulated in column 8.

This is shown symbolically as follows:

$$G = 584\ T\ L/\theta\ d^4.$$

$$\therefore d_e = \sqrt[4]{584\ L/G \times T/\theta}.$$

The values of T/θ taken are the italic numbers given in column 8, Table II. Comparing the equivalent diameters (column 11) obtained in this way with the inside diameters (column 4) for the castellated shafts, it will be seen that the former are a little greater than the latter. Fig. XIV. gives the plottings for the equivalent diameters (d_e) as ordinates with the corresponding full diameters (D) for the castellated shafts as abscissæ. The equation to the line obtained is—

$$d_e = 0{\cdot}857\ D, \quad \text{or}$$
$$D = 1{\cdot}167\ d_e.$$

These formulæ are true for this series of mild steel specimens, where, as will be seen by reference to Figs. XII. and XIII., all the keys for shafts of different diameters are defined by an angle of twenty-eight degrees. Equations may also be obtained giving the diameters of equivalent solid shafts having the same moments of resistance as the castellated ones.

Nickel Steel Test Results.—Dealing next with the nickel steel specimens and the test results given in Table No. III., it will be noticed that in this series we have four pairs of specimens, those in each pair being of identical dimensions. One of each pair was tested in the condition (except for the machining) in which it was delivered from the forge, while the other was oil-hardened before machining and testing.

Column 7 gives the torques at the elastic limits, and column 10 the elastic resilience, and the first information of practical importance which we derive from an inspection of these test results is that the effect of the oil-hardening has been to raise the elastic limits to about double the values of those of the shafts tested in their normal bar condition, while the resilience or elastic spring energy has been raised to from three to five times that of the untreated shaft. This last result is obviously of very great importance, and worthy of more than passing attention in dealing with the treatment of shafting to be used for motor car work where considerable shocks and wide ranges of stress variations will be experienced.

A reference to column 8 will show that the ratio of elastic torque to elastic twist is practically constant for both treated and untreated specimens. This being so, it follows that the maximum elastic play energy consistent with full recovery of spring on removal of torque is directly proportional to the square of the torque at the elastic limit, and this result is further borne out by the figures referred to and given in columns 7 and 11.

The equivalent diameters of plain cylindrical shafts calculated in the same way as for the mild steel specimens, taking the value of the modulus of rigidity derived from previous experiments, are given in column 9; and the plottings for these equivalent diameters and the full diameters for the castellated shafts are given in Fig. XIVA. The equation to the line obtained is:—

$$d_e = 0{\cdot}867\ D, \quad \text{or}$$
$$D = 1{\cdot}154\ d_e.$$

Similar equations may be found for

TABLE II.—MILD STEEL SPECIMENS *Plain and Castellated.*

No. of Specimen.	Form of Specimen.	Diameter of Specimen. Outside.	Bottom of Keyway.	Particulars of Keyway. Width.	Depth at Edge.	Limit of Elasticity in Pound-Inches.	Ratio of Elastic Torque to Elastic Twist.	Modulus of Rigidity. Pounds per Square Inch	Tons per Square Inch.	Diameter of Equivalent Elastic Shaft.	Elastic Resilience to Limit of Elasticity in Inch-Pounds. Per Unit Length.	Per Unit Volume.	Torque at Fracture in Pound-Inches.	Work to Produce Fracture in Inch-Tons. From Instrument Readings on 8 in.	Per Unit Length.	Per Unit Volume.	Ratio of Total Plastic Work to Elastic Resilience.	Angle of Torsion. By Instrument on 8 in. Length	Per Unit Length.	Angle of Helix.
		in.	in.	in.	in.					in.								deg.	deg.	deg. min.
108	Castellated ..	$2\frac{1}{2}$	2	$\frac{1}{2}$	$\frac{7}{32}$	16100	*53080*	—	—	2 144	5·266	1·251	135000	457·16	57·14	13·58	24300	545	68·1	33 57
109	Plain	2	—	—	—	14800	40000	$11{\cdot}68 \times 10^6$	5215	—	5·970	1·9	90800	438·6	54·83	17·44	25890	740	92·5	41 29
110	Castellated ..	$1\frac{57}{64}$	$1\frac{33}{64}$	$\frac{7}{16}$	$\frac{3}{16}$	7400	*17420*	—	—	1·623	3·429	1·484	57700	257	32·12	13·89	20980	708	88·5	34 17
111	Plain	$1\frac{33}{64}$	—		—	6400	13720	12.01×10^6	5380	—	3·256	1·804	41600	281·5	35·19	19·51	24200	1034	129·25	30 20
112	Castellated ..	$1\frac{5}{8}$	$1\frac{3}{8}$	$\frac{11}{32}$	$\frac{1}{8}$	4430	*9166*	—	—	1·382	2·334	1·283	38000	161·7	20·21	11·10	20490	690	86·25	39 16
113	Plain	$1\frac{11}{32}$	—	—	—	4850	8387	$11{\cdot}79 \times 10^6$	5265	—	3·049	2·127	28100	195·6	24·45	17 07	17960	1040	130	33 9
114	Castellated ..	$1\frac{1}{4}$	1	$\frac{1}{4}$	$\frac{1}{8}$	2750	*3300*	—	—	1·070	2·498	2·39	16550	101·1	12·64	12·11	11330	999	124·9	36 17
115	Plain	1	—	—	—	2680	2568	$11{\cdot}47 \times 10^6$	5120	-	3·048	3·812	11700	122·6	15·32	19·16	11260	1563	195·4	30 9
1	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**	**21**

TABLE III.—NICKEL STEEL SPECIMENS. *Castellated.*

No. of Specimen.	Treatment of Material.	Diameter of Specimen. Outside.	Bottom of Keyway.	Particulars of Keyway. Width.	Depth at Edge.	Limit of Elasticity, in Pound-Inches.	Ratio of Elastic Torque to Elastic Twist.	Diameter of Equivalent Elastic Shaft.	Elastic Resilience to Limit of Elasticity in Inch-Pounds. Per Unit Length.	Per Unit Volume.	Torque at Fracture in Pound-Inches.	Work to produce Fracture in Inch-Tons. From Instrument Readings on 8 in.	Per Unit Length.	Per Unit Volume.	Ratio of Total Plastic Work to Elastic Resilience.	Angle of Torsion. By Instrument on 8in. Length	Per Unit Length.	Angle of Helix.
		in.	in.	in.	in.			in.								deg.	deg.	deg. min
100	Normal	$2\frac{1}{2}$	2	$\frac{1}{2}$	$\frac{7}{32}$	41800	52810	2·162	36·079	8·56	226200	174·09	21·76	5·165	1351	123	15 4	71 26
101	Oil Hardened ..	$2\frac{1}{2}$	2	$\frac{1}{2}$	$\frac{7}{32}$	78100	52080	2·154	127·95	30.39	265200	460	57·5	13·66	1007	268	33.5	53 51
102	Normal	$1\frac{57}{64}$	$1\frac{33}{64}$	$\frac{7}{16}$	$\frac{3}{16}$	23000	17250	1·634	33·43	14·46	107200	140·5	17·56	7·6	1773	204	25·5	67 5
103	Oil Hardened ..	$1\frac{57}{64}$	$1\frac{33}{64}$	$\frac{7}{16}$	$\frac{3}{16}$	39600	17100	1 630	100·022	43·4	116000	370·4	46·3	20·03	1037	452	56 5	46 52
104	Normal	$1\frac{5}{8}$	$1\frac{3}{8}$	$\frac{11}{32}$	$\frac{1}{8}$	11800	10270	1·435	14·783	8·12	68000	117·8	14·72	8·1	2231	276	34·5	63 56
105	Oil Hardened ..	$1\frac{5}{8}$	$1\frac{3}{8}$	$\frac{11}{32}$	$\frac{1}{8}$	26800	10580	1·446	74·023	40·65	77000	289·9	36·24	19·91	1096	545	68 1	46 0
106	Normal	$1\frac{1}{4}$	1	$\frac{1}{4}$	$\frac{1}{8}$	5900	3309	1·081	11·472	10·51	30360	70·6	8·76	8·38	1710	360	45 0	63 52
107	Oil Hardened ..	$1\frac{1}{4}$	1	$\frac{1}{4}$	$\frac{1}{8}$	12200	3329	1·083	48·771	46·6	33400	113·9	14·24	13 62	654	522	65·25	54 34
1	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**

plain cylindrical shafts of equivalent strengths or moments of resistance.

Finally, considering the remainder of the figures given in columns 12 to 19, some interesting results may be gathered from the tests on oil treated specimens as compared with those from the untreated ones. First, the torque to produce fracture is appreciably greater, and the unexpected result that the angles of twist for the treated specimen are of the order of twice the magnitude of those for the untreated ones. (See column 17). These angles are in agreement with the helical angles of torsion in the last column. The increased values of the torque and the angles of torsion together result in more work being required to fracture the oil-hardened specimens than the untreated specimens. These results and the work areas for the four pairs of specimens are represented graphically in Figs. XV. to XVIII. The curves for these diagrams are obtained from the plottings of the torques and twists—the angles being determined by means of a special instrument kept on the specimen throughout the tests in gauge points 8 ins. apart.

Figs. XIX. and XX. are reproductions from the autographic diagrams, and represent the torque-twist curves for the full 12 in. length between the shoulders.

The figures given in column 16, Table III., and in column 18, Table II., are also of importance. For the nickel steel specimens the elastic spring energy varies from about 1/1000 to about 1/2000 of the total work necessary to destroy the bars, which is a much more useful result from a practical point of view than that obtained for the mild steel specimens where only from 1/11000 to about 1/26000 of the total work energy in the bar is available for elastic spring.

Configuration of Material during Testing.—In order to enable the eye to follow the kind of straining action which the specimens had undergone during twisting, the author had the mild steel specimens striped longitudinally on the surface by six black lines equally spaced round the circumference, while circles were painted round the circumference at one inch distances apart along the length. The castellated shafts had the circular lines only on them, the long key-ways being quite sufficient to define the behaviour in the other direction. Figs. XXI. and XXII., which are reproductions from photographs taken after the fractures, give a general idea of the surface flow of the material which has taken place. In order to present a complete idea of the surface behaviour of the material during twisting the author projected on the screen kinematograph representations of some torsion tests. The

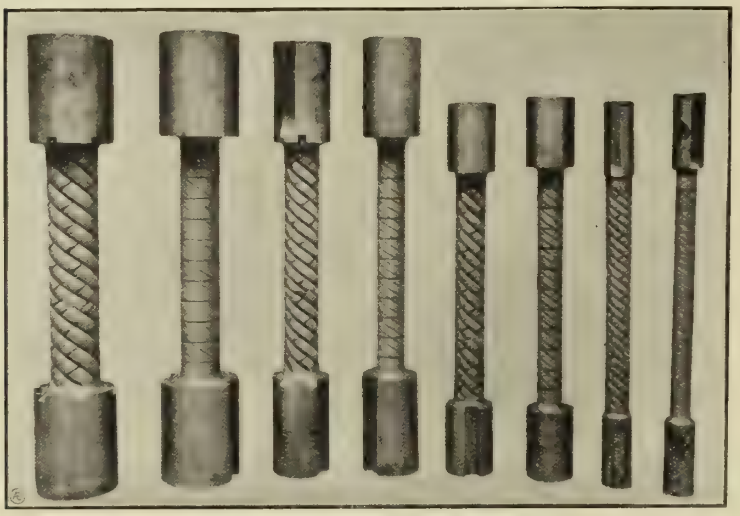


Fig. XXI. Mild Steel Specimens.

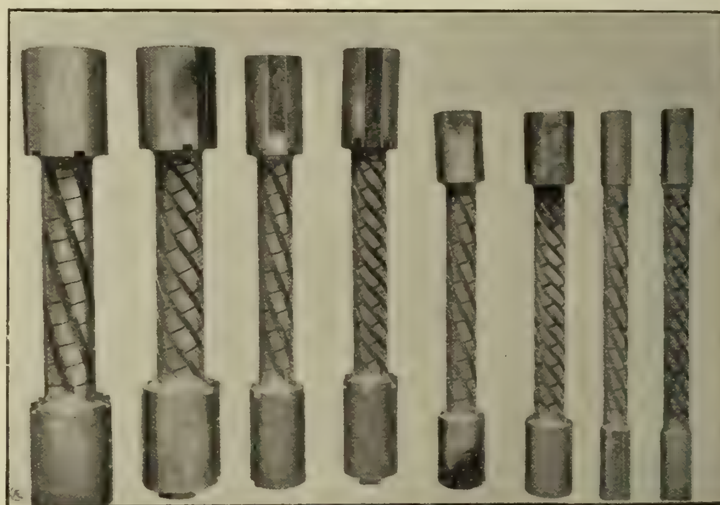


Fig. XXII. Nickel Steel Specimens.

Fig. XXIX. Spec. 100.

Fig. XXX. Spec. 101.

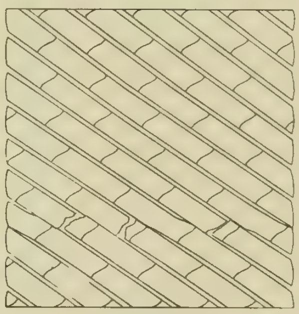


Fig. XXV. Spec. 108.

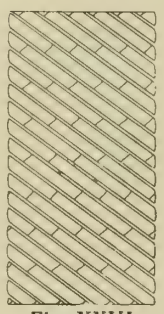


Fig. XXVI. Spec. 114.

Fig. XXXIII. Spec. 104.

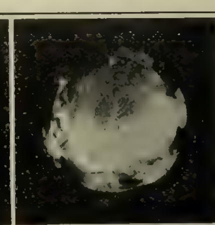


Fig. XXXIV. Spec. 105.

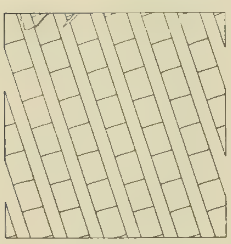


Fig. XXIII. Spec. 100.

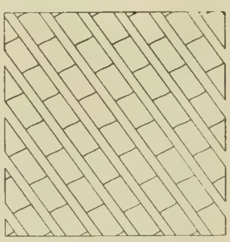


Fig. XXIV. Spec. 101.

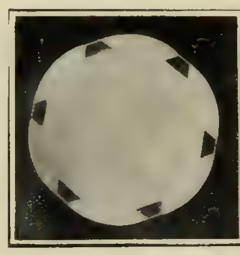


Fig. XXXVII. Spec. 108.

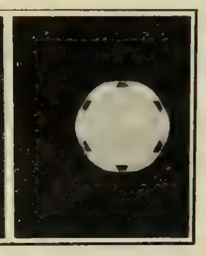


Fig. XXXVIII. Spec. 114.

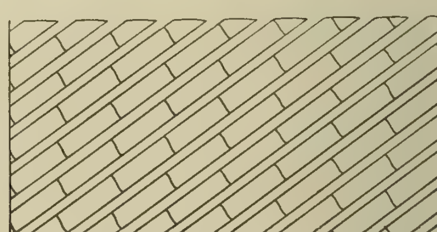


Fig. XXVIII. Spec. 107.

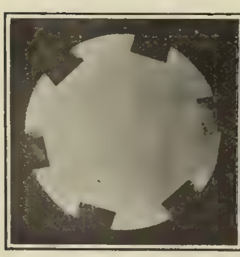


Fig. XXXV. Spec. 100.

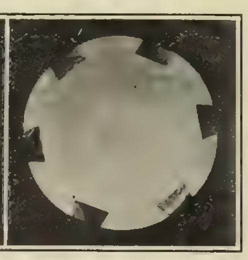


Fig. XXXVI. Spec. 101.

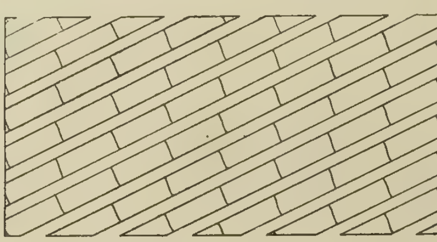


Fig. XXVII. Spec. 106.

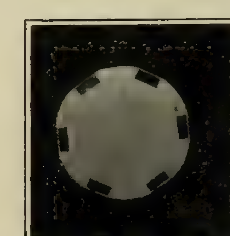


Fig. XXXI. Spec. 106.

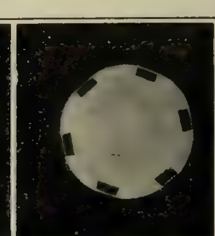


Fig. XXXII. Spec. 107.

first was for a specimen of mild steel $2\frac{1}{2}$ in. diameter twisted to destruction in about twenty-six minutes, the duration of the test being purposely prolonged, while photographs were taken at the average rate of one per second. This corresponds to about 97 feet of photographic films, and about 1,560 photographs. All these pictures were projected on the screen in about a fortieth of the time taken for the test; and gave a good idea of the surface flow of the material during twisting. The author showed a still more striking illustration of the behaviour of a piece of wrought iron of the same diameter when twisted to destruction, the kinematograph magnifying the defects in the structure of the material. It was noticed that the metal seemed to flow in turgid streams.

The ultimate surface configuration of the material for six of the specimens are represented in Figs. XXIII. to XXVIII. by developments, the diagrams being proportioned to represent the circumference of the cylinder compared with the distance apart of the gauge points. Figs. XXI. and XXII., which are reproductions from photographs taken, give a good general idea of the appearance of the specimens after fracture.

A study of the fractures of several of these specimens as shown in Figs. XXIX., XXX., XXXIII., XXXIV., will suggest the methods of failure, while Figs. XXXI., XXXII., and XXXV. to XXXVIII, representing photos from cut and faced specimens after fracture, give a general idea of the flow of the material into the key-ways.

Discussion.

In the course of the discussion Mr. Max Lawrence asked for information as to the percentage of nickel in the nickel steel specimens tested and was informed by the author that it was 3 per cent., but the author was not able to state the carbon content of the mild steel specimens. He emphasised the importance of exactness in the heat treatment of steels which were to be used in the manufacture of high duty parts, such as motor car shafts and castellated shafts, in order to get absolutely the best and most reliable results. From the heat treatment suggested by the author of the paper, and from a cursory inspection of the specimens, he personally thought the results were due to the amount of carbon in the steel, and that the heat treatment had its effect more on the carbon than the nickel. With regard to the strength of castellated shafts and how they failed, that was very interesting indeed. One always imagined how a shaft would fail, and he had always imagined the keyways and the standing up parts being stretched in the manner shown by the tests. The suggestion in the paper that no one had ever tested castellated shafts before was not correct, though perhaps they had not been tested in the same thorough manner. The Wolseley Co. had made tests on smaller shafts, but those tests bore out the results given in the paper. It was a great pity that the specimens were not carbonised, as the effect of carbonising, or putting a very high carbon content into the steel, was very marked so far as increasing its strength was concerned. Mild nickel steel when carbonised to a depth of 1/16th inch had its torsional strength put up from twice to three times beyond what it was previously, and it made the steel behave very differently when it failed. It would be interesting if the author would carry his experiments one step further and try some tests to destruction on carbonised shafting.

Mr. F. L. Martineau mentioned the fact that it was Mr. Lanchester who first used castellated shafts in motor cars and that the motor car industry was indebted to him for many of the ideas embodied in motor car construction. He (Mr. Martineau) some time ago designed a lot of castellated shafts to do certain work, and in order to obviate the use of a large number of milling cutters, and also to increase the strength of the shafts, he came to the conclusion that for varying diameters it was better to increase the number of castellations and take a standard sized cutter which would suit them all. One then got a uniform depth, but a varying number of castellations, and the shaft diameter did not go up at the same rate as the strength went up. He fixed on a cutter which was $\frac{1}{4}$ inch wide as being the most suitable size for the work, and he selected the sizes of the shafts so that the distance between the keyways would be $\frac{1}{4}$ inch also. In that way he was enabled to produce shafts more cheaply. The arrangement was particularly useful, as owing to the depth of the keyways being small it facilitated the fitting of hollow sleeves on the castellated shafts, so that one did not get the effect which was unavoidable with a deep keyway. One could work with sleeves of under 3/16 in. thick, transmitting a very large horse-power and running on a $1\frac{1}{4}$ in. spindle at fairly slow speeds, whereas to transmit the same horse-power with thicker sleeves and deeper keyways was impossible owing to distortion of the circular parts pressing in the bushes.

Mr. Larard, in replying on the discussion, said he had hoped to have received a good deal of practical information to supplement his own results, but he was disappointed. As far as his tests went he had given all the possible results, and although some of them might not be particularly interesting to automobile engineers, others certainly were and it appeared to him that they ought to be taken as a whole.

THE POSSIBILITY OF THE SMALL DOUBLE-ACTING INTERNAL COMBUSTION ENGINE.

By Bertram C. Joy.

It will no doubt be recalled that in almost every case the early internal combustion engine was double acting, and a study of the drawings of some of these engines will make it clear that their design was beyond doubt a direct copy of the contemporary steam engine. One may say, I think, that it was the four-cycle engine, with its compression producing very much higher working pressures and temperatures than had hitherto been the case, that was responsible for the evolution of the single-acting type now almost universal. There is reason to believe that the single-acting principle is not greatly superior to the double—if the two are compared with due regard to all their respective advantages and disadvantages—though I am well aware that there is one serious obstacle to be overcome before the high-speed double-acting engine can become a practical machine; this being, of course, the problem of piston cooling. It may be questioned whether this is an insurmountable difficulty. If it has not been overcome at the present time there does not seem to be any vital reason why the immediate future should not produce a solution.

At first sight it is perhaps not encouraging to learn that even in the early non-compressing engine, when pressures were no higher than twenty or thirty pounds above atmospheric, there was trouble with the over-heating of pistons—explosion taking place, of course, on both sides alternately. This appears to have been overcome satisfactorily by the injection or suction of water into the cylinder, such water being directed against the piston. This cooling system is, of course, not suitable for an automobile engine, so it is instructive to recall that an engine has been built—and, I believe, used in considerable numbers—which was not only double acting, but had a piston entirely free from any water-cooling device. One must not, however, pass from a description of the engine without a qualifying statement to the effect that its working cycle was not a four, but a six cycle, since after the exhaust stroke there followed two others for the purpose of drawing in and expelling a charge of pure air. This was called a scavenging charge, though its purpose must certainly have been more for that of piston cooling than for exhaust residue expulsion. Thus in a single-cylinder engine such as described there would be two power strokes to every three crankshaft revolutions, as against two power strokes to four revolutions for the four-cycle engine. It is clear that if this piston-cooling method were to be adapted to the automobile engine much of the advantage of the double-acting system would be counterbalanced by two extra idle strokes per cycle. Comparing a four-cylinder four-cycle engine with a two-cylinder six cycle, the former, of course, gives six power strokes for three revolutions, and the latter four power strokes in the same number of revolutions.

It seems that it is possible to work a piston and cylinder at a considerably higher temperature than is usual in the automobile engine, and still to retain satisfactory lubrication, for a well-known firm of engine builders are, I believe, at the present time using oil in the cylinder jackets in place of water, the higher boiling point of the former permitting a very

much-increased cylinder wall temperature. One is here reminded that the possibility of the double-acting, high-speed, internal combustion engine is governed largely by the question of the feasibility of efficient lubrication, and it may be that the engineer is, in this respect, to some extent, in the hands of the oil manufacturer, for

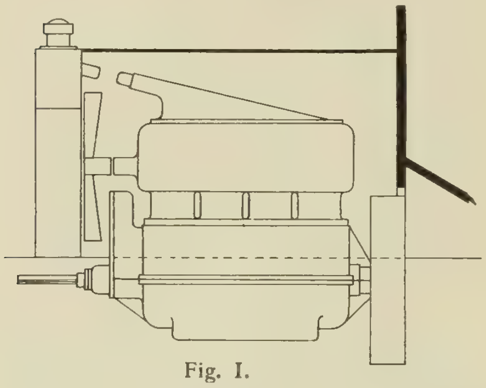


Fig. I.

it should not be difficult to retain a piston, even though attacked by hot gases on both sides, at a temperature sufficiently low to prevent the occurrence of premature ignition.

No mention has been made yet of the hollow water-cooled piston as a method for preventing over-heating troubles. Such pistons are, of course, very commonly used in large explosion engines, especially on the Continent, water for cooling being usually fed to the piston

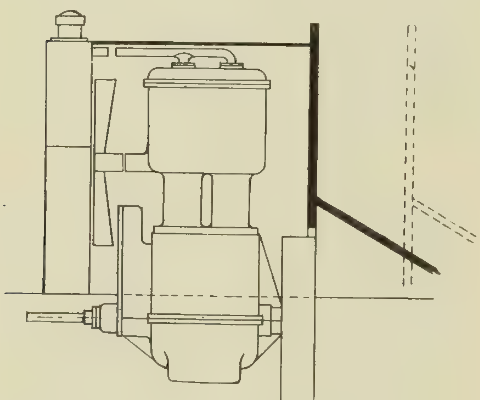


Fig. II.

through a hollow piston rod, and discharged through a tail rod. Water connections such as are fitted to these engines do not tend to much complication in an engine having a cylinder perhaps twenty inches in diameter, but the case is very different when pistons of only three or four inches in diameter have to be water cooled. It is difficult to see precisely how the necessary flexible connections could be made and maintained water tight for any reasonable length of time when the piston reciprocations are, perhaps, at the rate of a thousand or more per minute.

I do not imagine that piston rods would be the cause of much trouble as far as over-heating is concerned, for at the moment of ignition, and maximum flame temperature, only a short length of rod need be exposed, while the transmitted heat ought to be disposed of easily by conduction along the rod to the cross-head guides, etc. At the same time, it will be obvious that the piston rod would, to some extent, have to serve as a heat conveyor to the piston, so a fairly large diameter rod might be advantageous—considerably larger probably than would be justified on considerations of strength only. The stuffing-box difficulty has been overcome most successfully in the case of large engines by metallic packings. There should be no difficulty in applying these packings to rods even as small as $\frac{7}{8}$ inches diameter, and I am myself aware of one such application which has proved itself to be reasonably satisfactory in action, under the ordinary working conditions of an internal combustion engine.

It will now be well to turn to actual comparative dimensions of the single and the double-acting types, and, in order that the comparison may be a fair one, engines giving an equal number of explosions per revolution must be pitted one against the other. A four-cylinder four-cycle engine of usual design is therefore equivalent to a two-cylinder, four-cycle, double-acting engine, and, assuming that the cylinder bore and stroke are equal, the two engines should develop equal power at a given number of revolutions per minute. It is obvious that there is a considerable saving in length with the double-acting engine, and Figs. I. and II. are intended to give a fair idea of the extent of this saving. The two diagrams are drawn to the same scale, and the dotted lines in Fig. II. show the saving attributable to a double-acting engine. It may be argued that there is no very great advantage secured by this saving of length, but I fancy that the nine or ten inches gained in a four-inch engine could be put to excellent use by the body maker, while the length saved in the case of the six-cylinder engine would naturally be more pronounced. Equally obvious with the saving of length is the increase of height which the method of design compels, but the additional height is but an insignificant drawback, as, even in the existing type of engine, there are usually several inches to spare between the cylinder tops and the bonnet. There would be little objection, too, I think, even regarding the matter from the æsthetic point of view, to an increase of a few inches in bonnet height. The real disadvantages are that the height of the cylinder renders thermo syphon cooling not quite easy of application, especially if the radiator is forward of the engine, and that if the moving parts are not well balanced the vibration set up in a high engine might be more serious than it would be in the case of a considerably lower engine, supposing it to be equally in balance. Fig. III. shows comparatively the heights of the single and double-action engine. It should be noted that the minimum length necessary for the stuffing box is about $2\frac{1}{4}$ins., and that the length of piston rod necessary accounts for the remainder of the difference between the respective heights of the two engines.

Now a few words as regards relative cost of production of the two types. The double-acting type scores considerably as far as its crankshaft is concerned, for it will have but two throws against the four throws of the four-cylinder engine, and the cost should therefore come out at but little more than half, although it would probably need to be larger on account of the alternating stress. The cylinders, seeing that there will be two only and that these will be of no greater length than required for the usual type of engine, should reduce cost still further, though parts such as stuffing boxes, for instance, will add something to cost price. The double-acting engine requires two pistons only and two connecting rods, but, on the other hand, extra parts are essential, such as piston rods, cross-heads, and guides. Some expense should be saved in the crank chamber, which need be little more than half the length required for a four-cylinder engine, while assuming poppet valves to be the type of gas distributor employed in each case, it seems reasonable to expect that a comparison of cost should show no surplus on either side. As a general estimate I surmise that the cost of building a two-cylinder, double-acting engine should work out at 15% or 20% cheaper than the cost of a four-

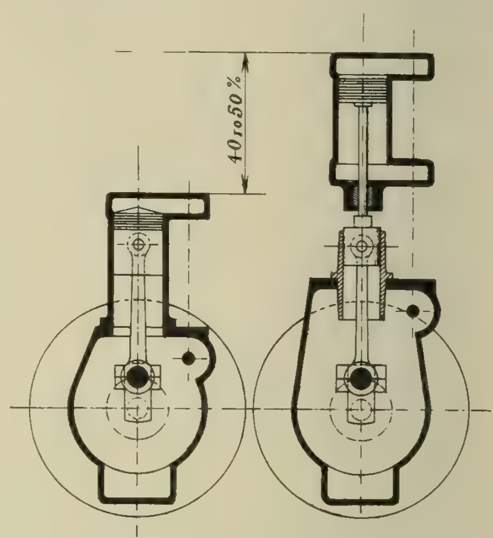


Fig. III.

cylinder, single-acting engine of equal power, or, at any rate, of approximately equivalent horse power rating

A number of further points occur to me as worthy to be reckoned with, but I will confine myself to one or two only. In the matter of cylinder wear then, should not there be an advantage on the side of the double-action engine? There being no side thrust on the cylinder walls from connecting-rod angularity, the wear is confined to that caused by the pressure of the piston rings, and this can only be slight; true, the wear takes place on the cross-head guide, but this can be worked at a lower temperature, and can be, if desired, flooded with oil, which a cylinder wall most emphatically cannot. On the other hand, the balancing of a twin-cylinder engine is not so easy to perform efficiently as is that of a four-cylinder of the ordinary type.

The conclusion one arrives at after a fairly thorough investigation of the pros and cons of the double-acting engine in comparison with the standard type of high-speed engine is that the somewhat formidable problem of piston cooling is responsible, more than any of the considerations mentioned above, for the conspicuous absence of the double-acting type of petrol engine as applied to automobiles, or, for the matter of that, to any other purpose.

When one considers the tremendous difficulties which have already been overcome, it seems most improbable that this single problem is so hard to solve that it can prevent the evolution of a double-acting engine, suitable for automobile purposes. If it should ever become obvious that there was a real need for a small motor of this type it may be regarded as almost certain that a successful design would not be long in making its appearance.

THE SOCIETY OF AUTOMOBILE ENGINEERS.

Extracts from three of the most interesting Papers given at the Meeting held in New York during January, 1911.

Novelties in Valve Systems.

By EUGENE P. BATZELL.

In the writer's paper, read at the S.A.E. summer meeting, a number of different types of valve arrangements brought forth to compete with the poppet valve were represented and discussed. During the last few months there have appeared some new valve systems, comprising among them also several constructions, embodying in their valve gear a combination of valve types formerly known and applied separately. In regard to these there can be expressed a presumption that inventors found it hard to work the field seeking new original valve types, and started to combine older ones in one or the other way. As a result, the constructions became more and more complicated, and to redeem this it is necessary to make claims as to their great superiority over the other simpler arrangements. Only a few of such combination valve systems

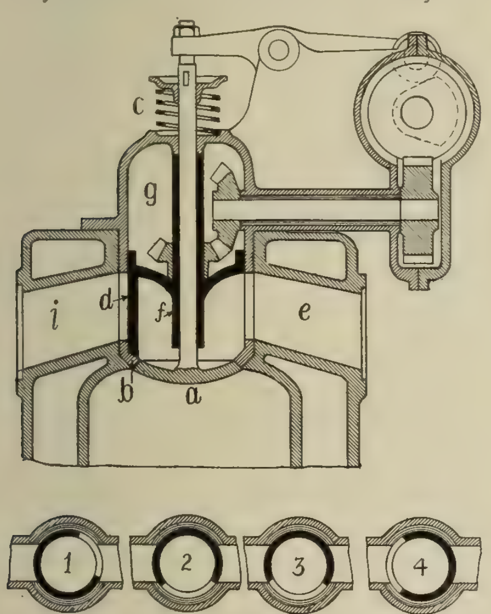


Fig. I.

are represented below, but it does not make it a hard problem to put different valve types together in some other manner.

COMBINATION ROTARY AND POPPET VALVE.

Probably the most interesting of the combination valve systems is represented in Fig. I. This system has been recently described in a number of trade papers, but for completeness a short description of it is also desirable here. It is a combination of rotary and poppet valve. The single poppet valve *a* is held to its seat *b* by the spring *c*. Inside the seat *b* is located the rotary valve *d* of cylindrical form, the central part *f* of which serves as guide for the poppet valve stem *g*. The cylinder inlet port is *i* and the exhaust *e*. The rest of the drawing clearly shows the method employed to secure combined operation of the poppet and the rotary valve. The timing of both valves is represented below, the exhaust duration being 231 degrees and that of the inlet 198 degrees of crank-shaft movement. The port in the rotary valve extends about 114 degrees over its circumference, and the inlet and exhaust ports in the cylinder head each occupy about 80 degrees.

The claims made in favour of this system, as published, are as follows:

1. A single poppet valve of large size will insure ample opening with a minimum lift.
2. The induction and exhaust passages in the cylinder head can also be made of very ample area.
3. These two facts insure a rapid discharge of exhaust, resulting in a cool running engine, and similarly an easy flow and good charge of mixture are obtained on the induction stroke.
4. The rotary valve is entirely shielded from the compression and explosion pressures in the cylinder. The only pressure to which it is subjected is the slight negative pressure during the suction stroke. As lubrication of the rotary valve does not seem to present any difficulties, there ought to be no wear to speak of.
5. As the poppet valve remains open twice as long as is the case with the ordinary arrangement of two for each cylinder, and as the lift is less than half, it follows that the action must be much quieter and sweeter, and that wear and tear will be lessened accordingly.

Analysis of Claims.

In regard to these claims the following analysis can be made: To get the full advantage of port size in this system, the minimum lift of the poppet valve will have to be determined under consideration that the greatest openings of both valves be equal. It can be assumed that a motor with a 5-inch bore has a rotary valve of $3\frac{1}{2}$-inch outside diameter and a poppet valve with $3\frac{1}{2}$-inch clear diameter. The cylinder port openings which register with the rotary valve might be of rectangular section 1 inch high and extending, as said, over 80 degrees, having an area of about 3 square inches. Consequently, the poppet valve has to have a lift of about 5/16 inch. Again, to get all the advantage from a large size of valve opening, the poppet valve should not start to close the inlet before the moment when the cylinder port has been completely uncovered by the rotary valve. Referring to position 3 in Fig. I., stating that the rotary valve opens the induction, and considering that 80 degrees of its motion is required to attain the maximum port size, it will be found that during but $114^\circ - 80^\circ = 34^\circ$, counting half crank-shaft time, the poppet valve should close. On the other hand, in an ordinary poppet valve motor with the same opening duration, the corresponding time of valve closing will be $44\frac{1}{2}$ degrees, or about 30 per cent. of the total time slower. This rejects the claim made as to quieter and smoother valve action.

The shape of the respective inlet opening curve *a* in Fig. II. is of a different character, comparing with valve systems described in the writer's former paper. The curve *a* is unsymmetrical, reaching its maximum quite late. Its rise is more gradual than with some other systems, but its drop is quicker. Concerning the inlet, a reversed form of curve should be preferred.

Less unfavourable is the exhaust. It is started by opening the poppet valve, which reaches its full lift during about $35\frac{1}{2}$ degrees of half-time motion, as compared with $57\frac{3}{4}$ degrees for an ordinary poppet valve. This quick exhaust opening really allows a quick and free escape of the burned gases. But with proper cam shape a similar quick opening, although of smaller total size, could be obtained in an ordinary poppet valve motor also.

The above-mentioned induces one to believe somewhat differently from what is stated in the claims accompanying the construction. However, some of its good features cannot be denied, and it might prove to be very reliable if applied in practice. A proper development of the oiling system for the rotary valve is not to be solved easily in such a manner as to prevent the oil from entering into the motor cylinder and the gas passages. A particularly bad feature of the system is the leading of intake and exhaust gases through the same chamber which is formed inside the rotary valve. In the foregoing example the volume of the chamber is about 9 cubic inches. With the motor having a 6-inch stroke, its piston

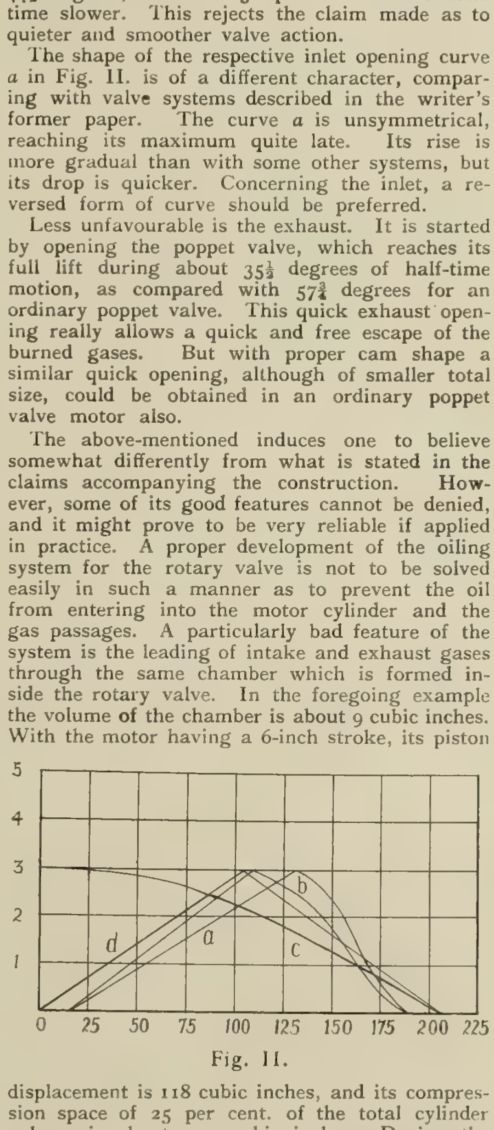


Fig. II.

displacement is 118 cubic inches, and its compression space of 25 per cent. of the total cylinder volume is about 39.5 cubic inches. During the induction stroke the 9 cubic inches of spent gases left in the valve chamber from the previous exhaust will be carried into the cylinder, thus increasing about 22 per cent. the amount of spent gases contained in the explosive mixture. After the inlet the space of 9 cubic inches is filled up by fresh gases, which are expelled wastefully during the following exhaust. For the chosen size of motor this will increase its fuel consumption about 10 per cent.

If it were not for the cited disadvantages in connection with a large chamber inside the rotary valve, the general valve action of the motor could be improved over that of the example taken. For instance, the same maximum size of 3-square inch inlet opening could be obtained with a cylinder port $1\frac{1}{2}$ inches high extending only 70 degrees over the rotary valve circumference. Retaining the 114 degrees extension of the port through the rotary valve, the difference $114^\circ - 70^\circ = 44^\circ$ would be left as time of duration for the poppet valve to close. This figure is about equal to that of an ordinary poppet valve. The shape of inlet opening curve also would be somewhat improved—*b* in Fig. II. However, these improvements are connected with an increase to 13 cubic inches of the chamber inside the rotary

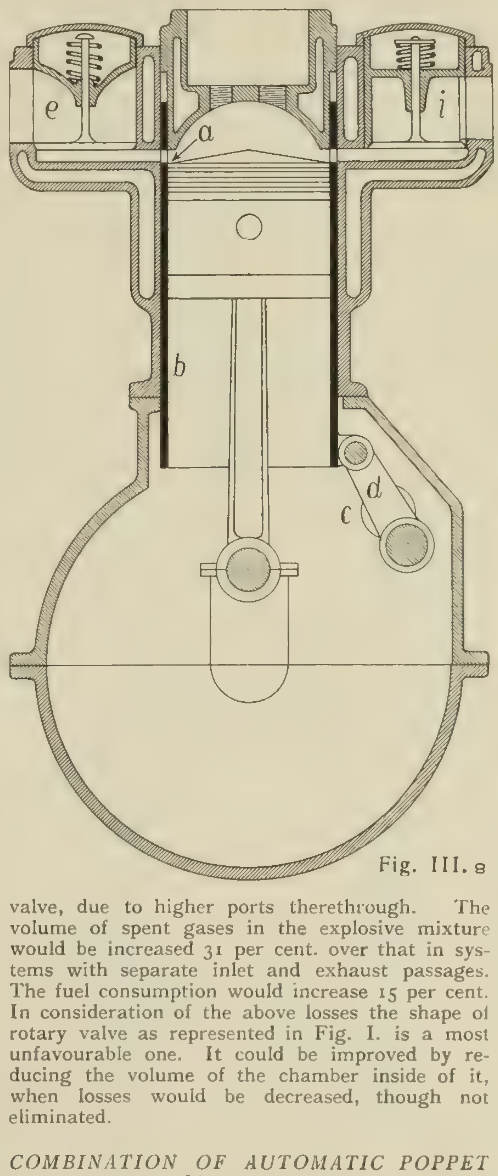


Fig. III.

valve, due to higher ports therethrough. The volume of spent gases in the explosive mixture would be increased 31 per cent. over that in systems with separate inlet and exhaust passages. The fuel consumption would increase 15 per cent. In consideration of the above losses the shape of rotary valve as represented in Fig. I. is a most unfavourable one. It could be improved by reducing the volume of the chamber inside of it, when losses would be decreased, though not eliminated.

COMBINATION OF AUTOMATIC POPPET AND SLEEVE VALVES.

In Fig. III. another valve combination is represented, comprising automatically acting inlet and exhaust poppet valves and a single sleeve valve. The action of it is as follows: At the beginning of the motor suction stroke ports *a* in the sleeve valve *b* are open and gas is admitted into the cylinder through the automatic inlet valve *i*. During the induction period the sleeve *b* starts an upward motion, being operated from a half-time shaft *c* by means of a connecting rod *d*. At the end of induction the ports *a* become hidden in the cylinder head, which cuts off this period. The sleeve *b* continues its motion up and afterwards down so as to start again to uncover ports

a at the time of exhaust beginning. The pressure inside the cylinder forces the exhaust valve *e* open, allowing the gases to escape during the whole scavenging stroke. The alleged idea of this construction is to shield the poppet valves from high pressure and temperature and to get large valve openings together with simplicity of mechanism.

The opening character cannot be analysed exactly on account of the automatic valve action.

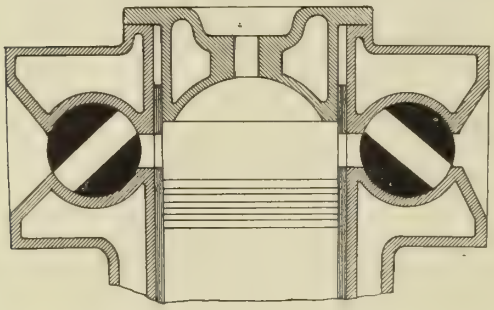


Fig. IV.

Some figures relating to the sleeve valve timing are of interest. If its ports close the inlet with a 20 degree lag, and open the exhaust 40 degrees early, then the time between is 300 degrees, counting upon the crankshaft. The combined inlet and exhaust time will be the balance of 720 degrees or 420 degrees, during which the valve port *a* remains open. Its maximum opening is reached at the middle of this period, viz., 210 degrees after exhaust beginning. With an exhaust lasting 220 degrees it will be found that the sleeve *b* has travelled 10 degrees on its upward motion at the time when the inlet should start to open, respectively, when the piston starts its downward stroke. Thus the size of inlet opening given by the sleeve ports will decrease with the continued suction stroke. On the contrary, the automatic inlet valve *i* opens more and more due to increase of suction, caused by the rising piston velocity.

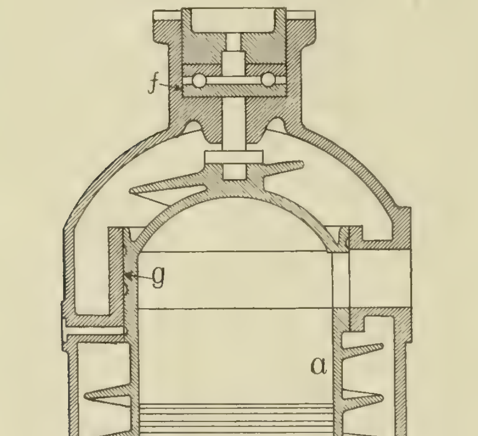


Fig. V.

Consequently there will be a moment somewhere between the extreme piston positions, when the port areas of both valves reach an equal figure. This port area has to be considered as the maximum for the inlet. Contrary to the figuring of the inventor, this maximum port area will be much less than the area of ports *a* through the sleeve, and generally the automatic inlet valve does not open quickly, and the less quickly it opens the less will be the inlet port area maximum. Its value will be variable, depending on the conditions under which the motor works.

Against the provision made for the exhaust in the above construction, it can be mentioned that the automatic exhaust valve leaves a certain pressure above atmospheric inside the cylinder, after completion of the scavenging stroke. Investigating this point by means of a common indicator diagram, it will be found that with increase of this remaining pressure is required a longer piston travel on the suction stroke before the fresh charge can be taken into the cylinder. This results in a decrease of fresh charge quantity drawn in, together with an increase of the relative amount of spent gases mixed with it. The explosive mixture will be bad and the motor will have a low efficiency.

COMBINED ROTARY AND SLEEVE VALVES.

The valve arrangement represented in Fig. IV. differs from that of Fig. III. only in that rotary or oscillating barrel valves are used in place of the automatic inlet and exhaust valves of the latter. Let the change of sleeve valve port size during the inlet be represented by curve *c* in Fig. II., and the lines *d* in the same figure correspond to the opening of the inlet rotary valve *i* in Fig. IV. The real shape of the opening curve with this valve combination will be represented by the starting part of line *d* until its intersection with curve *c*, after which it follows the latter. This resulting combination opening curve is shown in Fig. II. by heavy lines. Its insufficient shape proportion is easily noticed, particularly if compared with that of the rotary valve, and needs no further explanation.

Sleeve Combination Unsuccessful.

Neither Fig. III. nor Fig. IV. arrangements can be considered successful in any respect. The sleeve valve generally carries many strong objections against its use, and thus these objections become applicable also against the above constructions. The advantage of a favourable valve opening, which is being obtained with common rotary valves, is lost in the construction of Fig. IV. on the greatest part of the induction stroke. Mechanical complication is increased. As to the main object of these constructions, the shielding of the outer valves, this appears to be solved entirely in the wrong way. Poppet valves with a proper seat and stem cooling are more apt to withstand high temperatures than the sleeve valve. In regard to rotary valves this still cannot be stated positively, although even the little work done with this type of valve induces one to believe them not less reliable than the sleeve valve. Properly constructed they will do the work under any heat conditions at present used in connection with automobile motors.

ROTARY CYLINDER MOTOR.

The very curious construction shown in Fig. V. hardly appears to be practical, but it is being claimed for it that it not only works but works well. Even believing this, it would be interesting to see some figures as to the quantity of oil consumed. The cylinder *a* is provided with valve ports in its top part and revolves itself, whereby the ports (or port) register with inlet and exhaust passages located through the surrounding water jacket. The cylinder rotates at one-quarter crank-shaft speed, which is rendered possible by using double inlet and exhaust ports. Its drive is through a worm and wheel and can be understood from the drawing. The top of the cylinder fits on a slight taper *g* into the adjoining part of the water jacket housing. Oil under pressure of 6 pounds per square inch is led to this taper and also to the lower packing *b* as shown. The cylinder *a* has spiral fins outside, which should assist the water circulation. The oil being led to places as indicated, besides serving as lubricant, should also prevent the jacket water from penetrating into the crank-case through the packing *b*, and into the cylinder through the taper joint *g*. For the latter purposes the oil should be supplied in abundance, so that its flow overpowers the tendency of the water to penetrate into the joints, on their whole circumference. Thus it can be anticipated that a great part of the oil supply is wasted into the water jacket, the rest of it being burned and carried away by the exhaust gases.

It is also being claimed for the construction of Fig. V. that the taper joint *g* of the cylinder head is self-regulating, because the spring *d* has a tendency to push the whole cylinder upward. On the other hand, the ball thrust bearing *f* in the top takes up this spring pressure and also the explosion pressure acting from the cylinder inside against its top. Such being the case, the wear in the joint *g* can be taken up only by hand adjusting bearing *f*, because it is hard to presume that the wear in this bearing will equal that in the joint.

Fig. VI. represents a French sleeve valve construction, which has been actually built and tried and is said to be successful. The valve *a* is split lengthwise in *b*, allowing it a slight spring to keep tight the cylinder inside. The inlet port *i* and the exhaust port *e* are located one above the other, at both sides of the place adjoining the split *b*. The valve is operated through a lever *d*

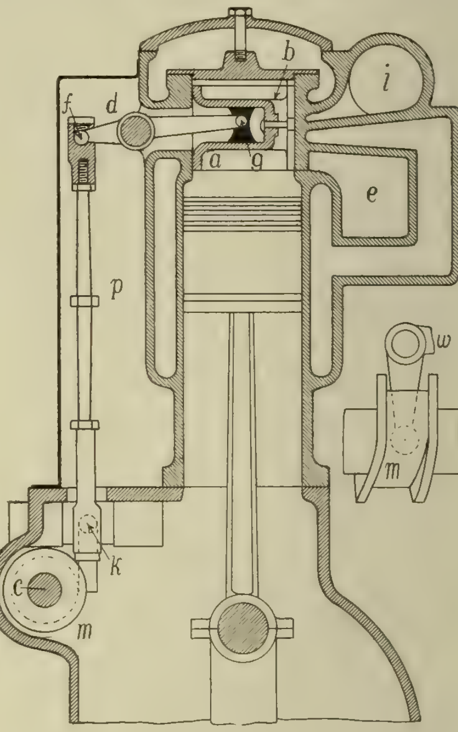


Fig. VI.

with sliding blocks *g* on one end of it and a ball joint *f* on the other. The motion is transmitted from the crank-shaft to a half-time shaft *c*, provided with a cam *m*, in which operates the bell crank *w*. This latter is also connected with a ball joint *k* to the lower end of the adjustable rod *p*, acting on the lever *d*.

The construction is very complicated, with many joints and other places subjected to wear. Though this wear can be taken up in some places, there are others bound to have lost motion. The valve openings can be made of a very favourable character by using the cam *m* of proper shape. However, the start and end of the opening have to be made more or less lingering; otherwise the sudden motion impulse will create great strains in the operating mechanism, which has a number of weak points.

CONCLUSION.

In conclusion the writer wishes to repeat his opinion that those valve systems which should and which will compete successfully with the poppet valve ought to be the least complicated mechanically. The combining of different valve systems with the intention to create an advantageous construction fails even in its avowed purpose. Moreover, in most cases the separate use of valve types, entering the combination system, ought to be better than the latter. Some further development of the individual valve types has to be accomplished before employing of combination systems can be justified, even as a means of getting around some doubtful points of the former.

Leaf Springs.

By E. K. ROWLAND.

THE object for which vehicle springs are used is first and mainly to prevent shocks which arise from irregularities of road surfaces from reaching the passengers, and secondly, to prevent damage to the vehicle. The modern spring dates from 1804, when a patent covering elliptic springs was granted Obadiah Elliott.

Until the advent of the motor car the vehicle spring was divided, roughly speaking, into those for passenger carriages and those for transportation of merchandise, the main requisite of the first being ease of ride, and of the second strength. When the motor became a commercial possibility, the demand arose for springs combining ease and strength, which demand the spring-maker has en-

deavoured to supply with more or less success.

Laws of Strength, Deflection and Resilience.

The laws relating to the strength, deflection and resilience of springs are that the strength of a spring varies—

(1) Directly as the breadth of its plates. That is, double the breadth and you double the strength;

(2) As the square of the thickness of its plates. That is, double the thickness and you increase the strength four times;

(3) Inversely as its length. That is, double the length and you halve the strength;

(4) Directly as the number of plates. That is, the plates being all the same thickness, the strength of the spring is the strength of its main plate multiplied by the number of its plates.

The deflection of a spring plate under the load it will bear without taking a permanent set varies—

(5) As the square of its length. That is, double the length and you increase the deflection four times;

(6) Inversely as its thickness. That is, double the thickness and you halve the possible deflection;

(7) As the load. That is, double the load and you double the deflection.

The resilience of a spring varies—

(8) Directly as its weight or volume.

The rate at which a spring oscillates under any safe load varies—

(9) Inversely as the square root of the deflection produced by that load, or, for a given spring, inversely as the square root of the load.

Using the well-known formulæ of Reauleaux, a spring may be constructed to carry with safety any given load, but at this point spring designing ceases to be a science and becomes an art, for the reason that the features of construction which produce ease of ride are covered by no formula; under this head I include the number of plates, their thickness and length, the length of the taper of each plate and resilience.

If one endeavours to check up the majority of motor springs in use it will be found that these obvious laws and standard formulæ are more honoured in the breach than in the observance, and that resilience is almost universally neglected as a factor in designing.

Material.

Assuming that a spring has been properly designed, the selection of proper material is of great importance, and the spring-maker is limited to three materials—chrome-vanadium, silico-manganese, and what, for want of a better name, may be called open-hearth steel of the standard French analysis, of which the following is the composition:—

Carbon	.60
Manganese	.70
Sulphur	.025 to .04
Phosphorus	.07
Silicon	.20% to .30%

Any of these materials properly heated possesses to a marked degree the necessary qualifications of a good spring steel, that is, high tensile and elastic limits with great elongation and reduction. Of these steels I unhesitatingly recommend chrome-vanadium as vastly superior to the other two, silico-manganese being a fairly good second, the greatest point of superiority in chrome-vanadium lying in its great resistance to fatigue. I have not included chrome-nickel as a suitable material, on account of its well-known inability to stand up in service.

Admitting that proper design and material have been secured, a good spring cannot be made unless the heat treatment is absolutely correct, and this can only be secured by the proper use of pyrometers, which must be frequently calibrated and checked. I will not go into detail of the chemical composition of alloy steel, as it would be but a repetition of Mr. Henry Souther's article on "Specifications for Materials, with Notes and Instructions Thereon," issued to the members of the Society last July. In this connection I can do no better than to quote from Mr. Souther's article.

"In connection with the manufacture of springs the heat treatment is of the utmost importance. Plain carbon steel properly heat-treated will give splendid results if the spring be properly designed, and, more important yet, properly attached to the axle. Any spring that is not properly attached to the axle will give trouble by breakage.

"The treatment of steel by spring-makers at this time is, as a rule, bad. Old and inexact methods good enough for horse vehicles are still in vogue, and they are of such a character as not to create any uniformity of physical condition in the several leaves of the spring. In some cases the physical condition of the metal is different in the various parts of the length of a given leaf. This condition should not exist in a spring any more than in a transmission gear. The difficulty is that a spring is long and thin, rendering it difficult to have all parts of the length uniformly heated or uniformly cooled."

As an example of spring design I submit the following specifications for a standard seven-passenger touring car, weighing between 3,750 lb. to 4,000 lb. The dimensions given in all cases are under load.

Front Springs.—$2\frac{1}{4}$ in. wide, 38 in. long; the right spring to be of 150 lb. greater capacity than the left, as it has to take the torque of the motor.

Rear Springs.—Three-quarter scroll elliptic $2\frac{1}{4}$ in. or $2\frac{1}{2}$ in. wide, the lower member to be 52 in. long, off-centered, not over 2 in., and to have an inside opening of not over 2 in.; the top member to be from 20 in. to 22 in. to centre.

Seats.—In seating the springs on the axle I strongly recommend a soft metal seat, although it is possible to obtain good results by the use of a leather pad, not exceeding 1/16 in. in thickness; heavier leather or wood has a tendency to crush in service, increasing breakage.

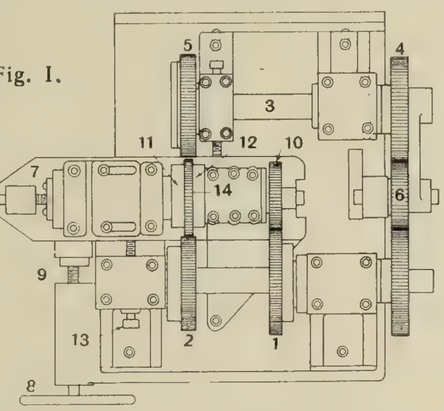
Clips.—The spring clips should be of ample size, and of a better material than ordinarily used. And the importance of keeping the clips tight cannot be over-estimated. This last fact should be impressed on the purchaser of a car, and he should be advised that at intervals of two to three months the springs should be lubricated between the leaves.

Hot Rolled Gears.

By H. N. ANDERSON.

NEARLY four years ago, when the automobile industry was expanding by leaps and bounds, and the capacity of the machine tools was the limiting feature, my attention was called to the fact that the shortage of the gear cutting machinery seemed to be the greatest, and at that time I conceived the idea of rolling the teeth in the rough blank hot, which idea upon investigation I found was not new, but had never been developed to a practical point.

Fig. I.



The knurling process was always used, that is, either the die was driven and the blank allowed to turn from its contact with the die, or the blank driven and the die allowed to turn from contact. The idea of rolling gear teeth in a blank first originated with John Comly as early as 1872, who shows the die driven and the blank rotating by contact. F. A. Brun, a Frenchman, in 1905, showed a different construction; the blank being driven by frictional contact from conical rolls, and the die allowed to turn from contact with the blank. Brun took care of the surplus metal by the adjustment of the conical rolls, the idea being to force this extra metal down into the web of the gears. Chas. H. Logue, who has written a treatise on gearing, experimented with rolling gears; but his method was to take the blank and notch or rough it on a milling or hobbing machine, which gave the correct number of teeth and spacing; then taking the blank, heating it, and rolling the teeth to form. These operations would be more expensive than cutting the teeth to size in the first place.

I then built a small machine and experimented with rolling lead blanks, as the cold lead would flow the same as steel at a forging heat and required but a small amount of pressure. This showed conclusively that the tooth would form itself correctly. A large experimental machine was then built and steel gears were rolled successfully.

THE MACHINE.

The machine in itself is very simple, as shown in Fig. I. On the end of shaft 3 is the breakdown or roughing gear 5. On the opposite shaft is the finishing gear 2; also a timing gear 1, which drives gear 10. This gear 10 is the same size as the blank to be rolled. This blank, marked 14, is held between two chucks, 11 and 12, which are opened and closed by a screw. The whole carriage 7, carrying the chuck and gear, is pivoted on the pivot in line with edge of gear 10, and the carriage is oscillated by means of screw and handwheel 9.

The process is as follows: Blank 14 is inserted and the carriage is thrown over toward breakdown gear 5, which does the roughing work. This operation is very interesting, as the blank to be rolled is a little over the pitch diameter of the gear, the metal being broken up by this breakdown gear and "flowed" out to a larger diameter. The blank is then brought into contact with finishing gear 2, and carried up until the carriage reaches the stop mark 13, then the proper depth of tooth is reached and also the proper diameter; at the same time the surplus metal thrown out on the end of the teeth is trimmed off by a cutter by the movement of the carriage by a screw, operated by hand wheel or lever, which brings the edge of the gear 14 in contact with the cutter. The gear is kept in this machine until it takes a permanent set, and is pushed off the holding arbour by a stripping device which cannot distort it.

The advantages of the process are:—

Cheapness, as the whole periphery of the gear is rolled in one or two heats, depending on the size of the blank. (A gear is in the machine for a period of not over forty-five seconds.)

A much stronger tooth, caused by the increased density of the metal, as each tooth is practically forged by an enormous side pressure on each flank.

A generated and developed tooth.

The tendency to warp in case-hardening is a great deal less than with a cut gear, as the structure of the metal at the periphery is changed while hot and there are no internal strains to be relieved, as is the case of the cut gear.

Any alloy steel gear, the blank of which can be drop-forged, can be rolled. With some silico-manganese steels which are practically impossible to machine, the blanks can have the teeth rolled and ground afterwards if desired; the hole being ground, making all operations forging and grinding.

As the gears are held at the periphery and the hole is bored afterwards on a chuck which chucks from the pitch line of the teeth, the hole must be concentric with the pitch diameter, eliminating the possibility of a sprung arbour cutting an eccentric gear.

The rolling process applies to bevel as well as spur gears, in which case the cost is decreased far more. In roughing a bevel with this process, the sides of the teeth have the true curve, with an evenly added amount of finish allowance. This is in contrast to the ordinary roughing machine, which cuts a straight side on the teeth; the generating machine consequently having to remove more stock in rounding the corners than on the pitch line.

All gears where high accuracy is desired should have a finishing cut, but there is no question but that, with more development of this process, gears can be used in the lower-priced cars without any finishing cuts being required.

THE EFFICIENCY OF A TWO-CYCLE ENGINE.

We greatly regret that in the report of the paper on the thermal efficiency of a two-cycle petrol engine by Dr. Watson and Mr. R. W. Fenning, which was published last month, several errors occurred in the diagrams on page 239, the scales shown being in several cases too open, thereby showing pressure valves for various stages of the cycle which are much too low. As this, to some extent, affects the interest and value of the diagrams, we shall be pleased to supply a corrected set to any of our readers who care to apply for it.

ALFRED HERBERT, LTD., have sent us a calendar for 1911 containing a number of excellent photographic illustrations of their machines.

CHAS. CHURCHILL AND CO., LTD., have recently issued a calendar for the present year, which is illustrated with a selection of machine tools, mostly of automobile interest.

CORRESPONDENCE.

No responsibility is accepted by the Editor for the opinions of correspondents, and the right is reserved to omit any portion of a letter if it is considered desirable to do so. Letters to be inserted with a nom de plume must be accompanied by the name and address of the writer. If publication in the next issue is desired letters should reach the Editor in no case later than the 20th of the preceding month.

HORSE-POWER RATING.

Sir,—I enclose herewith two diagrams which may be of some interest to your readers, in the matter of "Horse-power Rating."

Diagram I. consists of a number of curves which have been plotted according to the various h.p. formulæ set out in the accompanying list.

The ordinates represent the ratios of stroke/bore; the abscissae the comparative power ratings (the power of an engine with stroke/bore in every case being taken as unity). The slope

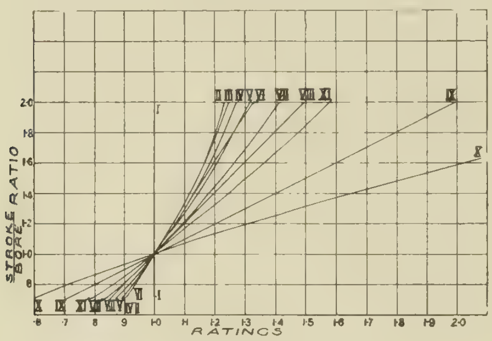


Diagram I.

of each curve therefore depends only on the value of the factors containing S in the corresponding formula.

This diagram is instructive as showing the tendency of the various formulæ to favour or punish particular stroke/bore ratios.

Diagram II. consists of curves plotted according to the same formulæ as those of Diagram I., but applied to a 4-cylinder engine having a bore of 4in. and a stroke varying from 2.8 to 8in. The ordinates represent the stroke in inches, the abscissae the h.p. As the location of a curve is dependant on the value ascribed to the symbol k, and the slope depends on the location (see curves VI. and VIa.), this diagram is of less general interest, it being simply an application of Diagram I. to a particular series of cases.

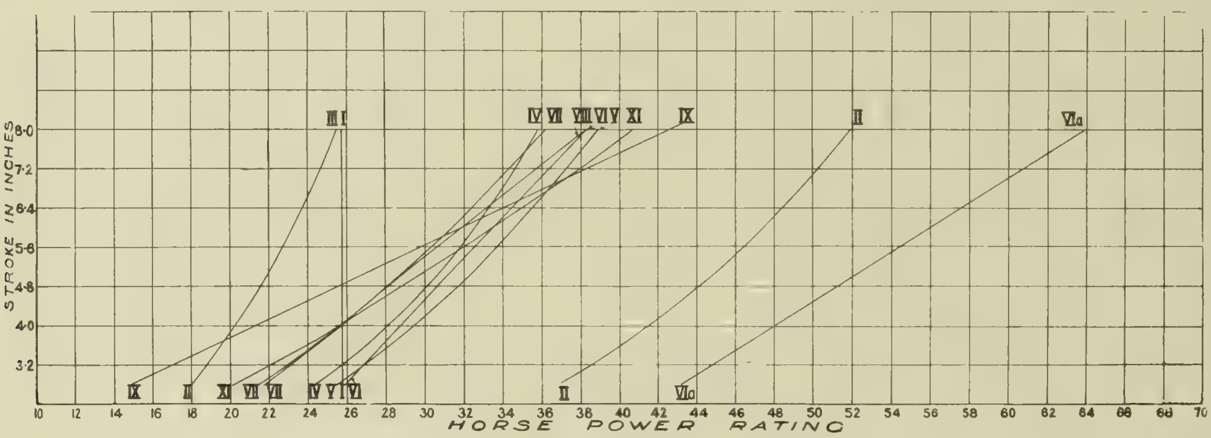


Diagram II.

Formulae.

I. R.A.C. $\cdot 4 . D^2 . N$.

II. Automobile Engineer (Lanchester) : $1000\ D^2 . N . \left(\frac{2R+1}{R+2}\right)$

III. Remington (Bourne) : $\cdot 2 . D^2 \sqrt[3]{S} . N$.

IV. Lanchester (a) : $\cdot 5 \frac{D^2}{\sqrt{\left(\frac{D}{S}+\cdot 3\right)}} \cdot N$.

V. O'Gorman (Lanchester) : $\cdot 4 . D^{1 \cdot 6} . S^{\cdot 4} N$.

VI. S.M.M.T.: $\cdot 2 . D . N . (D-1) . (R+2)$.

VIa. ,, : $\cdot 33 . D . N . (D-1) . (R+2)$.

VII. Lanchester (b) : $\cdot 4 . D^{1 \cdot 5} . S^{\cdot 5} N$.

VIII. Henderson (Lanchester) : $\cdot 2 . D . N . (S+D)$.

IX. Dendy Marshall : $\frac{D^2 . S . N .}{12}$

X. Thornycroft : $K . D^2 . S^{1 \cdot 5} . N$.

XI. Henderson : $\cdot 4\, D^{\frac{4}{3}} . S^{\frac{2}{3}} . N$.

With regard to Prof. Henderson's letter in a previous issue, where he says that "the R.A.C. formula will give a very close approximation at a piston speed of 1,000 feet per minute, and that at any other piston speed the h.p. is simply proportional to the speed," I should like to give two practical examples to show that, even when thus corrected, this formula is still entirely unsatisfactory :—

12-14 h.p. Crossley. Here the h.p. by R.A.C. formula is 15.8. In its bench tests the engine develops 29.5 h.p. at 2,000 revs. per. min., which is equivalent to a piston speed of 1,570 feet per minute. Applying the above-mentioned correction we get 24.8 h.p. which is about 20% too low.

28 h.p. Lanchester. Here the R.A.C. rating is 38.4. I have no definite statistics of bench tests by me, but the maximum h.p. which the engines are tested up to in the works is, I believe, about 43-45. The highest speed I have had the engine turning at is 2,800 revs. per min. (65 miles per hour) which is equivalent to a piston speed of 1,400 feet per min. Applying the correction for the piston speed we get 54 h.p., which again is about 20% out (too high in this case).

With regard to Professor Henderson's remark *re* the suitability of the R.A.C. formula for taxation purposes : Even the most infatuated member of the government or government official cannot be under the impression that the speed limit for motor cars is observed otherwise than when the driver is aware of the presence (visible or invisible) of a policeman. Would it not, therefore, be absurd to base taxation on the correctness of an assumption the falseness of which is common knowledge?

I find that I have made an arithmetical error in my letter, printed in your December number, in the h.p. value of the 28 Lanchester car by S.M.M.T. formula. It is there given as 41.5, while it should be 39.6.

Yours truly,

C. A. BRANTSEN.

Sir,—In the rating of petrol engines Lanchester's rule giving the weight of the reciprocating parts, viz., $D^3(1+0.3r)$, is based on the usual formulæ for proportioning those parts. But on certain hypothesis, which may be held to be quite reasonable, the weight could vary as $D^3(1+ar^2)$, where a is a constant. This, of course, would result in a different rule from that given by Lanchester for power. However, let us take his formula, $.4D^3\sqrt{r}$, and compare two engines of widely different proportions.

An engine, A, having a cylinder diameter$=1$, stroke$=1$, and $r=1$, would have the same power and inertia stresses as an engine, B, having cylinder diameter$=.92$, stroke$=1.5$, and $r=1.63$. Additional particulars are given in the appended table, in which proportionate figures are given.

The R.A.C. formula, namely $.4D^2$, it will be seen, gives a rating for engine B considerably less than it should do; and, as is well known, the effect of its application for taxation purposes has been to increase r.

The respective durability, cost and bulk, of the two engines is of interest to the manufacturer and the user :—

Assume that the wear is proportional to the product of the velocity of rubbing and the pressure per square inch of rubbing surfaces; that the length of main bearings is proportional to the diameter of the cylinder, and that the length of

Engine.	A	B
Rating by Lanchester Rule	1	1
Rating by R.A.C. Rule ..	1	0 85
Revs. and Power Per Unit Volume of Cylinder ..	1	0·785
Piston Speed	1	1·18
Durability	1	1·28
Cost prop. to S × D ..	1	1 38
Cost prop. to S × D² ..	1	1·27
Bulk	1	1·27

the piston is proportional to the stroke. Also assume that the wear of the surfaces is not affected by the alternating application and relief of pressure at those surfaces during the revolution. It can easily be shown that the wear follows the same law for cylinder, piston and bearings, and is proportional to $D^{\frac{2}{3}}/S^{\frac{2}{3}}$ or durability varies as $\frac{S^{\frac{2}{3}}}{D^{\frac{2}{3}}}$ or as $r^{\frac{2}{3}}$

The cost might be assumed to be influenced by both the volume of metal in the engine and the amount of machinery, say approximately varying, as (m SD^2 + n SD), where m and n are constants. The table gives the results.

The bulk of the engines may be taken as roughly proportional to the volume of the cylinder.

The conclusion arrived at if Lanchester's rule is correct for all cases (which it cannot be, owing to justifiable differences of opinion as to the design of the piston and the connecting rod) are as follows: The R.A.C. formula is in error by about 15%, as between the particular cases considered; and the long stroke engine is more bulky, costly and durable, by about 27%. Professor Callendar's correction would make the long stroke engine 4½% less powerful, while the difference in economy would probably be greater than this figure.

It seems that no simple formula based on proportions alone will be found that will give a fair comparison between different engines until designers all agree to use the same formulæ in design. The latter possibility is not likely to be attained. There is no doubt, however, than any rule to be adopted should not influence design in the unscientific manner of the R.A.C. rule of thumb.

W. ALEXANDER,
A.M.I.C.E., A.M.I.M.E.

SINGLE MODEL SPECIALISATION.

Sir,—I have read with great interest your article in "Automobile Engineering for 1911," on the progress of automobile construction in 1910. It is an article which one cannot disagree with to any extent, but I feel that you have left entirely untouched a very important factor in the production of a really good motor car at a reasonable price.

I have a fair knowledge of the French manufacturers, and a good one of both English and American, and the thing which strikes me most forcibly is the attempt on the part of many English manufacturers to get the whole market by having a very large range of models. It is of course

impossible for one maker to supply everyone, and if—as is done in the States—the manufacturer would concentrate on one model, we should be much further along the road towards the production of the really good small powered car at a reasonable price.

It is a self-evident fact to anyone who really attempts to manufacture and does not really build, that the question of cost is very largely dependent on the quantity of one particular article which can be put through at one time. It is also evident that a motor car works with a production of five to seven models, even though its total sales be large, cannot produce parts in sufficient quantities for them to be placed on a really manufacturing basis. Further, the amount of design work which is necessary is enormously increased, and in every department the cost of supervision is bound to go up.

In so complicated a mechanism as a motor car there are quite sufficient different processes introduced in making one size to keep any works fully employed, and it is along the lines of one car for one works that the greatest improvement in the future is bound to come.

Anyone knowing American cars well is quite ready to admit that in many cases they do not in any way compare with the English car, but at the same time in the class of car I am speaking of the prices are also in no way comparable. When a car has the following specifications—4-cylinder engine, 2 speed and reverse epicyclic gears, 3 passenger body, hood, wind screen, two gas lamps, generator and speedometer, 32 by 3in. tyres—and is sold at a price in the States of 680 dollars, it seems to me almost marvellous that such a vehicle should run at all, whereas one knows that there are many cars being built at prices of from 500 to 1,200 dollars complete with a specification similar to the above, and these cars certainly do run, and run well for a time.

I have had under my personal observation a car of this type which has run rather more than three years, it has done in the neighbourhood of 14,000 miles, and has cost—exclusive of depreciation—some £50 a year to run, this including insurance, licenses, petrol, oil and repairs. The driver is a doctor, who, although he is a careful man, is in no way a mechanic, and the car has nothing done to it except cleaning down and a periodic visit to a near-by garage.

Taking into consideration that cars at this almost incredibly cheap price can, and do, run—although they may have a short life—it seems to me that a British manufacturer with the best market in the world for raw material and machine tools, should be able to build a good car at let us say £200 complete. Such a car would of necessity have to have a rigid specification, which could not be changed to suit customers, but I know from personal experience, were it not for the sometimes justified prejudice against American cheap goods, I have literally hundreds of acquaintances who would buy one of these American cars which range in the neighbourhood of £180 to £200.

I therefore feel that the British manufacturer is doing an injustice to himself and to his trade by attempting to grab the whole market in all sizes of cars, instead of sticking strictly to one model and getting the utmost out of this one model, both as regards high quality and low price.

COSMOPOLIS.

BALL AND ROLLER BEARINGS.

Sir,—The letter of your correspondent, Mr. R. F. Hall, appears to be simply an evasion of the objections we raised to the extreme statements made in his previous letter. Therein he gives certain figures which ostensibly are relative areas of contacts and load capacities pertaining to ball bearings of different designs and taper roller bearings, but when it is pointed out that the premises upon which he seeks to build his arguments are essentially incorrect, he wishes your readers to understand that his former figures were not in accordance with the statement to which we take exception, but in some way or another make a liberal allowance for the inaccuracy of his statement.

We have, of course, no answer to this, but would simply enquire if it is safe to infer that a similar liberal allowance has been made for each of the different types of bearing illustrated, including, of course, the roller bearing, and if so and the figures given are not reliable, why were they published at all?

Exactly what Mr. Hall means when he says that "it would have been wiser to have challenged the figures and not the statement," we do not know; we certainly intended to dispute both the statement and the figures he based upon the statement, and we have no hesitation in repeating that to our knowledge, and we submit that it is fairly extensive, it is *not* common practice to make the race curvature of ball bearings of the annular type with a radius equal to .7 of the ball diameter, nor with a radius even approaching that figure.

This is of much greater importance than Mr. Hall apparently is willing to admit, but we think it must be obvious to all that the area of contact depends upon the radius of the curve of the race and is a maximum when the curvature of the race and balls coincide, so that if the figure given for the radius of this curve is incorrect the whole of the figures that follow must be incorrect too, and it is exactly upon this ground that we reject Mr. Hall's conclusions.

It is, of course, quite possible that when the annular form of bearing was first manufactured, seven or eight years ago, the race curve may have been made .7 of the ball diameter, as this was a very common figure for thrust and cup and cone bearings, but since that time experiments have been continually carried out with a view to finding out the best possible combination; *apropos* of which the following quotation from a paper read before a German Engineering Society by an engineer who had himself extensively experimented in this direction, may be helpful. He says: "As the flat surface commences to curve it is evident that the load capacity increases, but not the friction; this, as a matter of fact, decreases in proportion to the load capacity, because this increases at a rapid rate. As the curve of the surface is increased, the load capacity also correspondingly increases up to a certain limit without increase of friction.

"The most efficient form of curve section for the outer race is:—

$$r = .56d,$$

and for the inner race:—

$$r = .52d."$$

We are not able to state definitely that these proportions are used by all manufacturers, but they may be accepted as more nearly the "common practice" than the figure given Mr. Hall, and we may add that the same experiments have shown that the relative load capacities of a cylindrical race (that is, without any groove) and a race with a groove made to the proportions given above are as 1 to 4; therefore, as the cylindrical race is better adapted to carrying a radial load than the conical race illustrated at (*a*) in Fig. I., the figures given in Table II. for the annular type of bearing should be approximately 500% instead of 175%, as given by Mr. Hall.

It would be an easy matter to point out other conditions, more particularly connected with manufacturing problems, not favourable to roller bearings in a comparison between roller and ball bearings, if it were our purpose to attack this form of bearing. We have no desire to do this, however, but simply wish to enforce the manifest folly of any reasoning which pretends to show that the old cup and cone form of cycle bearing is equal in load carrying capacity to the annular form of bearing.

To turn now to the question of adjustability, Mr. Hall does not distinguish between his own statement and those made by ourselves; we simply took up his admission concerning the wear of a bearing being confined to approximately 25% of the circumference of the stationary member to confirm our argument that since all bearings must wear in the same manner, whether ball or roller bearings, it is not only undesirable, but impossible to adjust a worn bearing; undesirable because the highly finished surface has been destroyed upon which the efficiency of the bearing depends, and impossible because such wear is local, but now Mr. Hall says in regard to roller bearings, "it is quite clear that no such local wear could take place" obviously, therefore, no wear at all would take place, and if this were the ground taken by Mr. Hall we could heartily sympathise with him, because it is exactly the ground we take in regard to ball bearings when properly applied and protected. But that this is not the ground he takes is manifest, since so much importance is attached to the supposed ability of adjustment possessed by the taper roller bearing.

Referring to the last paragraph of our previous letter, Mr. Hall seems to have entirely missed the point we wish to emphasize, which was not only the fact that the bearings had successfully covered the distance stated, but that when removed from the front hubs they were to all intents and purposes in a condition equal to new and were certainly good for a further 45,000 miles. It is not surprising, however, that similar results are not always obtained, for the market is flooded by bearings that are cheap in every sense of the word, except in regard to the mileage obtained from them, and so long as motor car manufacturers continue to fit such bearings results similar to those mentioned by Mr. Hall are to be expected. If such results were obtained, however, when first-class bearings of English manufacture were fitted, it would indicate in a most decisive manner, not that the bearings were insufficient for the work, but that the car itself was faulty either in design or construction.

Mr. Hall may have disposed of our previous letter to his own satisfaction, but not, we think, to the satisfaction of any of your readers who will take the trouble to carefully examine the problem. That the taper roller bearing will carry a greater load than the annular ball bearing we freely admit, but that this greater load carrying capacity is required in motor car work, or is as great as he seeks to show, we emphatically deny, and would strongly endorse the closing paragraph of Mr. J. V. Pugh's letter in your last issue, "that keeping the bearings absolutely full of a suitable lubricant, and keeping the water (and every other extraneous substance) out, are of more importance with any reasonable design than all other considerations put together," provided, of course, that no permanent end thrust is put upon the bearings in assembling.

Given greater attention to these points by both motor car designers and constructors, and by users in seeing that the provision made is always maintained in an efficient condition, very little, if any, trouble would be experienced from front wheel or any other bearings wearing.

THE AUTO MACHINERY CO., LTD.

THE H. ENGINE.

The peculiar engine bearing the above-mentioned title, which was described briefly in the December issue of *The Automobile Engineer*, has lately been running on a test stand, for demonstration purposes and, though no brake tests have been made of an accurate nature, it has furnished proof that it develops a reasonable amount of power, while the sliding crosshead which connects the cross bar of the piston pairs to the crank pin does not heat up or show signs of undue wear.

An examination of the engine under working conditions makes the manufacturing advantages very obvious. By means of a detachable head on one side the four cylinders and the central crank case could be one comparatively simple casting, the crankshaft is, of course, cheap, and four cams only suffice for valve operation. As there are no small-end bearings the pistons can be light and cheap as well, and the whole engine can be assembled in a remarkably short space of time. Starting is easy, and the engine is capable of running at a fairly high speed, though the vibration is very noticeable, especially in the event of one cylinder misfiring. It will be remembered that two of the cylinders are head downwards (though it is claimed that the engine will run in any position), and it is a curious fact that the lower cylinders do not appear to suffer from over-lubrication. The chief advantage seems to be the undoubted low manufacturing cost, considered on a horse-power basis, while others are light weight, compactness, and the accessibility of the small number of parts. On the other hand, the most obvious disadvantages are the considerable vibration owing to the combined weight of the four pistons and their connecting link acting like a single large piston in a single-cylinder engine, the rather awkward shape for car work and the possibility of the crosshead proving not too durable, though time only can show whether this will be so.

We believe that the engine is suggested as being specially suitable for powerful but quite small and cheap cars and for boats, where it would be placed with the cylinders horizontal, and would therefore occupy very little useful space. It might also have possibilities for aero work if the vibration difficulty can be overcome. We understand that a car will soon be completed fitted with a small engine from which something like 15 h.p. is expected, and it will be interesting to observe the results of the experiment.

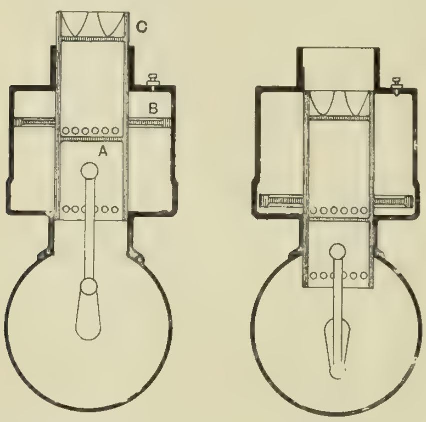
A SAMPLE OF ALUMINIUM foundry work has reached us from R. W. Coan. It is a distinctly handsome specimen of clean casting, and is also a useful design.

RECENT AUTOMOBILE PATENTS.

By Eric W. Walford, F.C.I.P.A.

A Two-stroke Engine.

The piston is formed with a trunk portion provided with a web A and an exterior disc part B, the disc part constituting the pressure surface. Various port openings are provided, as illustrated. With the piston at the top, ignition is effected by the sparking plug, forcing down the disc B and piston. Prior to this the upward movement of the piston has created a vacuum below the disc B eventually, when the parts are in the right position, drawing gas into the crank chamber below the disc B and into the upper part of the hollow trunk piston. The downward movement of the piston due to ignition compresses this gas in the space below the disc B, and eventually the piston reaches a point when the ports or slots C become uncovered

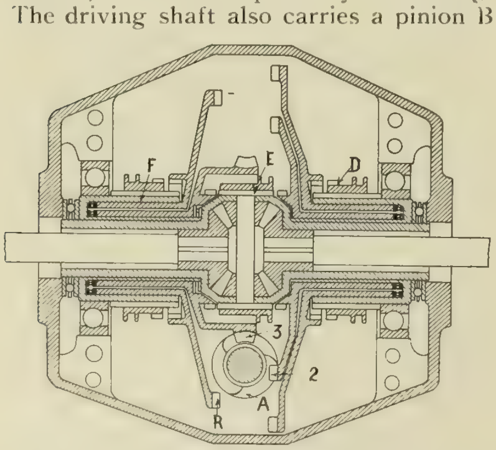


and exhaust takes place from the top of the combustion chamber. The disc B has now reached a part of the cylinder of slightly larger diameter, so that the compressed gas can get round the disc and fill the cylinder from the opposite end, driving out the exhaust gases.

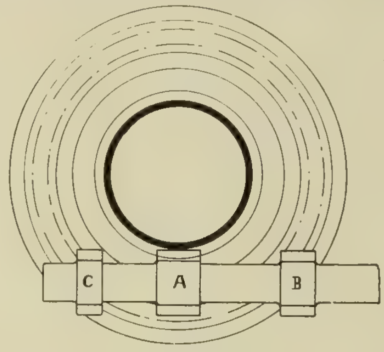
Société des Moteurs Gnome. No. 13,710/10.

Worm Change-Speed Gearing.

This gear, which is mounted upon and forms part of the back axle, provides a direct drive on all three forward speeds and also on the reverse. The driving shaft carries a worm pinion A, which meshes with the second and third speed wheels, marked respectively 2 and 3. The driving shaft also carries a pinion B



which meshes with the first speed wheel and another pinion C which gears with the back of the reverse wheel R. It will be understood that all the driven worm wheels revolve freely on the differential casing, and any one can be clutched thereto by means of sliding clutch members D E F. It will be seen that there is no relative motion between any worm

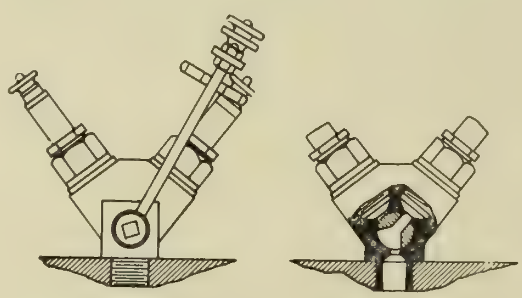


wheel and its bearing parts when it is transmitting power.

M. Buch. No. 20,938/10.

A Sparking Plug Switch.

A single body casting is provided to receive two sparking plugs, the body screwing into the usual sparking plug hole. A two-way plug cock is provided whereby only one of these can communicate with the cylinder at one time, and the plug has attached to it a bridle which carries a high tension wire and a clip engaging the sparking plug terminal. Thus as the bridle can be moved from the one to the other position either sparking plug can be used, and that previously in use can be not only switched off, but is completely insulated from the cylinder, so that it does not become fouled. A construction of this nature may be valuable for racing, aeronautical and motor

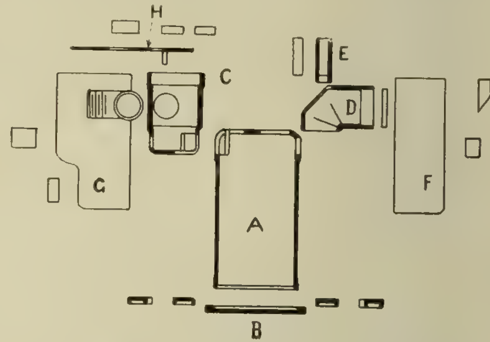


boat work, but general experience proves that it is not good practice to locate the sparking plug at the end of a passage of small bore, so that the construction illustrated may possibly have its disadvantages in spite of the excellence of the idea.

A. Christmas. No. 10,842/10.

Pressed Steel Cylinder Construction.

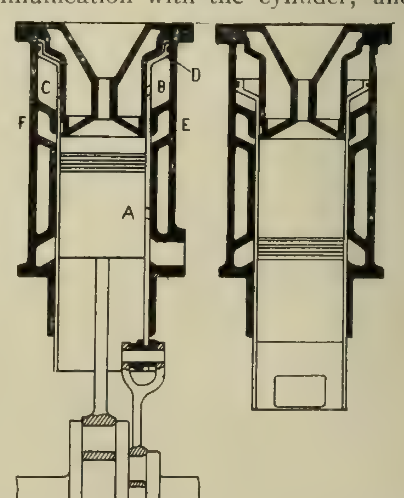
The cylinders are built out of pressed and stamped parts in the following manner :—The cylinder portion A is drawn to the desired shape and welded to a plate B, which forms the base by which the cylinder is attached to the crank chamber. In practice, two cylinders are welded side by side to each base plate. At the top of the cylinder portion is attached the inlet valve casing C, the valve seat being separate therefrom. The exhaust branch D is also welded to the top of the cylinder, and a suitable seating is formed in the upper face of the main cylinder. The exhaust branch is formed with a guide E for the valve stem, whilst the branch and the inlet valve chamber C are provided with pipe attachments. The cylinder is then enclosed in the water jacket, which consists of two parts F and G welded together, which are subsequently provided with nozzles for the pipe attachments, the top being then enclosed by a plate H. Also cylindrical sockets are attached to the cylinder for the reception of the sparking plugs. The illustration shows all the different elements necessary for the building up of a complete cylinder, some of the smaller parts which have not been referred to are filling or strengthening pieces.



Rheinisch-Westfälische Sprengstoff Ak.Ges. No. 12,041/10.

A Two-stroke Engine.

Between the piston and the cylinder lies a sleeve valve having a row of exhaust ports at A and inlet ports B C, which really are arranged at different levels. The sleeve valve is provided with a piston portion D, which operates on both faces to compress air for scavenging on one side and gas on the other. The air and gas so compressed are delivered respectively into chambers E and F, which are separated from one another by partitions. It will be seen that the sleeve valve is driven from a crank on the main shaft, and this crank is set 90° behind the main crank. At the end of the firing stroke the exhaust ports A come opposite the exhaust passages, and soon after the air port B comes into line with the air chamber E. The air previously compressed into the chamber E thus enters the cylinder and blows the exhaust gas out. Subsequently the gas ports C put the gas compression chamber F into communication with the cylinder, allow-



ing the gas to enter and be compressed on the return stroke.

J. Fielding. No. 2,567/10.

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Correspondence on interesting subjects, which come within the scope of this journal is cordially invited. All letters must contain the name and address of the writer, but not necessarily for publication. If a reply is desired by post enquiries must be accompanied by a stamped addressed envelope.

CONTENTS.

THE ELABORATION OF AUTOMOBILE MANUFACTURE.

TO say that the automobile carriage industry is diminutive by comparison with the future vastness of the trade in heavier vehicles for the transport of goods has become a platitude, but like the majority of oft-repeated prophecies it is well worthy of occasional detail examination if only in order that it may be seen how far we have advanced towards fulfilment. And concerning this particular matter it is questionable whether the time is not now ripe for a more detailed consideration of the prospects of the nearer future. Also the private car trade is growing in two opposite directions almost simultaneously, for the self-propelling waggon has only become *common* contemporaneously with the appearance of the aeroplane. Although this fact is, of course, pure coincidence it is likely to have a great effect upon the constitution of the automobile manufacturing industry, and has made the outlook to differ considerably from what it appeared to be a year or so ago. It is not too much to say that it was at one time anticipated in a general and somewhat hazy manner that the existing private car firms would eventually manufacture large quantities of heavy vehicles, but it does not require much introspection to observe that the touring car and the traction waggon are sufficiently different in characteristics to call for totally different treatment. Also the heavy vehicle is scarcely capable of definition even under the generic title just employed, because the three to six ton waggon differs as much from the one ton van as does the latter from a touring car, and the experience needed for the successful manufacture of either is entirely peculiar.

In his recent lecture to the Royal Society of Arts, the editor of our contemporary, "Motor Traction," outlined in a most comprehensive way the different needs of different classes of heavy vehicle users, with especial reference to overseas trade and the requirements of the Dominions and Colonies of the British Empire. Although the author did not himself lay any stress on the subject (which indeed was a side issue so far as he was concerned at the moment) his paper brought home most forcibly the fact that the conditions of working of the cab or pleasure car, the light van and the waggon or tractor are so totally different that if one manufacturer is to cater for each market he will need to establish separate departments for dealing with the design, if not also with the construction, of each class of vehicle. Add to this the influence of the aeroplane, creating a demand for machinery which is, so to speak, on the opposite side of car making, and a most interesting field for speculation is at once disclosed.

It is an outstanding fact that very few manufacturing concerns with high reputations as private car makers have been able to make satisfactory heavy vehicles at the first attempt, while in more than one instance they have tried to obtain a footing in the newer market only to abandon the endeavour as being damaging to their prestige. Of course, instances quite contrary to this can easily be quoted, but closer examination will usually reveal the fact that the different types which may bear the same name still emanate from what are practically separate factories. On the contrary it appears that the best aeronautical work can be done by those who have a wide knowledge of car construction, but this does not mean that this will necessarily be so in the future, because a precisely similar statement would have been equally true as regards heavy vehicles at the time when the latter were as much in their infancy as the aeroplane is to-day.

Examining the question first of all from the point of view of the designer, it is noticeable that though the design of aeronautical engines may be carried out successfully in the same office and by the same men whose chief task is the production of car motors, the design of the remainder of either an aeroplane or a dirigible balloon calls for much special knowledge which would be quite useless to a man who wished to confine his attention to any form of land automobile. Similarly with respect to heavy vehicles, engine design has not proved difficult, but the man with car experience has found troubles almost throughout the rest of the chassis. It is questionable whether a complete study of any one of four separate products of automobile engineering is not sufficient fully to occupy almost any one man, the four being flying machines, touring car chassis, vans or light waggons and heavy waggons or tractors, while it is even now possible to find quite a number of manufacturing firms whose history enables them to be grouped accurately in accordance with this classification.

However, even assuming that separate designing staffs are required to evolve the different types of automobiles, it does not by any means follow that there will not be many large concerns manufacturing all patterns. There have been frequent discussions in engineering circles as to the comparative commercial efficiency of specialised and generalised manufacture, some holding that it is best to make one small article to the exclusion of all others and some asserting, with at least equal

vehemence, that the best structure (be it a battleship or a bicycle) is obtained when every piece of it is made in the same factory under the same supervision. Certainly as far as the automobile car-building trade is concerned the assembling of separately made parts has met with very little success, but this is probably owing to the poor quality of the parts which have been offered at sufficiently low prices to tempt assemblers to make use of them. Against this must be set the fact that there are but few private cars which are made entirely in the works from which they emanate, the specialists on gear manufacture supplying very large quantities of both change-speed and axle drives even to quite leading car firms. Bearings, too, have become a specialised product, principally owing to the influence of the automobile, while an ever-increasing trade is being done in engines of the highest order of merit.

Possibilities of Specialisation.

Having thus digressed for a moment from the main point under consideration, we may carry the matter a step further and say that it appears to be most easy to find arguments in favour of specialised work, and that the strongest opposition to the economic tendency towards it is to be found in the prejudice against it which is shared by producers and users alike. The appreciation of distinctiveness is nowhere stronger than it is in this country, and makers of components for any type of vehicle can usually tell much concerning deviations from their standard patterns which have had to be made for one customer and another. That is to say, there is an inclination on the part of each firm using the part to have it altered in some detail so as to become peculiarly their own. British automobile makers have so far been fortunate in that they have maintained their prices at a reasonable level, so that car building has become a sound branch of engineering and an important British trade. This means it can be reasonably profitable if the manufacturing organisation is good, and therefore the economic advantages of specialised parts production have not been made so obvious as has been the case in America, where price competition has been very fierce indeed. It is sincerely to be hoped that cut-throat competition will never rule conditions here because it cannot result otherwise than in the depreciation of quality, and therefore it seems probable that, in this country, automobiles will continue to be constructed almost entirely by their titular makers for at least a good many years to come.

However, to return to the consideration of the probable future position of the traction vehicle and the aeroplane, both could be made with practically the same tool equipment as is needed for private car manufacture, and thus far all types could be turned out of the same factory, if there were a sufficient number of specialising designers. Still this is not everything, because the workmen (including in the term the managerial works staff and the inspection staff) have to be taken into consideration. In car making the best works have been trained to a certain accuracy and, more important still, a certain finish of work, which in a general sense if correct for cars will be too good for heavy vehicles and scarcely good enough for the lighter and more delicate air machine. The exact differences are perhaps not too easy to define, but anyone who has had experience of making an article in a shop which has been accustomed to something of a quite different class will appreciate the truth of the generality. Here again an important and confusing contra-argument is encountered, because it is a fact that some most successful cars have been made by large general engineering firms who produce one part here and another there throughout the multiplicity of shops under their control. This, however, is probably accounted for by the fact that such works have been trained to do any job demanded and have not done sufficient repetition work on one class of product to acquire a definite style. Thus it is conceivable that there are a few existing firms that could produce all types of automobiles in a commercially satisfactory manner by the addition of separate designing organisations and perhaps an erecting shop or so. On the other hand, it is fairly certain that a firm that has made nothing but touring car chassis is not *per se* suited to produce either heavy vehicles or aeronautical machines, immediately and without fresh study.

There are also indications that the natural tendency with regard to aeroplanes is for entirely new concerns to take over the construction of the peculiar and special parts, leaving the engine making to older engineering firms, because the class of work is totally different in the two cases Again, many of the smaller mechanical details of aeroplanes are made in large quantities by specialists, so the youngest branch of automobile engineering is beginning in a way which makes it very much dependent upon the older branches; although even here it might be said that the most successful aero engines have not been made by old or well-known car manufacturers If it is the intention of the latter to take a large part in the development of the aeroplane so as to reap the benefit of future trade, it behoves them to begin to do so without delay or they will find that they have been left behind at the start and are faced with a long uphill struggle. A firm in starting a new branch of business has more to do than merely to engage one or two specialists: before it can make any name it must accustom the whole of its executive to the necessarily somewhat different conditions of manufacture.

As for the aeroplane, so for the petrol waggon, the leading makers of the future are far more likely to be the comparatively small concerns which are now specialising on traction vehicles than the at present much more famous leading touring car makers. The latter are not to be blamed for inactivity outside their present principal business. Whatever may happen a century hence there will certainly be a very large market for the private car class of vehicle for very many years yet, and a market well worth specialised effort. So where the position of a firm depends upon the personality of its coterie of good men, as is usually the case, it is probably bad commercial policy to give them divided interests.

The Division of Interests.

Taken all together, therefore, there is good reason to believe that automobile construction will soon be too wide a term to enable any one concern to cover all its branches, and the natural divisions, both for commercial considerations and for the point of view of quality of product, are probably not far removed from the four main sections mentioned previously. If this is so, then Great Britain is in the happy position of being at the top of the tree in respect to all but one class because the average quality of the steam waggon, the light traction vehicle or van and the touring car is higher here than in any other country, and will continue to be so unless the automobile industry falls a prey to that misplaced and fallacious commercial notion that price is of greater importance than goodness, a fallacy which has been the ruin of so many young trades. As the aeroplane cannot be made otherwise than as well as possible, if it is to be safe and satisfactory, it ought not to be long before the British flying machine occupies a place equivalent to that now held by the British car. It would, however, hardly be a matter for rejoicing if this point was gained at the expense of losing one of the other positions of precedence, and it is therefore sincerely to be hoped that car manufacturers will resist the undoubted temptation to allow their best men to concentrate their attention on the new field of enterprise, because to do so cannot fail to result in damaging their existing business. It is an undoubted fact that the rate of progress of car design in France has fallen off very greatly in the last two years, and it has been suggested that this may in part be owing to the fascination which flight has exercised over many of the best known French automobile engineers. Whether this is true or untrue is a matter for speculation and is entirely incapable of proof, but it at least seems possible if not probable. It is, in any case, worth while to be on guard against an occurrence of this nature in Great Britain.

Nothing is further removed from the motive of this article than to attempt to dissuade car manufacturers from taking up the construction of flying machines. It merely is desired to point out some of the broader difficulties and to urge those who contemplate an extension of their scope to see well to it that they do not make haste too quickly. Nor even is delay counselled: if a share in the aeroplane trade is coveted by any manufacturer he ought to commence to gain knowledge at once, but it can only be gained at first hand by rather costly experiment, and to equip for aeroplane *manufacture* without first spending a year or two in reasonable experimental *building* is ridiculous, unless simple imitation of some known type is aimed at, and then the imitator is always at least six months behind the originator—a long time indeed in aeronautics. Also the imitator without experience cannot improve upon the model he starts to copy, for any alteration he makes is as likely to be detrimental as advantageous. To commence by a close study of other people's experiments is excellent, but it is impossible for one man to begin quite where the other left off. It is wise to acquire knowledge of existing types of aeroplanes before giving any attention to actual making, but the knowledge to be of any service must be the knowledge of the *user* and not that of the observer pure and simple

A COMPLETE AMERICAN CARD SYSTEM.

In use in an Automobile Factory in U.S.A.

By C. T. Schaefer, M.S.A.E.

THE writer will endeavour to depict the card system in use in several of the leading automobile shops in America, and, though this system at first seems a trifle expensive, when taking into consideration the speed and time saved in locating any part, the amount of work done on it, cost of material, cost of material wasted, etc., it can readily be understood that an expensive system is better than a patched up or incomplete system, where considerable time is lost or wasted in looking up and comparing the different cards to ascertain their correctness.

The factors taken into consideration when this system was initiated were :—

First, the rapidity with which a thing looked for could be found.

Second, to keep an accurate account of all blue prints and patterns sent out from the engineering department.

Third, to keep an accurate account of all the material required for a season's output, so as to enable the purchasing agent to buy in large quantities, thereby reducing the cost of the material.

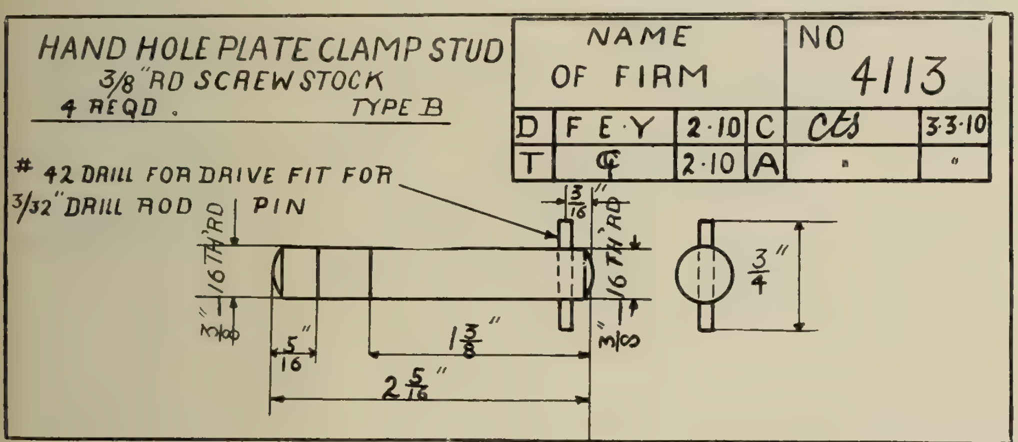


Fig. I.

Fourth, to obtain an accurate cost of the complete car.

The somewhat greater care needed to keep in order a complete system such as this, hardly amounts to anything compared with the time wasted in trying to ascertain desired information from an incomplete or patched-up system.

The Drawing Record.

The drawing sizes are 5×8in., 8×10in., 5×16in., 10×16in., 10×32in., 16×20in., 20×32in., 20×48in., 32×40in. and 40×64in. These sizes worked out very well in practice, especially in automobile manufacturing where a large number of small parts are used, as they are all multiples of 5×8in., and are filed in a cabinet in which the drawers are divided to suit the various sizes of tracings, while the blue prints kept in the drawing office are filed in another similar cabinet. The cabinet containing the tracings is always kept in a strong room to guard against possible destruction by fire.

Fig. I. shows the drawing with title and number. The title, the material, the number required per car and the type of car, are placed in the upper left-hand corner as shown, and the drawing number in the upper right hand corner. The title stamp and number are to be placed on the side which is a multiple of eight inches.

At 1, page 280, the drawing record card is shown. These cards are filed in a small four drawer cabinet, all numbers in consecutive order, and as the tracing is completed the tracer is required to take the next blank card, number the drawing accordingly, and fill out the blank spaces on the cards such as name, assembly number, drawer number and title, while on the space in the stamp he is required to place the draughtsman's name, his own name, and the date. The letters D, T, C and A in the stamp stand for "drawn," "traced," "checked," and "approved." The tracing is then passed on to the checker, and is checked and signed by him; after which it is passed to the chief engineer for his approval. After he has approved the drawing it is ready to be printed.

Blue prints are made by the clerk who is held responsible for the prints and the entire card system. This clerk notes on the card, in the proper columns, the date and to which department, or departments, the prints are issued; when the print is returned the department returning it is given credit for it on the card. At times changes are made and the date, nature of change and by whom ordered, are placed in the column marked "Corrections and Revisions." Should the part be replaced by some new design, this card is marked "Superseded by number —," and carried in the card file. It can be seen readily that in this way parts can be checked with ease. In cases where prints are sent away to companies or individuals, it is noted on the back of the card, the date, name, and address being recorded.

The Stock Lists.

Stock lists of assemblies are issued to different departments, showing such items as engine, clutch, gearbox, axles and chassis, and are arranged as shown at 2, page 280. The idea of these stock lists is to help the foreman in locating the numbers of the different pieces and the number required per assembly. In automobile work there are numerous small pieces that assemble into other parts and are permanently fastened. These parts are taken care of as follows. Referring to 2, page 280, in the first column the piece numbers are entered, while in the second column are entered the numbers of pieces that become permanently fastened to those entered in the first column as, for example, the hand hole cover plate No. 4,142 has four dowels rivetted into it. The numbers of these dowels are entered in the second column following the No. 4142, showing that they have become part of the piece No. 4,142. When the assembler draws the plate from stock, he gets the plate and dowels complete. By having stock lists arranged in this manner, the stockkeeper and the cost department can easily get the data required, and the advantage in so arranging the sheets can readily be seen. The sheets are put on file in the different departments and each foreman has his own file.

Blue Prints in the Shop.

Blue prints are issued to the tool room and from there they are issued to the shop with the necessary tools and fixtures for the job. When the workman in the shop is given a certain job, the foreman of that department gives him the shop card, and he is required to present this card to the tool room to get the blue print. When he receives the blue print he hands the blue print foreman a check, which is hung on a rack below the number of the print issued. This rack has the number of all blue prints marked on it, and by referring to the rack it can at once be seen whether or not a print is in use. When the workman returns the print, his check is taken from the rack and returned to him. This method has a point of economy attached to it, in that it impressed upon the workman that prints must be taken care of, and in this way the life of a print is lengthened.

As stated above, the drawings are multiples of 5×8in. and are filed in the tool room in boxes, one being 8½in. wide and the other 16½in. wide The 5×8in. and 8×10 prints are placed in the small box, the 8×10 being folded to 5×8 inches. The other prints are placed in the large box, the larger sizes being folded to 10×16in. All prints are filed in numerical order, and all prints, excepting general assemblies, are kept in the tool room, the assemblies being issued from the drawing office direct. In caring for blue prints in this manner it is very easy to locate every print when corrections are to be made.

The Pattern Records.

The patterns are numbered when completed, like the drawings, and are recorded on the card shown at 3, page 280. By referring to the card, the master and gate pattern can easily be located; also the cost of both patterns and the total cost, the shrinkage allowed, the number of loose pieces to the pattern and core boxes, as well as the date and by whom the patterns were made. In the upper right hand corner of the card provision is made to take care of the pattern, should it be carried in the company's pattern room. A space is also allowed for a description

A COMPLETE AMERICAN CARD SYSTEM.

Examples of the principal card headings.

1

DRG NO 4113 DRAWER NO A-5x8

NAME Hand hole plate clamp stud

ASSEMBLY NO 4212

CORRECTIONS OR REVISONS

DATE	NATURE AND REASON	ORD. BY
2 26 10	Thread on both ends changed to 16 per inch was 24 per inch	[illegible]

BLUEPRINTS

PATTERN SHOP		TOOL ROOM		ASSEMBLY ROOM		PURCHASING AGENT		WORKS MANAGER	
ISSUED	RET'D	ISSUED	RET'D	ISSUED	RET'D	ISSUED	RET'D	ISSUED	RET'D

2

Crank Case

NO 4212 Hand hole cover plate assembly 4 reqd

Piece no	Piece no	No reqd	Name	Material	Remarks
4142		1	Cover plate	alum casting	
	4112	2	1/4 x 1/2 dowel pin	1/4 drill rod	
	4114	2	1/4 x 3/4 " "	1/4 drill rod	
4113		1	Hand hole plate clamp stud	3/8 round screw	
4115		1	Thumb nut	Forging	
4125		1	Hand hole plat clamp	1/4x1 FTCR steel	

3

NAME Crank Case hand hole cover PATTERN NO 4142

DESCRIPTION	LOOSE PIECES	CORE BOX	LOOSE PIECES	SHRINK	MASTER PATT SECTION	TIER	SHELF	PATTERN SECTION	TIER	SHELF	O
Inspection Cover	x	x	x	11/64	4	12	5				
Engine base holes											

CATED BY	DATE	GATE COST	MATL COST			TOTAL	COST	REMARKS
Gunners	2/2/10	16	00	4	30	20	30	4 pieces to gate pattern

SENT TO	DATE	RETURNED	SENT TO	DATE	RETURNED
Georgia foundry Co	3 1 10				

4

ORDER NO 6268 PART NO 4142

NAME Crank case hand hole Cover

ORDER ISSUED 3 · 3 · 10 NO OF PIECES 100

MATERIAL Aluminium Casting AMT REQUIRED 100

NO PER CAR 4 FOR 25 CARS

REMARKS PIECES REC'D 98

ORDER CLOSED 4 8 10 TOTAL 97

PIECES SCRAPPED 1

REMARKS 1 Casting was defective SIGNED [illegible]

5

SHOP CARD

NO 6268 CARD NO 6268 PRINT NO 4142

PART NO 4142

THIS CARD TO FOLLOW WORK

DATE	PIECES WANTED	RECEIVED BY STOREKEEPER DATE	NUMBER RECEIVED	DELIVERY ON ASSEMBLY REQUISITIONS DATE	ASSEMBLY ORD NO	NO DELIVERED
3·5·10	100	4·7·10	97	4 8 10	5000	4

6

MATERIAL REC'D	KIND OF MATERIAL
98 pieces	Aluminium Casting
	1 scraped defective

MATERIAL ISSUED

PART NO	W'T IN ROUGH	NUMBER OF PIECES	KIND OF MATERIAL
4142	73 1/2 #	98	aluminium casting

7

Machine DEPT SHOP REQUISITION DATE 3·15·10

DELIVER TO Machine DEPT CHARGE TO CARD 6268

NO ORDERED	NO DELIVERED	NAME OF ARTICLE	UNIT PRICE		AMOUNT	
100	98	Crank case cover plate 4142		30	29	42
		aluminium casting				
		73 1/2 lbs @ 40				

8

STORE AND STOCK RECORD CARD DRG NO 4142

NAME OF PART Crank case hand hole cover

TYPE L.N QUANTITY PER CAR 4

MATERIAL Aluminium Casting Bin 14 tier 8 section A

RECEIVED IN STOCK DATE	CARD NO	Q'TY	EXPENDED DATE	CARD NO	Q'TY	BALANCE DATE	QUANTITY	SCRAP RETURN DATE	CARD NO	Q'TY	SCRAP SENT OUT DATE	CARD NO	Q'TY
4 7 10	6268	97	4 8 10	5000	4	4 8 10	93						

9

NAME Crank case hand hole cover CARD NO 6268

AVAILABLE MINIMUM

ORDERED DATE	ORDER NO	QUANTITY	REQUIRED ORD NO	QUANTITY	BALANCE QUANTITY	MEMORANDUM OF MATERIAL PRICED ON BILLS DATE	ORD NO	QUANTITY	UNIT PRICE	DATE	ORD NO	QUANTITY
3 3 10	9863	100	6268	100								

10

NAME

RECEIPTS DATE	INVOICE OR ORDER NO	PATTERN OR DIE NO	QUANTITY	UNIT PRICE	ISSUES DATE	ORDER NO	PATT OR DIE	QUANTITY	UNIT PRICE

11

KIND OF MATERIAL 3/8 Rd screw rod

SIZE OR NAME 3/8 Rd

PER 500 CARS

DRAW NO	TYPE LN	TYPE	TYPE	AMT	ORDERED & RECEIVED DATE	WANTED	FROM	REC'D
4113	417 ft			2000 ft	3 3 10	immediate	Chg Steel Co	3 26 10
4208	223 ft							

DRAW NO	REPAIR	SPECIAL	SCRAP FO	AMT

12

TIME STARTED 7 45 a.m.

ORDER NO 6268 ITEM NO 4142 CHECK NO 362

HRS	RATE	AMT
4 1/4	20	

TIME FINISHED 12 n

DAY WORK ✓

HRS	RATE	AMT
8 3/4	20	1 65

PROD. HRS 8 3/4 AMT 1·65

NON PROD HRS ——

DATE 3·2 10 CHECK NO 362

WORKMANS NAME L M Jones

FOREMAN [illegible] ENTERED 4·9·10

13

CARD NO 6268			CARD NO			CARD NO		
DATE	PC WORK	DAY WORK	DATE	PC WORK	DAY WORK	DATE	PC WORK	DAY WORK
4 10 10		1·65						

14

CARD NO 6268			CARD NO			CARD NO	
DATE	AMT MATERIAL	COST	DATE	AMT MATERIAL	COST	DATE	AMT MATERIAL
3 15	73 1/2 lbs @ 40¢						

SCRAP OR RETURN STOCK

TOTAL MATERIAL

LABOR

SHOP EXP

COST OF JOB

15

NAME Crank case hand hole cover PART NO 4142

ORD NO	FINISHED	AMT	AMT	LABOR	COST (ACT)	BILLING COST	REMARKS

16

JOB NO DATE

CHASSIS NO TYPE

COLOR STRIPE

BODY NO

TYPE

COLOR STRIPE

COLOR TRIMMING

GENERAL EQUIPMENT

DATE WANTED

DATE DELIVERED TO BODY SHOP

DATE DELIVERED FROM BODY SHOP

17

FACTORY ORDER NO DATE

DELIVERED TO

CUSTOMERS ORDER NO

SEND BILL TO

INVOICE NO AMOUNT

CHARGE TO

BILL SENT

NO		PART NO	BILL AT	UNIT PRICE	AMOUNT
	LABOR				

ABOVE SHIPPED 191 VIA TOTAL

CAR NO WHOLE, PART, SHIPMENT, SHIPMENT NO

ORDER NO

ROAD TEST REPORT.

CAR NO......... MOTOR NO..... G.C. NO.......

CHECK NO....... TESTED BY...... DATE......

HOURS ON ROAD...... APPROX. MILEAGE......

TROUBLES DISCOVERED.

IGNITION	A
CARBURETTOR ...	B CARB.
CYLINDERS	C
CONNECT. RODS ...	D
PISTONS	E
VALVES-EXH. ...	F
VALVES-INL. ...	G
ENG. GEARS	H
WATER CIRCU. ..	I
STARTING HAND. ..	J
TIMER	K
COIL	L
HOOD	M
STEERING DEV. ...	N
CLUTCH... ...	O
PUMP	P
OILER	Q
MAGNETO	R
WIRING	S
APRONS	T
PEDALS...	U
CHAINS	V
TRANSMISSION .	W
LEVER-CH. S'P'D. ..	X
LEVER-EM. B'R'K....	Y
BRAKE-DIFF. ...	Z
BRAKE-EMER. ...	

REMARKS : ....

....

WRITE PLAINLY AND KEEP CLEAN.

Fig. II.

and remarks on the pattern to assist in locating it, while the lower portion of the card forms a record to locate the pattern at any time.

Thus we have a very efficient engineering system taking care of everything that is important enough to record. One can tell at a glance the location of any patterns or prints, also any revisions that may have been made.

The Shop System.

The blue prints having been issued to the tool room are passed to the shop from there. When prints are issued to the tool room, the engineering department issues the card shown in 4, page 280, to the works manager's office. This notifies him that the print has been issued, and that work is to be started on these parts. The clerk in the works manager's office fills out the card shown at 5 and 6, page 280 (this being front and back respectively of the same card), entering upon the card the piece number, card number, date, and number of pieces required, after which this card is sent to the foreman of the machine shop. The foreman issues the requisition shown at 7, page 280, which is on the rough store room for material, and merely authorises the storekeeper to issue material to the shop.

The storekeeper notes on the card 5, on the reverse side 6, the number of pieces issued, the weight and kind of material. While the requisition 7 is extended the unit prices and total price, which is to be entered on the cost cards every week. Card 5 and 6 follows the work through the shop, while card 7 is kept in the rough store room, where the cost department is located.

Finished Store Room Record.

The method of keeping a record of the parts in the finished store room is quite simple. When parts are finished they are sent to the store room by the inspector and the store room keeper, upon receiving parts, fills out the card shown at 8, page 280. He enters the date and the quantity finished, in the first column, while the second column takes care of the amount sent to the assembling rooms, the third showing the balance on hand in the store room and from this the inventory is taken. The fourth and fifth columns care for the defective parts. The card numbers, as will be understood, distinguish the various jobs passing through the shop. This card gives all the data that may be required at any time regarding finished pieces.

Rough Store Room and Purchasing Records.

These two departments are combined to simplify the records, 9, page 280, shows the card which takes care of the amount ordered on one side, while 10 shows the reverse side of the card, which takes care of the receipts and issues to shop and balance on hand, enabling the purchasing agent to ascertain the quantity to be ordered. He also has a duplicate of 4, always at hand. Bar stock is cared for by the card shown at 11, which is very simple and requires no discussion.

Cost records are dealt with as shown at 12, page 280, giving the time which takes care of the labour, and at the end of every second week the amounts are charged to the different job numbers on the cost cards 13, while the reverse side of the card, 14, takes care of the material, which is entered from the reverse side of the card 5—that is 6, page 280. The reverse side of the cost card is shown at 14.

When a job is completed, the card 15 is filled out and, from time to time, as the different jobs are finished they are entered on this card, which is carried on file until the particular piece is discontinued. This card shows the advantage of the different manufacturing methods, should different men do the work or different fixtures be employed for it.

READJUSTMENT REPORT.

Item	Adjusted by	Check No.	Hrs.	Min.	Hrs	Min.	Date

SPECIALS ON CAR: ....

TIRE EQUIPMENT: ....

INSPECTED AND O' K'D BY......................

DATE....................

KEEP CLEAN AND UP-TO-DATE.

Fig. III.

Road Testing.

When a chassis is completed and ready to be road tested, the card shown in Fig. II. is attached to the chassis, which as the card shows, takes care of all troubles that arise in testing and makes an excellent record for the designer to find some of the faults in his work, also in the manufacturing and the material used. Fig. III. shows the reverse side of this card, which gives the readjustment report, the inspection and "O.K.," as well as the date when the chassis was delivered to the paint shop.

Factory Order for Chassis and Body.

When an order for a car is received the card 16 is filled out and attached to the chassis, while a duplicate is carried on file in the office for reference. This card takes care of all the necessary instructions, such as painting, trimming, equipment, body type, chassis type, etc.

Should repair parts be ordered, the sheet 17 is used, which has a duplicate for reference in the office. The sheet has all the necessary instructions and also the cost and billing price and is returned to office, with all necessary data for billing.

JAN.	FEB.	MARCH.	APRIL.	MAY.	JUNE.	JULY.	AUG.	SEPT.	OCT.	NOV.	DEC.

Name	Owner of	Price
Enquiry	Type	Delivery Rqd.
Seen	H.P.	
?	Needs	Order No.
Agent	Type	
Address	H.P.	

REMARKS

Fig. IV.

Selling Records.

Fig. IV. is a facsimile of the selling record and is very complete. It is of such a size, that it may be conveniently carried in the salesman's pocket for ready reference, should he be calling on a customer, and there is space on the reverse side for a report of the calls made.

In conclusion, the writer might add that the entire system is the result of careful study and experimental work along these lines and has been in use for several years now, giving excellent results. The writer recommends it to anyone who is desirous of installing a new factory system.

There is a little expense attached to it, but when the functions achieved are considered, it can be seen readily that it is a very conservative expenditure and a necessity in a large plant. We all know that system is the first point of manufacturing, and a concern cannot succeed unless is has a well-planned system.

SMALL BORE ENGINE DEVELOPMENT.

By Robert W. A. Brewer, A.M.I.C.E., M.I.M.E., M.I.A.E., F.S.E.

THE year 1910 will always be remembered in the motor car world as that in which the most remarkable development of the small bore engine took place. Such change has not come about suddenly, it has been making steady progress for the last two years or so, but no doubt owing to the action of the Chancellor of the Exchequer the small engine has received a stimulus the result of which has passed all ordinary anticipation.

The small engine to-day is a most remarkable piece of mechanism, its capabilities of power production seem to have no limits.

Each year higher horse-powers are developed and faster car speeds obtained both in track and road racing, starting with the early Tourist Trophy races, and ending with the recent Voiturette race on the Boulogne circuit, progress has been continuous and rapid. Two years ago we heard of 90 h.p. being developed by a four-cylinder engine of 102 mm. cylinder diameter, and now we have 45 h.p. at least, produced by four 65 mm. cylinders, rated at 10.6 h.p. by the R.A.C. formula. With this latter engine we see a car carrying two persons travelling at a speed of 140 kiloms. = 87.5 miles per hour and averaging 65 miles per hour round a somewhat hilly course with several nasty corners.

The Voiturette race of 1910 failed to solve what has been for some years a moot point in the design of low rating engines, viz., whether one, two or four cylinders were the most efficient power producers for equal piston area. Excellent performances were made by each type, as instance those of the Hispano Suiza four-cylinder engines, and the Lion Peugeot four-cylinder, also the two-cylinder engined car of the same make, the single-cylinder Corre la Licorne being not far behind. We may take it that each of these engines was the highest example of design and construction of its particular type in 1910, and the difference between their performances was not very marked.

When the voiturette races were first instituted, I believe I am correct in the statement that the general opinion of engineers was that the single-cylindered engine would be the most efficient, and there are good grounds for such an opinion.

Consider, for instance, the friction between the pistons and the cylinder walls. If we consider the total piston circumferences we find that the single cylinder has 314 mm. for this dimension, the two-cylinder a total of 502 mm., and the four-cylinder a total of 818 mm. It would be reasonable to assume that the friction of the piston rings would be proportional to these figures.

Then again, the number of crankshaft bearings in the four-cylinder engine is greater than in the single-cylindered or V-shaped two-cylindered engine, which might or might not effect the mechanical efficiency of the engine, according to the design and the load on the bearings.

Let us now consider the efficiency of combustion, and this is synonymous with the heat losses, and the rapidity of propagation of the flame. Taking the former first, Prof. Callendar's formula over-estimates the efficiency for large cylinder dimensions except when running such an engine with a weak mixture, and if the efficiency of engines within the ranges of size we have in mind were approximately the same theoretically, any slight difference would be due to the increased pocket area and surface to volume ratio on the multi-cylinder as compared with the single-cylinder engine.

Callendar's formula is based on the air cycle efficiency η, which depends on the compression ratio, and if D is the diameter of the cylinder in *inches*, the efficiency of the engine $= 0.75\,\eta\,(1 - \frac{1}{D})$. The air cycle efficiency η is obtained from the following expression: $\eta = 1 - (\frac{1}{r})^{0\cdot}$ where r = the compression ratio.

Before proceeding we will see approximately what is the air cycle efficiency of certain engines tested by the well-known authorities, Profs. Callendar, Watson and Hopkinson.

Engine	A	B	C
Diam. in mms.	60	85	117
Stroke in mms.	70	120	129
Compression ratio ..	3·75	4·71	4·18
Air Cycle Efficiency ..	0·411	0·462	0·435
Revs. per minute ..	1270	900	650
Observed efficiency (thermal)	19·9%	23·5%	24·7%
Relative thermal efficiency	0·485	0·509	0·546

I have expressed the observed thermal efficiency as a percentage, as that is the most usual form in which it is considered by engineers.

Now if we consider engine B in which the stroke/bore ratio, and also the compression ratio, is the greatest, we naturally find that the air cycle efficiency is the highest. What the observed efficiency will be will depend upon heat losses and the shape of the compression space, and, as the length of stroke is increased it is obvious that, even though the compression ratio remains the same, the shape of the compression space at the moment of firing is materially effected.

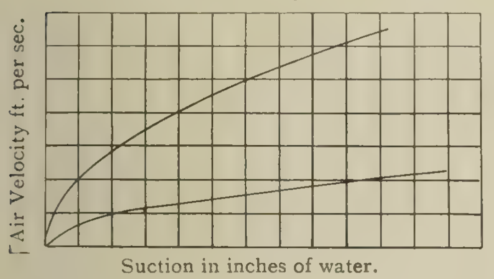
Up to a certain point the increase of length of stroke and consequently the length of compression space between the cylinder head and the piston head both tend to result in a reduced ratio of surface to volume, which decreases as the length of stroke increases. Considering first the engine with the valves in pockets, it is most important that the ratio of the volume of compression space in the cylinder to that in the pockets should be as great as possible, and the longer the stroke the greater this ratio will be. It will be obvious that if the bulk of the mixture is fired in the cylinder, as distinct from that in the pockets, the rate of propagation of the flame will be greater. This is probably so for two reasons, the first being that the reactions taking place within the mixture as it burns will be more direct in the cylinder space which is uniform, than it would be in an ununiform space such as is formed by valve pockets; and secondly, the intensity of the heat and the temperatures throughout, will be higher in volumes approaching spherical, rather than in such flattened volumes as are present when the length of space between the piston head and the cylinder head is small and the volumes are made up with pockets. Thus it appears, from simple reasoning, that a greater thermal efficiency may be expected from a long stroke engine than from a short one.

There is perhaps one interesting fact which can be associated with the small bore and long stroke engine, and that is, assuming the rate of reversals of stress in the reciprocating motors has an upper limit, it is that rather than piston speed which is the limiting factor in engine design. Hitherto piston speed has probably been considered a ruling factor, but in modern construction the error of giving this dimension some sort of a limiting value has been clearly demonstrated; for instance, a piston speed of 4,100 feet per minute average was maintained by the Hispano Suiza, whilst that of the Lion Peugeot was generally approximately 4,000 feet per minute, but higher on many occasions during the Voiturette race in France. Of course, piston speed, as such, is principally a matter of lubrication, and in modern design most efficient means are provided for feeding the oil to the pistons as well as to the bearings.

Apart from purely theoretical questions my experience on Brooklands track with splash lubrication has been most convincing, as I have noticed in making timed circuits that the speed for the third circuit was invariably less by several seconds than that of the first or second, and also that the introduction of a quantity of fresh oil has invariably given the engine renewed life. The reason has been that, in this particular engine, oil worked up the tappet guides and the time

taken during two circuits was sufficient to work out the bulk of the oil. When amply lubricated the piston speed of this engine could be increased to 1,600 feet per minute, though this was a short stroke engine having a diameter and stroke of 100 mm. and 105 mm. respectively. In this case undoubtedly the limiting factor was the number of reversals of stress rather than linear piston speed, as the revolutions per minute were 2,300, which is not very high, and we must therefore look for the cause of limitation of rate of rotation in another direction, and as

Curve D.



the number of reversals of stress per minute is apparently limited on account of centrifugal force and its relation to the materials upon which it acts, we must consider what will be the factor governing stroke length. We shall not be far wrong if we write this down as gas velocity.

The idea has been suggested in some quarters that the velocity of propagation of the flame had an important bearing on piston speed, but we know from results of practical experiments (see page 12, Vol. I., of the Proceedings, Inst., A.E.) that in a correctly proportioned mixture the flame velocity is very much higher than any possible speed. The reduction of restrictions to inlet and exhaust which have been adopted in modern small engine design have, as we know, produced really extraordinary results, for instance, the small engine which won the Voiturette Race, namely, the Hispano Suiza, had valves of 60 mm. diameter, one on each side of a 65 mm. cylinder, and the exhaust pipes were of the same diameter, each cylinder being fitted with a separate pipe projecting at right angles to the crankshaft centre line.

Gas velocity follows a common law which we may write down as $v = \sqrt{2\,gh}$, in the case where air is rushing into a vacuous space, curve D showing the type of graph obtained by plotting velocities against suction. The same law holds in many other instances and may also be applied in calculations of petrol efflux from small orifices, such as carburettor jets, although at present we cannot say definitely whether this law holds exactly, as further investigation is required upon the subject. We may, however, take it that, for the case in point, the velocity of gas through a passage—such as is found in the inlet or exhaust pipe of an engine or the ports and passages between this pipe and the cylinder—follows this law. Investigating the effect of this law upon the design of a high speed small bore engine, we must look to the factors which govern the behaviour of the gas and see how these factors become most apparent in their effect upon the working of the engine. Of course, "g" the acceleration of gravity remains the same, and "h" which is in this case equivalent to the difference in pressure at the two ends of the pipe must be carefully studied. When this pressure is high and momentary, and provided that the area of gas passage is large, any slight increase or decrease of either pressure or area does not affect the case materially; for instance at such pressures as are prevalent at the moment of exhaust opening. The efficiency of the engine will not be greatly affected if this pressure is 10lbs. higher or lower, provided of course that the cylinder is properly emptied at the end of the exhaust stroke and that there is a freedom from back pressure during the exhaust stroke.

With the admission system a much more difficult case arises, and we find that even at the end of the year 1910 there are wide differences of opinion with regard to gas velocities both through the inlet pipe and through the smallest area of the carburettor. A case in point is worthy of examination, and referring again to the two cars we have previously considered, namely, the four-cylinder 65 mm. Hispano Suiza and the two-cylinder 80 mm. Lion Peugeot, making the same calculations as before for piston speed, the following results are obtained.

The four-cylinder engine had a carburettor diameter of 30 mm. at its smallest part, and calculating the mean gas velocity during the suction stroke of any one piston in the ordinary way, as the engine was a four-cylinder, the mean gas velocity through the carburettor for the whole time was in the neighbourhood of 269 feet per second. The gear ratios of the bevel pinion and crown wheels being 15 into 47, at 125 kiloms. an hour, we have for 810 mm. tyres

$$\frac{125 \times 1000 \times 47}{60 \times 2{\cdot}5 \times 15} = 2611 \text{ r.p.m.}$$

$$= \frac{2611 \times 2 \times 200}{305} = 3430 \text{ feet per min. piston speed.}$$

Velocity of gas through a 30 mm. diam. carburettor =

$$= \frac{3430 \times 65^2}{60 \times 30^2} = 269 \text{ feet per second (for the Hispano Suiza).}$$

In order to pass the requisite quantity of petrol a jet diam. of 1.40 mm. was employed. From such figures the flow of petrol and the proportions of the mixture of petrol to air can be calculated by my direct method of using the characteristic curves for carburettor jets shown in curve B for all ordinary purposes.

Turning now to the two-cylinder engine, each of its 80 mm. cylinders was supplied with a separate carburettor of 30 mm. diameter. Now, assuming for facility of calculation that the piston speed of the two engines was the same, the velocity of gas through the carburettors in the case of each engine will be directly proportional to the piston area, and in the case of the Hispano Suiza this was 33 sq. cm., while in the case of the Lion Peugeot it was 50 sq. cm. That is to say, the gas velocity through each carburettor of the two-cylinder engine = 1.5 times that through the carburettor of the four-cylinder engine. It is not my purpose to criticise carburettor arrangements of this description, and it would really be difficult to explain why such widely different gas velocities proved, as a result of numerous bench tests, to suit the individual requirements of these two cars and to result in the excellent performances put up by them on the road. It will be remembered that the two-cylinder car had an engine with the axes of the cylinders at an angle of 15°, and that as far as the carburation was concerned it would be equivalent to providing for two separate single-cylinder engines taking their charge in a rapid succession of gulps. This fact may account for the size of the carburettor employed, but it is interesting to observe that the jet diameters in these two carburettors were 1.25 mm.

Coming now to actual consumption figures, these work out for the Hispano Suiza at 16 litres per 100 kiloms., and for the Lion Peugeot at 11 litres per 100 kiloms., and relating consumptions to cylinder volumes swept out, we have :—

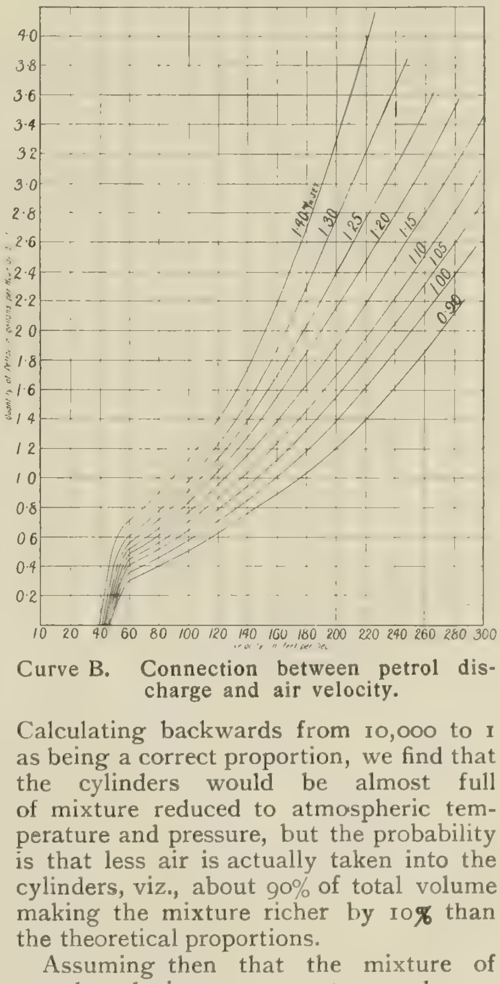
Hispano Suiza 1. Cyl. Volume $= 33 \times 2$c c.c. = 660.

Total Volume per minute $= 660 \times 2 \times 26\;11 = 3{,}450{,}000$ cc.

supposing 100 % full of mixture all the time.

Neglecting the volume of the vapour and assuming a speed of 125 kiloms. an hour, the petrol consumed in a minute = 333 cu. cms.

Therefore the ratio of liquid to air = 10,603 to 1 for 100% cylinder volume.



Curve B. Connection between petrol discharge and air velocity.

Calculating backwards from 10,000 to 1 as being a correct proportion, we find that the cylinders would be almost full of mixture reduced to atmospheric temperature and pressure, but the probability is that less air is actually taken into the cylinders, viz., about 90% of total volume making the mixture richer by 10% than the theoretical proportions.

Assuming then that the mixture of petrol and air was correct, we observe that a jet of 1.25 mm. diameter in a gas velocity of 404 feet per second (i.e., 269×1.5) will supply 1.5 times as much petrol (to the Lion Peugeot engine) as does a 1.40 mm. diameter jet in a gas velocity of 269 feet per second to the Hispano Suiza. It will be observed here that in the case of the two-cylinder engine not only is the gas velocity periodically 1.5 times as high, but the quantity of air to be carburated in one aspiration is also 2.1 times as great, because it only flows through the carburettor during half the time dealt with.

Whatever conclusion we may come to in this matter one obvious point is that, owing to the small differences in the pressure head acting upon the inlet mixture, even slight increases will seriously affect the quantity of mixture entering the cylinder, and secondly the actual efficiency of the engine. Referring to the previous formula we find that it is this pressure head which has practically the sole bearing upon the efficiency of charging in any well designed modern high speed engine. My opinion, based on personal experiments, is that for a four-cylinder engine the maximum gas velocity through the smallest part of the carburettor (i.e., in the vicinity of the jet) should be of the order of 200 feet per second, and that this gas velocity should not be subjected to any great changes either of value or direction on its way to the inlet valves. Such a velocity will give a back pressure of the order of ten inches of water and would generally be amply sufficient to maintain a regular working of all ordinary sized jets. Much lower velocities will be attained with erratic petrol discharge unless the arrangement of the carburettor device is very carefully carried out. The calculation previously given for the Hispano Suiza can also be made for the Lion Peugeot from the data given in this article.

The year 1910 has witnessed steady but marked development in small bore engines; not only is the power obtained from these engines very materially higher on account of lengthened stroke, but the general design of the more normal types of engines than those we have just considered has been improved to a great extent, and the material used in the construction of vital moving parts has increased enormously in quality. In addition to the increase in size of the inlet and outlet gas passages, their shape is of very great importance, particularly the shape of the induction pipe. Occasionally we have seen on Brooklands track and elsewhere really high speeds obtained by engines fitted with extraordinary long induction pipes, but generally speaking if the carburettor functions as it should do, a long inlet pipe is not necessary to assist the carburation process. In many cases where such a pipe is adopted (and there are persons who still prefer a long inlet pipe), any improvement to the running of the engine is probably due to the fact that the carburettor alone is not in itself really efficient.

We cannot overlook the remarkable performances of cars with engines of 20 h.p. rating and under on Brooklands track during the past season, and in many cases these extraordinarily good speeds have been achieved by careful study of certain details which have already been alluded to.

Principally these may be classed into (1) improvement of the lubrication system, (2) reduction of weight of the reciprocating parts, as, for instance, fitting pistons of steel weighing about half-a-pound each, also hollow tubular connecting rods, (3) the employment of special metal for bearing surfaces, and last, but not least, ample valve area and more efficiently shaped cams. This latter detail has not been considered in length, as it has been somewhat fully discussed in the *Automobile Engineer*, October number. It remains for the coming season to show us what valves other than of the poppet type can do, and also we may hope to see some interesting performances made by two-cycle engines of various types in 1911.

It is satisfactory to note that British firms have been well to the fore in the development of the small bore engine, and several whose names are household words require every credit for the excellent performances put up by their productions. The White and Poppe firm have given an interesting lead in their competition for efficient engines which should still further stimulate our efforts in the production of a type of engine which will develop the maximum possible power upon a minimum taxation rating and one which at the same time should be smooth, comfortable and convenient to drive in an ordinary touring car, pulling well at slow speeds whilst having the maximum amount of controllability.

THE DELIVERY OF WORKS SUPPLIES.

A brief consideration of the troubles which commonly occur, owing to overdue delivery of raw materials and partly finished supplies, with an outline of a scheme for overcoming the difficulty.

ONE of the most serious problems which confront manufacturers at the present day is the best method by which delivery of raw material can be obtained in such a manner that the erection and delivery of cars may not be impeded. In very many works, consequent on the great number of cars which are being turned out, considerable trouble is taken to stop the almost perpetual cry that the shops are standing for either castings, stampings, forgings or some of the hundred and one minor details, which latter are often the principal cause of delays, for unlike most problems, the minor matters are very often those which create the most trouble. Such things as bolts and nuts, cotter pins, eyelets and small springs having a knack of holding up progress entirely their own, while their cost is increased considerably when the subsequent telegrams and telephone messages have been reckoned in.

This trouble is by no means so prevalent in other works, principally because the motor trade needs supplies of such a very diverse nature, necessarily drawn from many different factories, and it is this which adds greatly to the difficulty of dealing with the smaller parts. Were it possible financially, the best solution would lie in gathering all these allied trades into the main factory—as is often done with casting and forging and less often, in this country, with stampings and pressings—but it is impossible to erect departments which will deal with glass windscreens, springs, radiator tubes and copper or steel pipes.

The systems at present in vogue with large works are not always those which will lead to a speedy solution of the difficulty, or which are likely to create amicable feelings between the works and that department whose unfortunate duty it is to provide them with material, since it is inevitable that the works will consider no real attempt is being made to keep them supplied, while the department is invariably worried by unnecessary requests from every man who has anything whatever to do with the handling of material, be his rank never so small.

The Question of Responsibility.

In some works it is the custom to make that department which is concerned with the purchasing of material also responsible for its delivery. This is not to be recommended, since purchasing is in itself enough for any one department, and the members of it should be specialists in their own line, with a considerable extra knowledge of the strength and composition of materials and an ability to distinguish between breakdown due to design and that due to bad treatment of the metal in question. In nearly every works there is one department whose duty it is to take accurate record of all which may arrive in the works, to allocate it to separate stores, and to keep a careful check on the consumption of raw material by the shops. This is generally best done by means of a large card system, from which, on being given the number or description of any one piece, its entire history may be traced, together with the number on order at present undelivered and all particulars of that nature. Also, it is usual for the same department, often termed the Stock Department, to issue all orders. This stock department is not generally under the same head as the purchasing, but issues orders subject to certain supervision from the latter. It follows therefore that all records, which must be kept by the purchasing department for purposes of watching material, are in reality but duplicates of those kept by the stock department, and it would save time and expense if the latter assumed responsibility for the whole undertaking.

To give some idea of the course which is generally followed, it would be well to observe exactly what steps are often taken to obtain material when the lack of it is felt in the shops. The first proceeding is to make up a complete list of things lacking, from the requisition notes which cannot be cleared in the shops. This list is generally rendered confusing by the unwonted enthusiasm of the stores man, who frequently improves it by the addition of many things that he does not want immediately. The list is passed to the forwarding department, who straightway look up the number of pieces on order, their date, and the probable date delivery was requested, in the meanwhile discovering certain mistakes. These errors are checked by the stock department, and a long and wordy argument started with the original scribe. This settled, letters, telegrams and telephone messages are sent, and promises (which are often broken) obtained from the vendors. During this time, however, the machine shop superintendent has generally made out another list of practically the same material, which

he also sends in, and numerous subordinates have telephoned, stating that they are standing for each item separately. The old and the new lists do not resemble each other until studied closely; accordingly, a report is probably compiled for each and much more time wasted. There are also usually a considerable number of items which do not agree when the duplicate records of each department are examined, while there is a crop of items which the vendors state to have been sent, but the works deny receipt. All this leads to a very pretty argument indeed, and one in which every works department is speedily involved, until their united efforts have obtained the necessary article.

Some Avoidable Difficulties.

Now it would seem that much trouble can be avoided if a department or subdepartment is evolved having as its sole duty the careful watching of orders, and this department should be a subsidiary part of the stock department, using the same card system and working entirely under the head of that department, thereby bringing it into closer touch with the stores and reducing the card system to a single main set.

One of the best arrangements for dealing with the incoming material is the scheduled system, in which a set programme of so many cars a week is laid down. On issue, it is immediately translated into orders for material to make up the said number of cars, and each supplier is furnished with a schedule stating the date on which his particular wares must be in the factory, he being also notified that instant reply must be given should he consider the date given to be an impossible one. Each date is so arranged that a clear week's grace is allowed before the stores requisition note will come in, and it is the duty of the forwarding department to check its schedule with the stock cards, and to watch carefully that material is in all cases delivered to the specified date. Undoubtedly there will be much which will not be so delivered, and this is the reason for giving a clear week in which it can be obtained before there is any call from the shops. A week should be sufficient margin, even when the required object is a speciality of one firm only.

A sufficient number of men ought be be provided, so that there should be no possible chance of a delivery date being overlooked. Then, when the time for the shop requisitions comes, the material is likely to be in the stores, or a full explanation of its absence ready at hand. It is, of course, necessary that special care be taken to keep the receiving stores up-to-date, so that nothing may remain there unbooked for any length of time, thus not appearing on the stock cards. A circular letter can be composed which may be sent to a firm in order to advise them that the date of delivery is near at hand, and a further letter stating that it is overdue, and requiring a telephone message saying what steps have been taken to despatch.

Works lists of overdue material should be freed from duplications before arrivin gat the forwarding department, and it should be the duty of one man to annex thereto the supplier's report obtained previously, though a small safety margin will certainly be found necessary, as promises are seldom of value.

The Importance of Accurate Ordering.

A vast amount depends on the order date, as a firm frequently requires a quite impossible delivery (owing to some oversight, or to a special rush job), but in the main this should be counteracted by special attention to the clause requiring manufacturers to announce impossibilities immediately. In the case of malleable castings, very little can be done, as the annealing time and the possibilities of scrap render it a hard subject to hurry; the only way open is to order at the earliest possible date. Substitution of castings in another metal at the last moment upsets stock records, and is seldom cheap. Trouble due to a man allowing his store to run out can never be really overcome, but much can be effected by choosing the men with care and impressing this point on them.

Very often such things as bolts and nuts are not regarded as serious items, but, perhaps for this reason, they are often responsible for a serious delay, more especially since motor firms have many bolts made to special drawings and in special steels, and it is a bad practice to leave things until sudden arrangements have to be made for manufacturing these details in the shops. It is also necessary to watch the making and despatching of patterns, and to keep a record in conjunction with the stock department card system. Suppliers are only too anxious to use a pattern as an excuse for non-delivery, stating either that it has been withdrawn for alteration, or that it is not in their possession, men with care and impressing this point caused by such means.

The Position of the Supplier.

The weak point in any system of this description is the supplier, but there is much to be said on his side that is too often neglected by the manufacturer. The trouble experienced by the supplier generally takes the form of too little notice. From his point of view it would be best to order material for an entire season's cars in a batch, and to supplement this order at a future date, with delivery instructions, so that he may put the quantity needed into stock, and deliver from that without trouble either to himself or his clients. This has an additional advantage, in that the price for a quantity would be less than that for small orders totalling eventually the same quantity, and would be excellent practice were it possible, but there are certain underlying facts which prevent an order being placed on these lines. In the first place, to estimate the quantity of cars of a certain definite type that will be sold is more difficult than is usually supposed. Secondly, there are many things which may render a particular pattern or die obsolete, which may not become apparent until a considerable number of pieces have been in the hands of private owners. The first of these features has been emphasized in the pressed steel frame trade, where numbers of frames already in stock could not be used, as the season's requirements for that particular type were not up to the expectations of the firm who ordered them.

Another complaint frequently heard from suppliers is in connection with the manufacture of bolts and nuts, because those which are made from special steel give additional trouble, as the steel cannot be ordered in sufficient quantities to ensure rapid delivery. Undoubtedly one could not expect a big steel firm to pay extra attention to an order for some dozen feet of special steel bar, while a considerable quantity of nuts can be made from quite a small length. The only remedy seems to be that special steel should be supplied to the bolt makers by the car manufacturer, because it can then be ordered with the other necessary steel, forming a considerable aggregate quantity.

The more general trouble is due to every manufacturer expecting his supplier to pay attention to his own particular orders, irrespective of other people's, no matter whether the order be for two feet or for two thousand feet of steel Of course, this is natural, and is scarcely likely to alter, since each firm must consider itself alone, or it will get but little attention. On the other hand the unfortunate supplier, trying to satisfy each several patron generally succeeds in annoying them all.

Much might be done if more attention was given to promises, for however hard it may be to resist the temptation to give a suitable, but unpremeditated, promise in order to get rid of an importunate enquirer, there is always worse trouble when the sequence takes the form of a broken promise. After all, the railway company is a poor excuse at the best.

Undoubtedly an excellent impression is created by the discovery of some firm who can be relied on to keep a promise made, and it has been known sometimes that this has caused the placing of an order in the face of an unfavourable price.

Just so long as great outputs are necessary difficulties will crop up in connection with the delivery of material, but these can be reduced in an extraordinary way by making them the whole work of one department, and not a duty which can easily be attached to the work of a department as a species of side issue.

Necessary Staff.

The department which watches over the orders of its firm is not likely to be popular, either with those it strives to feed or with outside suppliers, but its condition can be immensely improved and much smoother working would follow if common-sense was brought to bear on its arrangement.

The mistake is often made of limiting the staff necessary for the accurate supervision of works supply. If there is one department in the works which should have an ample supply of men it is the forwarding department, because it must be remembered that this department must watch every order which is placed, however small or insignificant that order may be, and must be able to give its complete history on enquiry being made. Probably it is best to divide the various orders into several main headings and detail one man to each section. Thus stampings, pressings and forgings might form one division and castings another, while bolts, nuts, and small washers may be a third. Carriage building should be separated from the engineering side of the department if this branch of the industry is included in the factory. Care must, of course, be taken to see that all card systems are kept rigorously up to date, whatever may occur to encourage delay, since the accuracy of the knowledge supplied to the entire works depends thereon.

THE WAR OFFICE AERO FACTORY.

The system of the works and the nature of the manufacturing carried on.

THOUGH bearing, owing to old association, the name of "Balloon Factory," the works adjacent to the excellent flying ground at Farnborough are really better described by the more comprehensive title used as a heading for this article, because at present manufacture is divided between balloons, aeroplanes and dirigibles. Flying machines of all kinds are repaired there, and are being designed and may be heard of before long. Like most of the executive departments under Government control, the balloon factory has not escaped a deal of adverse criticism at the hands of those who know little or nothing about it, and it should therefore be explained that the factory is entirely distinct from the Balloon school, which in no way controls it and is a completely separate department, although the latter carries out a certain amount of air testing for the factory. That is to say, the school take machines from the factory and is responsible for the handling of them, afterwards reporting their behaviour to the factory, and the arrangement by which the civilian staff of the factory is represented on all airship tests and trips has worked admirably by conducing to the equal acquiring of knowledge by both those who are to use, and those who are to make, the various types of machines.

Fig. I. General View of the Factory.

Much of the work under the control of the factory superintendent consists of keeping in repair the machines belonging to the school conducting experimental constructive work, manufacturng gas for balloons and, finally, designing and making full-sized dirigible balloons and aeroplanes. For these several purposes there are three main divisions of the factory, each in the hands of a separate manager, consisting of the mechanical department, the physics department, and the fabric department. All these, of course, are under the control of the superintendent, the work of one department being corelated to that of another in the superintendent's office, from which all orders are issued and to which all requisitions are made.

The most interesting from the point of view of the average automobile engineer is, of course, the mechanical section, which consists of the designing and drawing office, a woodwork and pattern making shop, and a machine shop. Erecting complete chassis for dirigible balloons is carried out in the new shop shown in Fig. II. This shop is not large enough to contain an envelope in the inflated condition, but will hold three or four of the largest frames which are likely to be made for some considerable time to come. Down the walls on each side there are adjustable steel arms which can be manœuvred into positions such that the frame can be suspended exactly as though it was hanging directly from the balloon, thus enabling experiments as to balance and so forth to be carried out quite conveniently. This erecting shop is the most prominent building in the general view of the factory shown in Fig. I., and part of it is at present full of aeroplanes under construction. The lower shop on the near side of it contains the wood and metal working sections and foundry. In these there is a fair equipment of tools, mostly of the size and type used for automobile work, and an interesting peculiarity of the shop is the part devoted to the manufacture of frames and tanks of various kinds. While it is possible to make very strong structures from exceedingly thin sheet steel, the joining up of one piece to another is not easy to perform securely without adding a good deal of weight in the form of webs, rivets, etc. Acetylene welding was tried, but it was found that there was considerable difficulty in obtaining sufficient width of weld, so the process now most often employed is a combination of electric and acetylene welding. In the case of a tank, for example, a flange will be pressed or hammered up on the end pieces which are united to the sides firstly by electric "spot" welding. That is to say, current is applied at a series of points all round the flange, a second and third row (staggered as to the first series) sometimes being added. This unites the two parts in exactly the same way as a large number of diminutive rivets and, to make certain of the tightness of the join against fluid or gaseous leakage, an acetylene weld is finally made all the way round the edges of the flanges. Although the electric process perhaps sounds somewhat complicated it is certainly more rapid than rivetting and equally secure.

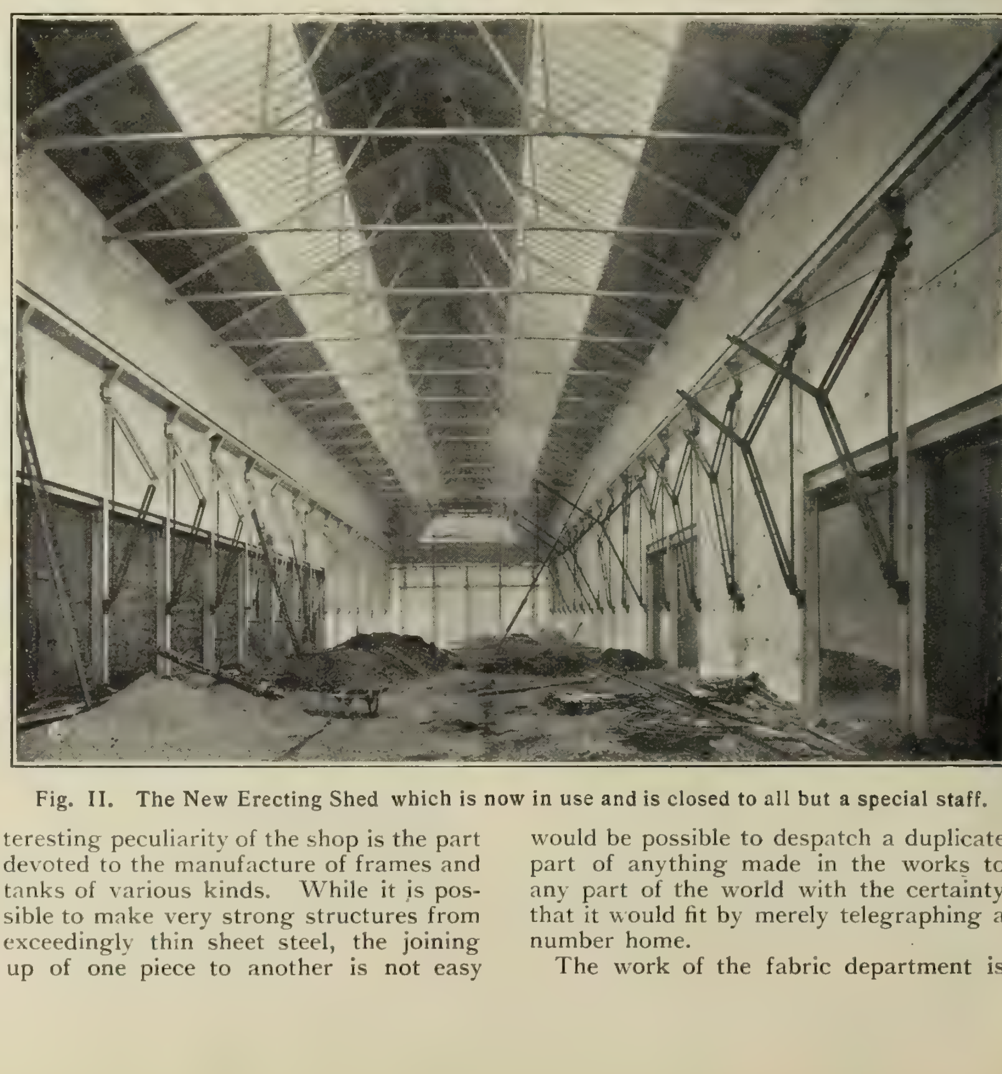


Fig. II. The New Erecting Shed which is now in use and is closed to all but a special staff.

The total output of these small shops is, of course, not large, but it is thorough, nothing being made except to full drawings and very little except to jig. As parts are but seldom duplicated this seems at first sight to be an unnecessary waste of time and money, but it is important to remember that all machines constructed at the factory are intended for use in warfare, and it is impossible to say whether any one machine will or will not see active service. In the event of an unexpected war it would be necessary to make the best use possible of the air-craft in existence and, with the present system, it would be possible to despatch a duplicate part of anything made in the works to any part of the world with the certainty that it would fit by merely telegraphing a number home.

The work of the fabric department is

explained by its title as it consists entirely of the testing of materials and the making of envelopes, the covering of planes and the making of kites, sandbags, and the canvas roofs and sides of portable aeroplane sheds. A part of the section is devoted to the repair of the balloons used by the school and the making of new balloons to replace worn out ones.

The Physics Section.

The physics department controls the tests of materials and the manufacture of hydrogen for the use of the school. The gas is made by a number of different processes, a portion being produced by electrolysis, the power for which is derived from the testing of the large airship and aeroplane engines for long hours at a time on suitable tilting test beds in an artificial 30 mile an hour wind. This department also takes responsibility for model experimenting at the suggestion of the mechanical department. The principal tests of models now being conducted are directed towards the discovery of the best proportions for dirigible envelopes, and Fig. III. shows

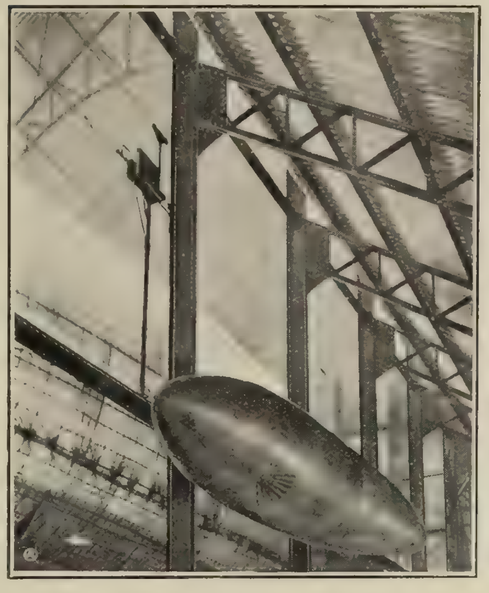


Fig. III. An 18 foot model.

an apparatus for determining the head resistance of fairly large models, and the arrangement and size of stabilizing fins. This arrangement is contained in the largest shed of all, seen in the background of Fig. I., and it will be observed that the model floats freely in the air, being towed by an arm hanging from a wire above. A recording apparatus registers the pull and the speed, while there is an additional contrivance which makes it possible to release the model at a pre-determined point in order that its behaviour, as regards stability and the effect of fins, etc., can be observed, the speed of motion at the instant of release, of course, being known.

To the most casual visitor it would be obvious that the exact and elaborate system of the factory is applied rigidly, and that the different departments work with each other with the utmost smoothness, and this is probably largely attributable to the dependence of each section on the general or superintending office The exact record of all work which is made should, in the future, prove to be most valuable.

SOME POINTS OF POSSIBLE DEVELOPMENT.

A speculation concerning automobile design of the nearer future.

By A. R. Hendon.

NOW that we have safely passed the time when motor journals take stock of the past, the present writer may perhaps be pardoned for looking towards the future and indulging in a little mild speculation on the lines of development that are likely to be taken by automobile design in the nearer future. By this term "the nearer future" I mean the period that is likely to see the next few immediate distinct steps in progress, apart from such developments as the introduction of turbines or rotary pressure engines. These, though they may be of the probable, are not of the nearer future —at least, for practical work on the commercial basis of large outputs to standardised designs.

Taking then engine design first, one can hardly help noticing the growing attention that is being given towards the two-stroke cycle—or indeed the increasingly successful results that are being obtained from engines embodying this principle. Indeed, there seems a fairly general opinion among engineers that the apotheosis of the two-stroke system is rapidly approaching. But general also is the opinion among those best able to judge that, when this type of engine arrives at the maturity of reasonable mechanical perfection, the fuel and air will both be supplied to its cylinders under pressure, but separately, and the fuel supply at least will be measured mechanically at each charge.

But whatever form it takes in the future, the two-stroke engine is likely to loom very much larger in automobile design than is the case at present, for it has been undeservedly neglected. The better known and less experimental Beau de Rochas cycle is a heritage from the makers of gas and oil engines in the pre-automobile days, but it might be suggested with some show of reason that motor engineers have been somewhat biased in favour of the latter owing to its being so well known, while its reputation for efficiency may have led many to overlook other qualities which, though of little value in stationary gas engines, might be worth some sacrifices in other directions when considered for road work.

Still, whatever the advantages possessed by the fully developed two-stroke engine, he would be a bold man who would assume that the development of the rival system would remain stationary and be rapidly superseded.

Rotary Valves.

How then is the four-cycle type of engine likely to be developed? Already the introduction of the cylinder liner sleeve valve is lapsing into ancient history, the ordinary piston valve is being given extensive trial, and the rotary valve has made public appearance. If the writer may be allowed to hazard an opinion, he would like to suggest that in the long run the more even and more perfectly mechanical action of the rotary valve will displace the reciprocating movements—with all their rapidly alternating stresses—of the other types of valve (to say nothing of the stresses caused by having to lift unbalanced large diameter valves against pressure, as in the case of the mushroom valve—an objection that, with the increasing sizes of valves, may possibly be greatly underrated). Still, the difficulties in the way of the rotary valve are considerable, but already there are evidences that these are being overcome. In fact, their appearance at the 1910 exhibitions already justifies their inclusion as developments of "the nearer future."

An engine feature of minor importance, but one of which I think we shall undoubtedly see more before long is some form of accommodation for a dynamo to supply the complete set of lamps with current, and here it might well be suggested that some sort of agreement be reached as to standardising of the platform and pitch of holes for the holding down bolts. In this age of luxury it is quite conceivable that electric lighting plant will be regarded more or less as an essential on cars—at any rate, of the more expensive types.

Reference has already been made to the accurate measurement of the fuel charge in the case of two-stroke engines, and I should like, before leaving the subject, to suggest that even the most scientifically designed carburettors of the present day leave much to be desired in this respect; and this, because we do not consider the conditions under which they have to work. The scientific research that has been brought to bear on them has worked wonders—up to a point—and no one would be foolish enough to deny the valuable influence of science in this direction; but I do maintain that, so long as the present form of carburettor with float feed mechanism is retained, we can only reach very partial success. Consider the mechanism and the conditions under which it has to work, and then let us ask ourselves whether we can expect a comparatively delicate mechanism to work accurately while the float is constantly being shaken about by the combined efforts of engine vibration and road shock. These facts are sufficiently evidenced by the difficulties experienced by motor cycle riders with their carburation at Brooklands, when travelling fast. Perhaps the advent of solidified petrol may have a beneficial influence in this direction by turning us to various other methods of carburation. With the greatest respect for the genius of those who invented the float feed carburettor, designers might well question whether that invention has not led them along lines rather too stereotyped for the stage

of development which has now been reached.

More and ever more strongly is the tendency evident to place the gearbox as close to the engine and make it as short as may be, so as to obtain as long a propeller shaft with as small an angle at the joints as possible; but there is no evidence as yet that the sliding gear is likely to be rejected in the near future. In fact it is rather curious that, in spite of the vast increase of range in engine controllability, or flexibility—call it what you will—there has been a most marked tendency towards a larger number of gear changes. Though we may not see much change in the gearbox, however, it is quite likely that the side change lever may depart.

At the time of its introduction the side lever became a feature of fashion, though to my mind there was much to be said for the placing of the lever on the steering column. The adoption of the gate system of gear changing strengthened the case for side position, but we have now reached the time when two present-day usages are at war with each other, for few will commend the combination of high sided body and side levers, as it exists at present. At best it results in a botched job. In any case, the levers have to be placed inconveniently close to the side of the car, and whilst the outside position necessitates the driver reaching his hand over the side of the body, the inside position takes up room that should be occupied by the driver, and may necessitate his having to sit somewhat sideways, to say nothing of the possible effects of any part of his body inadvertently pressing against the levers. Therefore, whether outside or inside the body, the present position of gear lever is far from ideal. Assuming the necessity for sliding gears, the ideal system of gear changing would probably effect its purpose on one of several buttons or switches being pressed by the driver according to the gear required, or at worst by the movement of a lever, no larger than the throttle lever on the steering wheel, into one of several notches in a sector. But though as yet we cannot perhaps expect to attain to such sophisticated mechanisms, there is no reason why a modified gate change lever should not be placed more conveniently than is possible in the present position.

And in this argument for altered location of the gear variation lever, I would include the brake lever, though it is not easy to see where else to place this, unless it also is fitted on the steering column. Such an arrangement might over-stress a steering pillar of the type usually made to-day, and so might call for some form of staying from the dash, which in turn might involve a more rigid dash constructed as part and parcel of the chassis.

But to revert to the change speed mechanism. Suggestion has already been made concerning the ideal method of lever operation, and to obtain such a result a pneumatic system would probably provide the simplest solution. This may seem a complicated method for obtaining simple results, but when we come to realise the number of other uses on a car to which a supply of compressed air might be put (as, for instance, engine starting, tyre filling, power and speed reduction, etc.), it is quite reasonably conceivable that a system of gear variation control by pneumatic power may result in comparative simplicity, if due regard be had to the complete car as an entity. In any case, the added complication would go far towards providing other advantages as well as those of simple gear changing. If such a system of air pressure were adopted for petrol and oil feed an excellent method of using the pressure might be borrowed from a certain French car. In this vehicle the petrol and oil pressure are inter-connected so that when either oil or fuel are exhausted the pressure in both tanks escapes, and thus fuel cannot be supplied without oil, or oil without fuel.

At the beginning of this article I tied myself down as far as possible to the evolution of the car in "the nearer future," but there is a certain species of development in invention that cannot with certainty be assigned, or confined, to any given period. The foregoing possibility is a case in point, while yet another possible development may take place in steering mechanism, which, as shown by Mr. J. L. Napier's admirable article in *The Automobile Engineer* of July, 1910, still leave much to be desired.

Personally, the writer would like to see something more attempted with the hollow stub axle containing the pivot within itself in such a position as to bring the axis of the pivot into coincidence with the vertical plane of the wheel. I know that such an arrangement has objections as regards accessibility. Still, when everything is considered, the advantages of the central pivot system to my mind outweigh the drawbacks. It would certainly tend towards easier steering, and need not prove so inaccessible as is generally believed, if we may judge by the embodiments of this form of construction that already exist—and exist most successfully—for heavy motor work. I purposely mentioned the hollow front axle as that appears to me to indicate the most likely form of central pivot steering for private cars at present, but, with the further development of the all-steel wheel for this branch of the industry, we may quite possibly see front wheels with flat spokes dished so as to leave accommodation for the steering pivot in the centre line of the wheel. This also is a method that has met with entire success in heavy industrial motor work.

Differential Imperfections.

Yet another point at which there is room for improvement is the balance gear, which is far from perfect in its action, as may be realised by anyone who studies the behaviour of automobile traffic on "patchy" or greasy pavements. Although the present mechanism differentiates *for* distance, it does so with resistance as the direct cause of action, and as the limiting factor of this resistance is the adhesion of the road wheels, the balance is liable to be upset when the two driving wheels act on surfaces having different coefficients of adhesion. What is really wanted is a mechanism that will act *directly by* distance, and although it is not quite clear how such a result is to be accomplished, for the device would have to be somewhat anticipatory in its operation, still efforts in this—the right—direction have already appeared.

As regards frames, we are witnessing a fairly partial disappearance of the tubular cross member, and possibly, with more perfected methods of three point suspension for engine and transmission units, we shall see the almost total disappearance of the subsidiary frame. It is to be hoped too, that in many frames greater care will be taken to locate rivet and bolt holes as far as possible along the neutral axis of the girder. Apart from these minor details, however, the modern frame has reached the point at which it is not easy to see any sweeping changes in structural development, unless the dash is to be found as an integral cross member as already suggested, with the webs of the longitudinals perhaps brought upwards at the junction, so as to follow the slope of the footboards. It might be suggested, too, that front wheel mudguards be curved round over the front top quadrant of the wheel instead of being continued forward from their top point, in more or less a straight line, as is so often the case. If they followed the curve of the wheel all the mud flung from the wheel would travel at a sharp angle down towards the road surface, whereas in present arrangements it is frequently allowed to fly upwards to a considerable extent, the space between the end of the mudguard and the wheel covering a fairly wide "field."

Generally speaking, the most advanced, provedly successful, features are at first only found in the most luxurious and expensive types of cars, and consequently most of the developments already suggested, if they take definite shape, will only appear on *vehicles de luxe*—at any rate, for some time to come. It is on the larger cars, too, that any radical alterations to the present general system of spring suspension would be introduced. Even if we assume that in the three-quarter elliptic springs, now so much in vogue, the short upper quarter of the spring takes up the vibrations that are of too rapid a periodicity to be absorbed by the longer lower half of the spring, the use of laminated springs is at best only a compromise owing to the varying periods of vibration set up by the different types of road surface, and it is also curious that the properties of the volute form of spring have not received more attention from automobile engineers: its advantages in accommodating its action to loads of varying magnitudes have long been appreciated in railway work. Reverting to vibrations of varying period, Mr. Lanchester only recently suggested the possibility of a spring system that could be adjusted to changing requirements, and here once more an opportunity of utilising the pneumatic system, suggested for other parts of the car mechanism, presents itself. Like several other of the foregoing suggestions, this has already been accomplished singly, by itself, but I venture to think that the employment of a multi use pneumatic system has possibilities, despite the difficulty in making a tight air pressure joint.

In the writer's opinion, with the exception of the tyres, and perhaps the petrol feed, there is no *single* use to which compressed air can justifiably be put, but hardly anyone has yet realised its possible wide range of usefulness in automobile design, if all possible uses are considered in the aggregate.

THE ARRANGEMENT OF BRAKES.

With particular respect to four-wheel and front wheel systems, and an account of some experiments on diagonal braking.

THE imperfections of the standard system of car braking, particularly as regards its liability to cause skidding, has now been generally appreciated and quite a number of manufacturers have recently been experimenting either with duplicate brakes on the rear hubs or with front wheel brakes, in place of the usual propeller shaft drum. From a mechanical point of view the transmission brake is certainly vicious, because its power is great enough to stress the differential and driving gear very highly indeed, but from the view point of a user the arrangement has one outstanding advantage, which is that the shaft brake can always be relied upon to pull up a car very quickly except where the road surface is very greasy. Even under such circumstances it is often preferred to the rear hub brakes because the application can be made by the foot, via a pedal, with much greater delicacy than by hand.

These facts have, of course, been realised for some considerable time, and it is now a matter of several years since the front wheel brake was offered as a solution of the difficulty, and it certainly has advantages which are not shared by the older types, the chief being that it is as safe, or even safer, in mud than a propellor shaft brake, while it throws no stress on the transmission. On the other hand, the "stopping power" of the shaft brake is absent and the effect of locking the front wheels, by violent brake application, destroys all power of steering instantly, which is not the case when a rear wheel is locked. Up to a certain point, depending upon the weight of the car, the front wheel brake is therefore excellent, but it is insufficient for emergency stops and so needs supplemental rear wheel brakes. Also the supplemental brakes ought to be foot applied, because a hand brake is awkward to use for sudden retardation, partly because it necessitates single handed steering at a possibly critical moment, and partly because it cannot be controlled with the same quickness or the same delicacy as a pedal, especially as regards release.

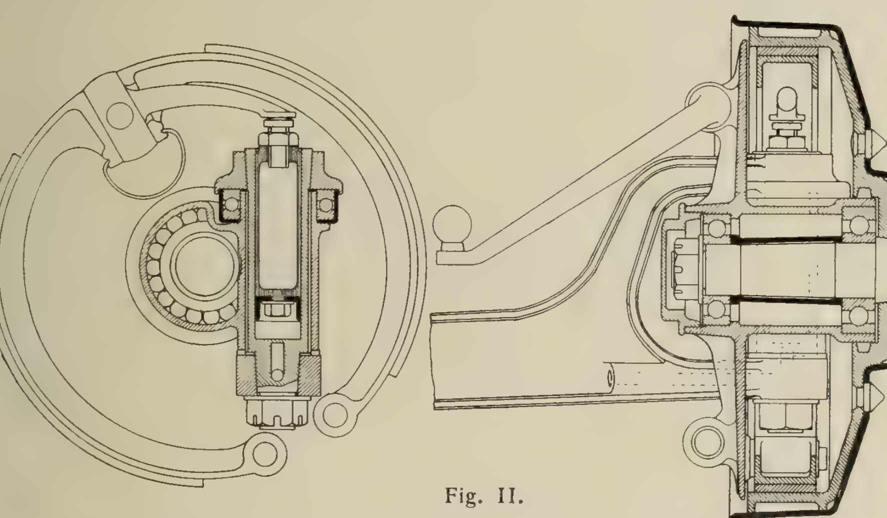


Fig. II.

Thus it is possible to make out a good case theoretically for a four-wheel braking system, the doubtful points being the effect upon the steering of braking all the wheels at once, and the ability of four hub brakes to stop a car as rapidly as a geared-up shaft brake. Still, speaking on theoretical grounds alone, there are some reasons for anticipating that diagonal braking—that is, retarding one back and one front wheel on opposite sides of the car by a compensated mechanism—would be the ideal system, because then, even if both wheels are locked, there is always one front and one back wheel still rolling and so available for steering. This last theory was borne out by some experiments which were made with a model car and published in our contemporary, *The Autocar*, some years ago. The method adopted was to cause the model to run by gravity down a greasy inclined plane and to try all possible combinations of locked wheels, from one to all four. Repeated tests showed that the model would only travel in a straight line when the wheels were locked diagonally. Of course, these tests are not conclusive, because brakes ought not to be applied sufficiently violently to actually stop all rotation of the wheel. Recently, however, owing to the courtesy of the makers of the Weight Hydraulic brake, we have been able to make some experiments on an actual car, though unfortunately, a greasy surface was not at the time available.

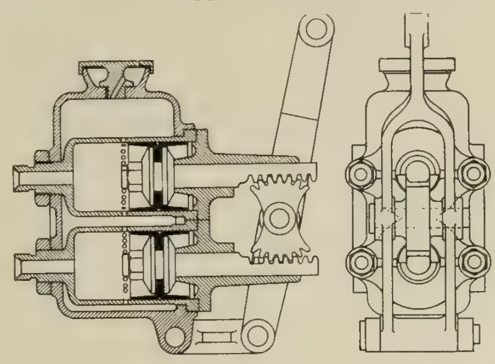


Fig. I.

Of course, diagonal braking is only a possibility when the steering pivots are in the centre of the front hubs, as otherwise the application of one brake of itself tends to deflect the steering, but it has several times been proved that with such pivots a single front wheel brake can be used at its full power without any disastrous results. The experiments which we were able to witness were carried out with a Metallurgique chassis having inclined pivot steering, which is almost equivalent to central pivoting. Tests were first made with all four brakes in action, they being controlled by a single pedal, and the speed was in most cases about thirty miles an hour, while the road was dry and rough. The four brakes proved to have rather more "stopping power" than the propeller shaft brake (which was fitted to the hand lever) without skidding the wheels, while it was possible to put on the brakes, at the full speed mentioned and with the utmost vigour, without touching the steering wheel either on curves or when travelling in a straight line. Having failed to notice any difference between the steering when braked and when free, two of the spreading cams, one on the off-side front wheel and the other on the near-side rear wheel were disconnected, and the trials were repeated with only the remaining two in action. There was then a distinct falling off in the power, though it did not appear to be reduced by so much as half, and the steering was as good as before, no means being discovered whereby it could be affected through the brakes. Tests on dry roads are, of course, inconclusive, and leave it still undecided whether the diagonal would prove superior to the all-four arrangement on greasy surfaces, but, owing to the fact that there is

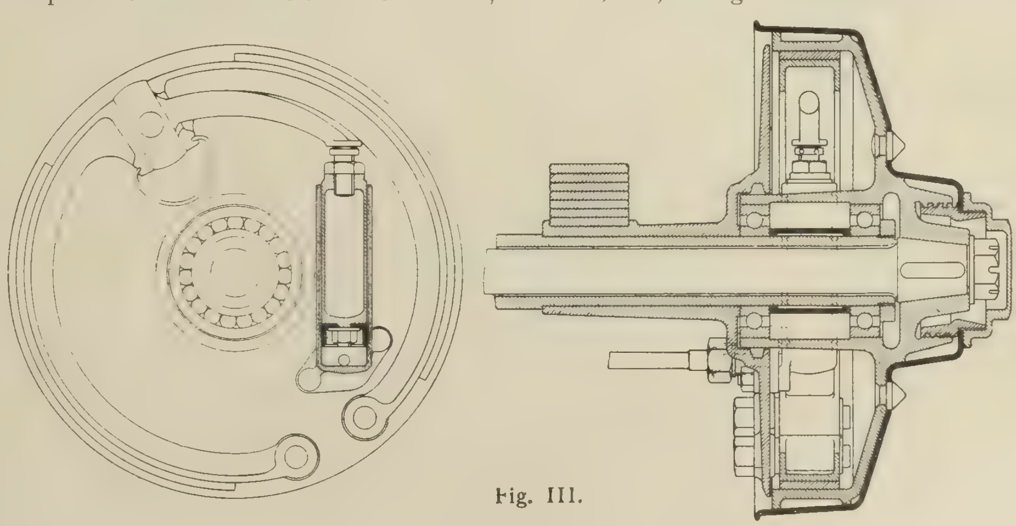


Fig. III.

no tendency to skid one wheel before another, the diagonal system should not exhibit any superiority, except when the wheels are locked, and the four-wheel system enables greater retardation to be obtained without skidding than does the diagonal system with skidding—on dry roads.

As regards the actual mechanism of the Weight brake, this is shown in the three illustrations on page 289, and it may fittingly be remarked here that the action is quite surprisingly smooth, while there is distinctly less spring or "give" noticeable than with any but the very best of rod and link mechanism. The whole system is filled with oil and normally the front and rear pairs of brakes are operated by separate pistons in separate cylinders, though through the same pedal, as shown in Fig. I.; the reason for this arrangement being that the two systems are then independent, and an accident which might damage one would probably leave the other unaffected. From the two cylinders, branched pipes are led to the axles, and the method of application to the shoes is shown in Fig. II. for a front wheel, and Fig. III. for a rear wheel. These two designs are those recommended by the Weight Company, and are such that detachable or interchangeable wheels can be used, but they are, of course, not essential to the principle, for the brakes can be applied to any chassis with or without central pivot steering.

As the drawings reproduced herewith are quite self-explanatory as far as mechanical details are concerned, it is only necessary to point out that there are no flexible joints in the fluid system of the gear, save only the pistons. Leakage of oil appears to be extremely slow—so slow indeed, that it is not a drawback to the brake—and replenishment of small quantities is very easy, besides being automatic within the limit of capacity of the reservoir which surrounds the cylinders, as can be seen in Fig. I.

THE 15 H.P. VALVELESS CHASSIS.

This car is provided with one of the oldest and most successful two-stroke engines, and also has an interesting gearbox.

It has been said with truth that the earlier 20 h.p. Valveless car was too ingeniously designed to be commercially successful, although its performance was quite satisfactory. Apart from the engine, the principal peculiarity lay in the frame and back axle, and, though the type has not been made since 1907, a description of its chief points is interesting now that sheet steel threatens to come into more extended use. Fig. I. is a diagrammatic view of the original frame, which was in one piece with the undershield and all of quite thin metal, the depth being about twelve inches. There were no cross members in the ordinary sense of the term, except at the front and rear ends, but there were two beam-form pressings mounted in the bottom of the shield part and on these the engine rested. From the engine two central silent chains ran to the back axle and could be engaged separately by duplicate clutches, so that two speeds were given in the forward direction. For the reverse there was an epicyclic train inside the flywheel which came into operation if the clutch pedal was held out to the limit of its travel. The back axle was built up entirely from sheet steel about a tenth of an inch in thickness and was both light and strong, while it is said to have been cheap. From the axle there was an open ended chain guard, also of sheet steel, which acted as a radius and torque rod in addition.

Although the original engine was as flexible as most four-cycle engines of its day, and although it was wonderfully well balanced, the violence of the explosions in the single large combustion chamber detracted considerably from the many advantages of the engine, because the torque variations could be felt quite distinctly. In this respect the smaller 15 h.p. type shows a vast improvement, for, judging by sensation alone, an experienced passenger would probably assume that the engine was a particularly well balanced and moderately lively four-cylinder, supposing that he was not aware of the facts, though at certain speeds the suggestion is more that of a two-cylinder 90 degree V engine. It is not our intention to discuss the good and bad features of the Valveless as compared with the normal type of engine. It is enough to say that the running appears to leave little to be desired from the ordinary private owner's point of view, while the simplicity of the engine will be made apparent by the description which follows.

Reference to Fig. II. will make all mechanical details clear, but it may not be out of place to recapitulate the cycle of operations. As the pistons rise air passes through the large circular orifice which is shown at the back of the carburettor. This lifts the large mushroom valve to which is attached the needle that normally closes the jet orifice, thereby allowing petrol to overflow and dribble down into the pipe below. As the pistons descend again the air in the crank case is compressed until the piston on the carburettor side uncovers the induction port in the cylinder wall when the air expands and fills the cylinders, picking up the petrol and making the mixture as it passes up the induction pipe. The exhaust port in the other cylinder is opened just a little in advance of the inlet and remains open a little longer, so the incoming mixture has time to expel most of the burnt gas. Compression in the cylinders is the next stage and then, of course, firing and a repetition of operations. The success of the engine may probably be attributed to three chief causes. Firstly, the exhaust port is a long way away from the inlet, so that the fresh gas has to fill both cylinders before it can escape. This causes the scavenging to be unusually good, especially for a two-cycle engine. Secondly, as practically pure air only is compressed in the base chamber there is no risk of firing back when the ports open. Thirdly, the crank case is sealed against leakage in an extremely simple and effective manner, so that good compression is obtained in it, while the very small quantity of oil present in the crankcase enables the mixture to be kept clean, so that the petrol consumption is not heavy for the power. Control is by the throttle shown in the inlet pipe, and the petrol feed

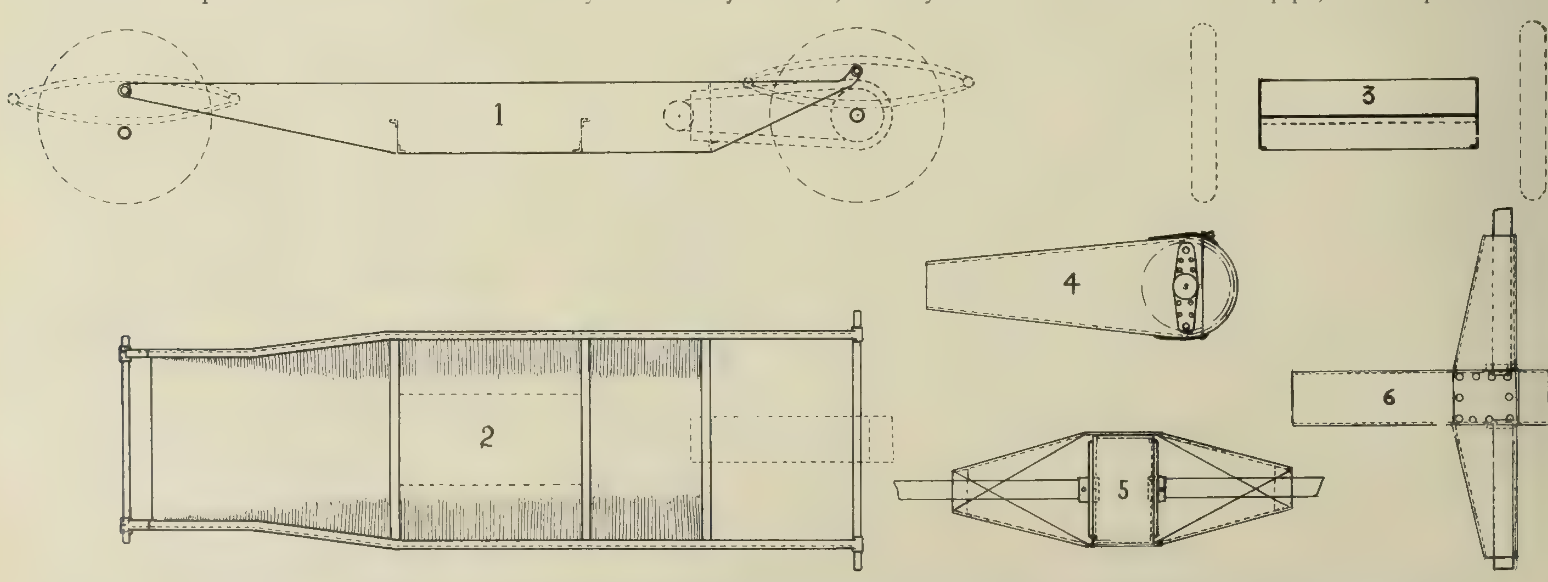


Fig. I. 1.—Elevation of frame. 2.—Plan, dotted lines near number show engine position. 3.—Section showing engine support. 4.—Chain Case. 5.—Rear view of back axle. 6.—Plan of back axle and chain case.

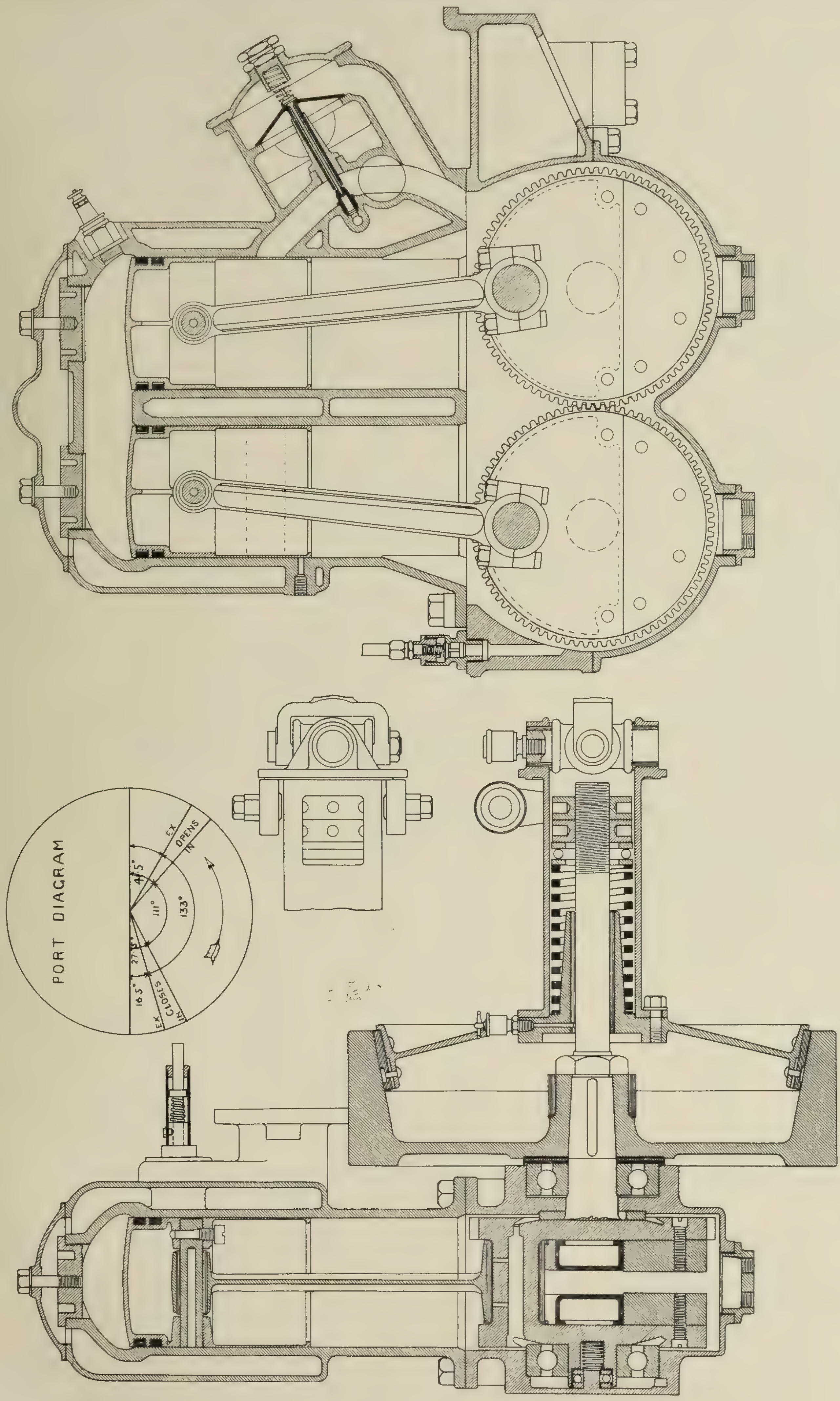


Fig. II. THE 15 H.P. VALVELESS ENGINE.

can be adjusted by altering the relative positions of the air valve and needle.

The cylinders and the carburettor are a single casting of a good grade of iron, the water jacket and carburettor covers being brass. The pistons are pressed steel machined all over, and are very light for their size, especially as they have to be unusually long so that they may seal

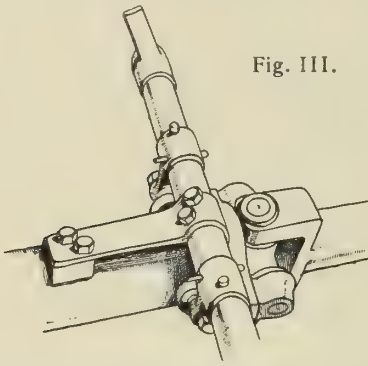


Fig. III.

the ports effectively and prevent leakage between either port and the crankcase. In the cross section of a single cylinder in Fig. II. it will also be noticed that the gudgeon pin is drilled out for lightness. Although no doubt it is an extremely rare cause of trouble, it seems regrettable that some other method of locking the pin could not have been devised, because a set screw could do a very great deal of mischief if it became detached and fell into the crankshaft gears, in fact it could not very well fail to burst the crankcase. As regards the crankshafts, one of the reasons for the use of ball bearings is the small amount of oil which they need, as has before been hinted, and we understand that no trouble has been caused by these bearings in engines which have been under observation for several years. For the gears which are cut on the crank discs, and serve to connect the two crankshafts, a case-hardened mild steel is used, and the teeth are of the single helical variety, thus thrusting the crankshafts in opposite directions. This thrust is utilised to seal the crankcase by maintaining pressure between the discs and the phosphor bronze washers which separate them from the aluminium. There can, of course, be no leakage through the outer bearings, because they are capped. In respect to the gears, it may be said at once that they are scarcely audible in working, in fact they are an excellent testimonial to the manufacturers of the chassis, who are David Brown and Sons, Ltd., of Huddersfield. Above the section of the balance weights a dotted outline may be observed, and this indicates two hollow aluminium boxes which are attached to the crank discs in order to lessen the volume of the base chamber and so increase the degree of compression obtainable.

A distributing pump is used for lubrication, and this, together with the magneto, is driven by a skew gear situated outside the crankcase at the front. Oil is forced by a plunger pump to each of six leads in turn, the pump being driven at a twelfth of the crankshaft speed and the distributer rotated at a sixth of this rate. Two leads feed gutters cut inside the crankshaft discs (*via* the ball bearings), whence oil is thrown out through the big ends, two others feed the cylinders, one feeds the gears themselves, and the last one a tell-tale drip on the dashboard. The quantity of oil fed at each stroke may be adjusted by a set screw which limits the extent of the downward—or suction—travel of the plunger, the latter being lifted by a cam and returned by a spring. An illustration of the pump and distributor was given on page 45 in *Automobile Engineering*, the annual of *The Automobile Engineer*, published in December, 1910. The non-return valve shown on the crankcase in Fig. II. is to control the supply of compressed air to the petrol tank at the rear of the chassis. Owing to the simple form of the cylinder jacket passages particularly good thermo-syphon cooling is obtained, so that a fan is but rarely necessary, though it is a standard fitting.

From the near-side crankshaft the drive passes through a normal pattern of leather-faced cone clutch shown in Fig. II., and there is a clutch brake consisting of a pad of fibre carried at the outer end of a spring arm pinned to the clutch pedal shaft. Depressing the pedal thus brings the pad in contact with the boss of the clutch, but the braking action is quite gentle, although sufficient, and cannot cause a perceptible shock. The arrangement is shown in Fig. III. From the clutch there is a double-jointed shaft 270 mm. long from centre to centre, the front joint being a ring and the rear a De Dion type coupling. It will be noticed that both these joints are very large and the effect is to reduce the liability to rattle which is always present with a pin and block pattern of coupling. The gearbox, shown in Fig. IV., has the peculiarity of being very narrow, so that the shafts are altogether exceptionally short, the unsupported length of main shaft being but 150 mm. The bearings also are of ample size for the work, their description being as follows: Mainshaft front end, two Hoffman No. 14 light type, main shaft rear end one Hoffman No. 13½ medium type, layshaft front and rear ends Hoffman No. 12 medium type. All the gears are six pitch inch diametral, and the teeth proportions are:—layshaft drive 18 to 30, first speed 15 to 33, second speed 20 to 28, third speed 25 to 23; giving gearbox ratios of 3.67, 2.33, and 1.53 to 1. The back axle reduction is 4.4 to 1, so the effective ratios for the indirect speed are 16.1, 10.3, and 6.7 to 1: rather lower perhaps than would have been anticipated. All gears are cut on a special tool made by David Brown and Sons, and are case-hardened.

The striking arrangements follow usual practice, except as regards the locking gear which is very simple and apparently effective. It is shown inset in Fig. IV., and consists of two steel balls separated by a distance piece. The central striking shaft is drilled through to take the distance piece, and the balls lie partly in holes in the dividing walls of the brass box and partly in small recesses in the side striking shafts. A little play is allowed so that one shaft can be moved at a time, and it is obvious that directly either shaft is out of centre the other two must be locked. In addition, there are spring controlled plungers working in slots in the striking rods and the cap screws for these parts are somewhat ingeniously locked by the tongue ended bolt which is shown at the top of the horizontal section of the gearbox in Fig. IV.

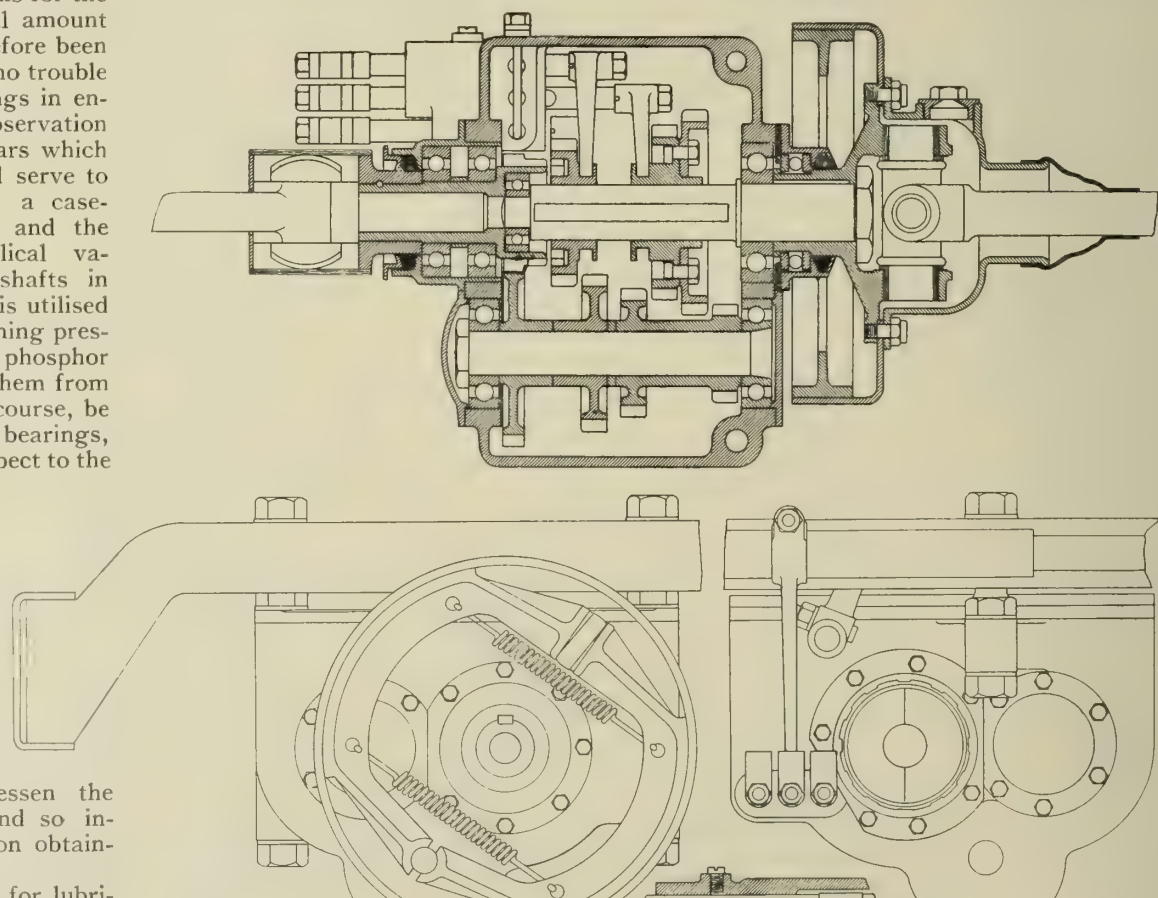


Fig. IV.

The joints of the cardan shaft are simply duplicates of those used on the clutch shaft, and the foot brake construction is explained with sufficient clearness by Fig. IV., so the next component to be dealt with is the rear axle. This is made in two patterns, with a bevel or with a worm drive, though the latter form of transmission will probably become the standard. Fig. V. shows a section of the shorter half of the axle (the differential is, of course, out of centre owing to the engine arrangement) and the outer case is malleable cast iron, the weight of the chassis being taken on the driving shafts, which have their outer bearings brought as close as possible to the planes of the wheels. As in the gearbox, so in the axle, the ball bearings are of good size and the thrusts are particularly big. Concerning this part of the axle it will be seen that there are two spacing washers, behind the thrust rings, upon the thickness of which the longitudinal location of the worm wheel depends, and to prevent these becoming interchanged by a careless repairer one is made considerably thicker than the other so that it would at once be obvious if a mistake should occur in re-assembling. By comparison with the size of the car the axle is small and should be light. While the mounting of the wheels direct upon the shafts and the placing of the differential out of centre are debatable features, they have no pronounced drawbacks on a reasonably light car, and it must not be forgotten that a wheel supported rigidly on a shaft of good diameter is much to be preferred to a wheel with bearings on the sleeve when the latter are small and close together, because under such circumstances looseness is bound to develop rather rapidly.

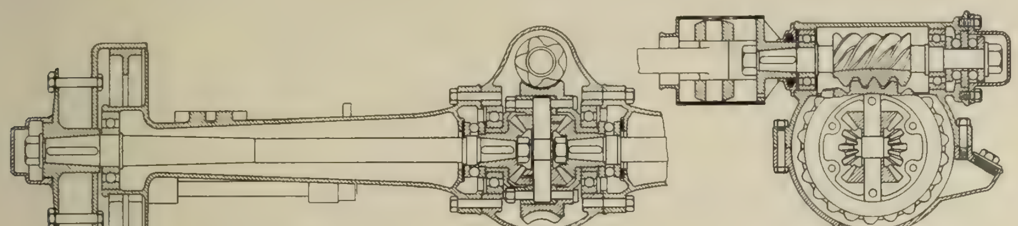


Fig. V.

The worm and wheei are of much the same type as the Dennis, and their manufacture is an interesting job. Blanks for the worms are placed in the carriage of a special machine where they are both traversed and rotated, and the full depth of thread is obtained by a single cut, only one thread being dealt with at a time. Behind the traversing carriage there is a milling cutter ground to a very little smaller than the exact form of the finished threads, the head which holds it being angularly adjustable to suit the pitch of the worm. This operation is surprisingly quick, the cutting of one thread on a worm taking only a few minutes. After being cut each worm is ground to size in a very simple manner. The threads are ground on the sides firstly, and the tops are rounded off at a second operation. For this operation the worm is mounted between centres and revolved very slowly, meanwhile a slide carrying the grinding wheel traverses on the bed. The wheel is saucer-shaped, and the grinding is done by the edge, the axis of rotation being set in precisely the same manner as that of the milling cutter in the first machine, so that the whole depth of thread is ground at once. The phosphor bronze worm wheels are hobbed from the blanks, the feed being

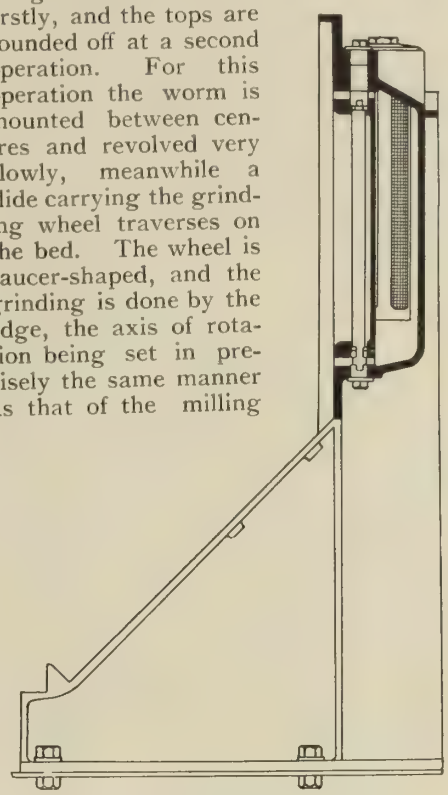
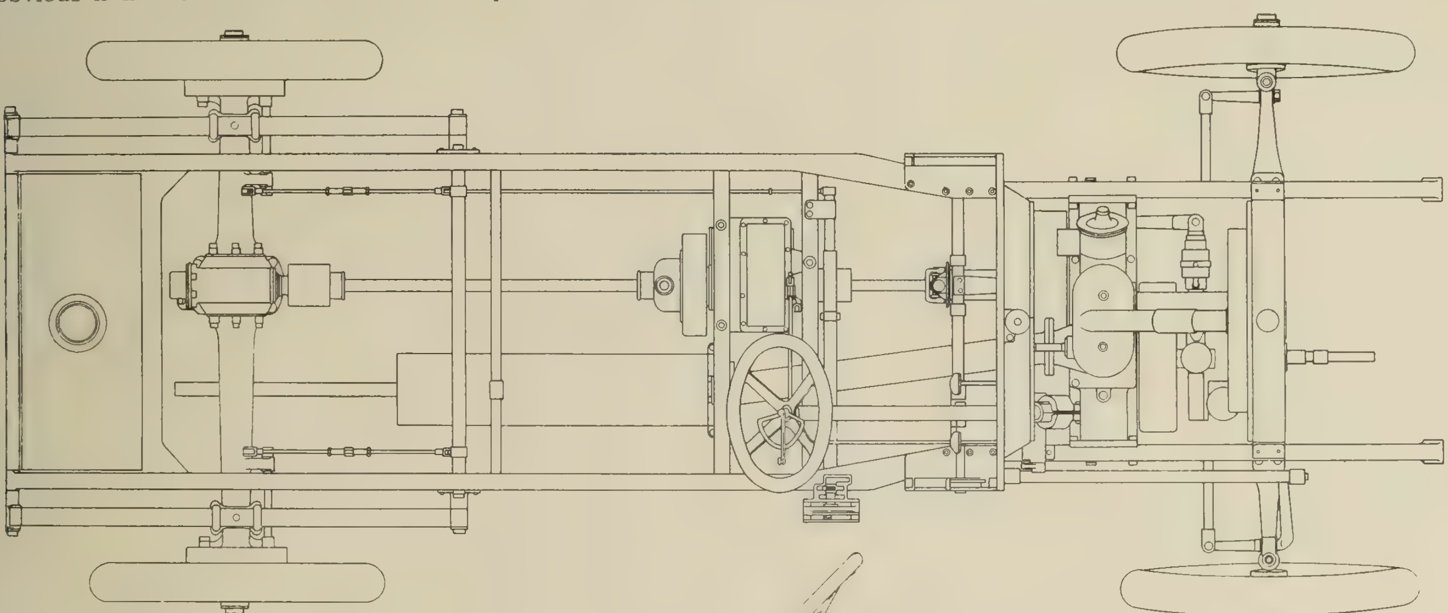


Fig. VI.



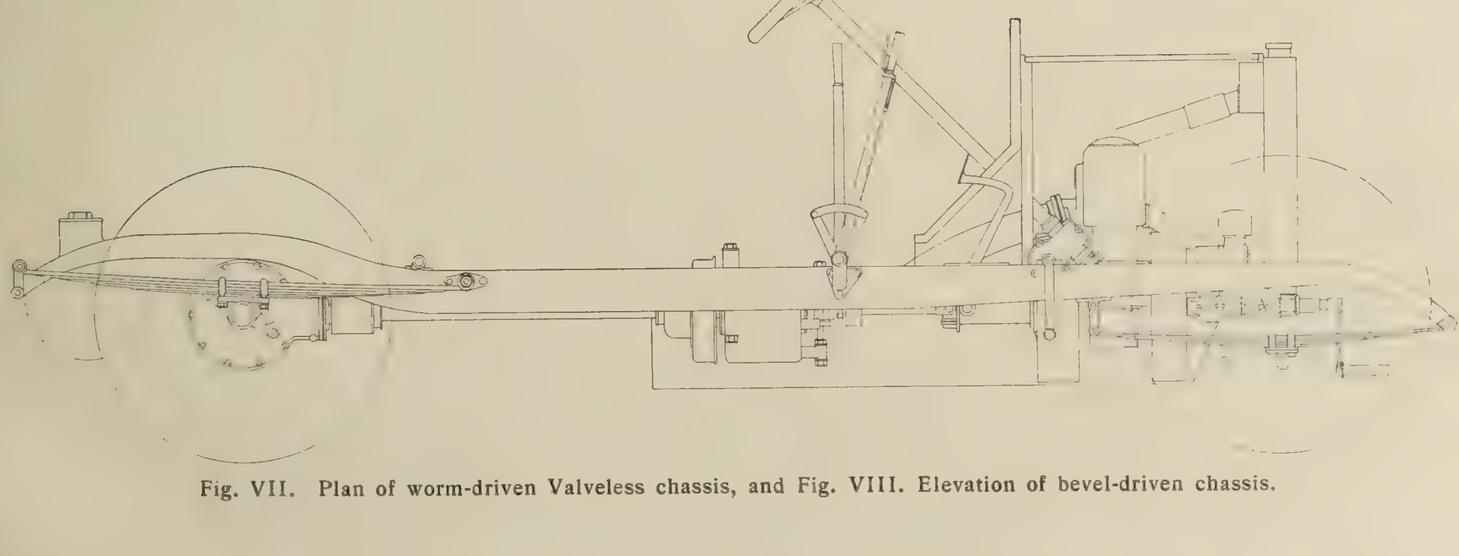


Fig. VII. Plan of worm-driven Valveless chassis, and Fig. VIII. Elevation of bevel-driven chassis.

radial, and are also finished at a surprising speed

Concerning the frame there is little to be said except to point out the way in which the gearbox is suspended from slightly arched cross members. The purpose of this form of construction is to obtain as rigid a frame as possible and to provide an easy means of lining up the gear. When working under load the box does not appear to vibrate as a whole, as is sometimes the case with a suspension attachment, while, of course, the design is cheap to make. The dashboard is a one-piece aluminium casting, including the oil tank, and it is shown in section in Fig. VI. Attachment to the frame sides is made by separately cast aluminium brackets and the cost of the whole job is said to be little, if any, more than that of the usual wooden dash. The appearance is good on account of the complete absence of sharp corners, and the rigidity is an advantage, as it has a decided strengthening effect upon the frame.

Details generally have been well cared for, and when the car is on the road it is noticeable that the smaller noises are conspicuously absent. The gears are rather quiet than otherwise, and the change is easy, the clutch brake appearing to exert just the requisite amount of power. That the whole chassis is a very clean job can be seen in Figs. VII. and VIII., which show a plan of the worm driven type and an elevation of the bevel pattern respectively, and although a certain amount of detail is absent, such as the oil leads, the omissions are not of any importance.

THE RATING OF PETROL ENGINES.

Report of the Horse-Power Formula Committee of the Institution of Automobile Engineers.

The Committee consisted of Mr. Dugald Clerk, F.R.S., M.Inst.C.E., (chairman), Messrs. A. Craig, J. S. Critchley, C. R. Garrard, L. H. Hounsfield, Max R. Lawrence, Mervyn O'Gorman,* L. H. Pomeroy, and D. J. Smith (representing the Institution of Automobile Engineers), Col. H. C. L. Holden, Capt. R. K. Bagnall Wild, Prof. Callender, Dr. W. Watson, Messrs. W. Worby Beaumont, and E. Russell Clarke (representing the R.A.C.), and Mr. G. A. Burls, M.Inst.C.E. (representing the S.M.M. & T.)

The report of the Rating Committee of the Institution was read and discussed at a meeting held on November 10th, 1908. It dealt with proposals put forward on behalf of the Society of Motor Manufacturers and Traders, by Mr. G. A. Burls, M.Inst.C.E. The Rating Committee agreed with the proposals in principle, but considered that the tests submitted did not support the modifications of the R.A.C. rating required by the formulæ. The report recommended the formation of a committee composed of members of the Institution, members of the R.A.C., and representatives of the Society of Motor Manufacturers and Traders. This committee was called "The Horse-Power Formula Committee." Several meetings were held, and a scheme was agreed upon, Mr. G. A. Burls being good enough to undertake the collection of the necessary material from the leading automobile firms in this country and abroad; he has now prepared the tables of particulars which accompany this report, and made the many calculations and deductions shown. Mr. Burls' letter to the chairman of the committee accompanies the report, and it describes the nature of his work. At meetings held on the 30th June and the 13th October, 1909, the committee resolved to recommend for consideration the following formula :—

$$K\ d\ (ad+s)\ N$$

as a formula giving a rating with a stroke-bore correction. The constants K and a were purposely left without any specified value to enable the committee ultimately to arrive at values from experiment and observation. In this formula, d is the bore, and s the stroke, in inches; N is the number of cylinders.

Before arriving at the above formula the committee considered the two main corrections proposed to be applied to the R.A.C. formula, viz., variation of mean pressure with dimensions of cylinder, and variation of piston speed with ratio of stroke to bore. The R.A.C. formula corresponds to a mean effective pressure of 67.2 lb. per square inch, per explosion, with a normal piston speed of 1,000 ft. per min. According to the R.A.C. formula, the rating $=0.4\ D^2 \times N$. This simple formula, notwithstanding frequent statements to the contrary, involves both specified mean pressure and piston speed. It assumed, however, that piston speeds did not vary materially from 1,000 ft. per min. Examination of the accompanying tables of results proves conclusively that piston speed does increase with the stroke-bore ratio.

The tables give some particulars from tests made of 144 engines, but only 101 tests contain all the data required for comparing the effect of stroke-bore ratio on piston speed for values of r from 1 to 1.61, and they do not necessarily represent the maximum brake horse-power which can be obtained. The following table shows broadly the increase of piston speed with stroke-bore ratio. Five groups have been taken with a variation in each of 0.1 in stroke-bore ratio as nearly as could be obtained.

TABLE I.

Change of Piston Speed with variation of Stroke-bore.
Ratio $r=s/d$.
Highest recorded b.h.p.

Number of Tests.	r.	Piston Speed at Max. B.H.P. ft. per min.
15	1·00 to 1·08	1,303
30	1·10 to 1·20	1,240
24	1·21 to 1·30	1,385
25	1·33 to 1·44	1,414
7	1·50 to 1·61	1,597
101		

Each value of piston speed is the mean of the values obtained in the number of tests given; to some extent, the taking of average results masks the variation between the different engines. Accordingly, Fig. I. has been prepared, in which piston speeds are plotted against stroke-bore ratio for the 101 engines.

The results plotted in Fig. I. show an undoubted increase of piston speed with stroke-bore ratio, but the results obtained are exceedingly irregular. For example, between $r=1$ and $r=1.1$ the maximum piston speed in three cases lies between 1,600 and 1,700 ft. per minute, but the very large number of tests show 1,400 down to about 1,000 ft. per minute. Between $r=1.1$ and $r=1.2$ the maximum values appear to be 1,400 and 1,500 ft. per minute in two separate tests, but the greater number lie below 1,300 ft. per minute. Between $r=1.2$ and $r=1.3$ the maximum values are above 1,600 ft. per minute, but the greater number lie below 1,400 ft. per minute. The highest value of all is found between $r=1.3$ and $r=1.4$, namely, 2,200 ft. per minute—but it is understood that this figure is open to doubt. In the same way between $r=1.4$ and $r=1.5$ the highest value is just over 2,000 ft. per minute, and the highest value at 1.6 is 2,100 feet per minute. From this it will be seen that it is impossible to formulate accurately any law of variation between stroke-bore ratio and piston speed Notwithstanding this, if 1,200 ft. per minute be taken as the speed to be expected with $r=1$, and 2,100 ft. per minute that corresponding to $r=2.5$, then, assuming a linear law between the two points, the piston speeds found for different stroke-bore ratios are such as can be obtained (bench test) at maximum brake horse-power from a carefully designed engine. These piston speeds, it will be evident, are higher than those given by about 50 per cent. of the engines examined. About 50 per cent., however, give greater values, so that for a maximum rating no hardship would be occasioned by assuming the law to be that given by the black line shown upon Fig. I. Any formula required to express b.h.p. on this basis must assume a piston speed of 1,200 ft. per minute for $r=1$ and 2,100 ft. per minute for $r=2.5$.

Forty of the engines, Nos. 1 to 50, of the appended tables were tested at approximately 0.9 of the highest brake horse-power recorded, and in all cases the tests show lower piston speed. These results have been plotted in Fig. II.

Here also the stroke-bore ratio increase is accompanied by increase in piston speed, but the piston speed for $r=1$ may reasonably be taken at 900 ft. per min. and for $r=2.5$ at 1,800 ft. per min.

Mean Effective Pressure corresponding to the Brake Horse-Power.

This value has been called ηp in the previous Reports. The results of the various tests have been tabulated and plotted against cylinder bore, but no increase can be deduced in this way as due to increasing cylinder bore. When, however, only one make of engine is taken the case is different, and the rise of ηp with d is apparent.

Fig. III. shows eighty-eight tests in which ηp at maximum brake horse-power is plotted against cylinder diameter. The values considered are those given by tests from 124 engines. Out of these, eighty-eight give mean effective pressures of 65 lb. per sq. in. and above. The pressures

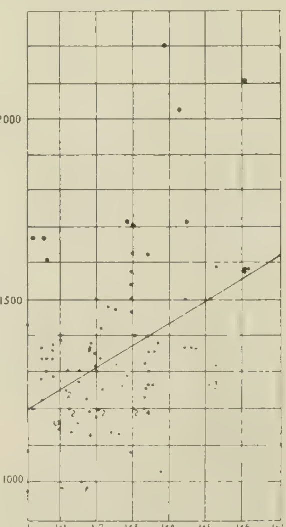


Fig. I.

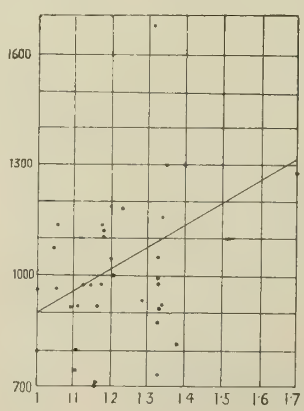


Fig. II.

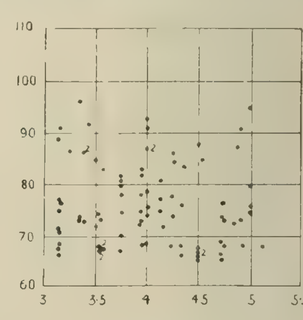


Fig. III.

Note that the base lines in these Figs. are not zero lines.

FIG. I.—Piston speed in ft. per min. plotted against stroke-bore ratio at max. b.h.p

FIG. II.—Piston speed plotted against stroke-bore ratio at about 0·9 max. b.h.p.

FIG. III.—Mean effective pressure corresponding to b.h.p. plotted against cylinder bore.

*Mr. O'Gorman also represented the R.A.C.

Speeds.			Estimated Abs. Compression Pressure $p v^{1\cdot3} =$ const., lb. per sq. in.	B.H.P. by Test at Various Piston Speeds.											
σ at ximum .H.P. corded.	Limiting $\sigma = 650 \sqrt{\frac{d^2 s}{m}}$ ft. p.m.	n r.p.m. corre-sponding to the Limiting Value of σ.		σ =	B.H.P. =	σ =	B.H.P. =	σ =	B.H.P. =	σ =	B.H.P. =	σ =	B.H.P. =	σ =	B.H.P. =
1200	2710	3610	119	600	8.1	750	10.3	900	11.6	1050	14.9	1200	15.6		
1200	2620	3490	119	600	16.3	750	20.4	900	23.0	1050	26.6	1200	29.8		
1130	2460	3470	119	567	12.2	710	15.8	850	17.7	995	19.2	1130	22.2		
	2600	3120	89												
	2600	3120	89												
1425	2365	3150	89	525	29.5	750	42.0	1050	56.2	1275	62.2	1425	63.0		
1200			70	492	2.8	787	4.1	1082	5.0	1278	5.0	1476	4.8		
1200			70	492	5.5	787	8.2	1082	10.0	1278	10.0	1476	9.6		
			70												
1300			70	492	11.1	787	16.4	1082	19.7	1278	20.2	1476	19.4		
			70												
1574			75	492	11.4	787	18.3	1082	25.0	1377	30.3	1574	30.7		
1400			75	492	18.3	787	27.4	1082	33.7	1377	36.8	1476	36.8		
1620			80	622	19.7	938	31.3	1108	37.4	1280	43.3	1450	47.3	1620	49.2
1375			89	492	21.7	787	35.2	984	40.8	1180	44.8	1377	46.3	1476	46.0
1375			89	492	29.6	787	43.8	984	50.6	1180	55.0	1377	55.8	1476	54.7
			89	984	59.5										
1315	1890	2880	96	525	9	788	13	1050	17	1315	21				
1263	1950	2470	96	632	14	948	20	1263	25						
1365	2070	2430	96	683	22	1024	30	1365	38						
1288	2150	2340	96	735	35	1103	48	1288	54						
1365			113	683	38	1025	58	1365	70						
			119												
			116												
	2240	2690	89	666	31.5	833	35.0	1000	43.3						
1225	2330	2660	121	525	25	700	35	875	42	1050	45	1136	46	1225	47
1136	2100	2400	121	350	13	525	19.5	700	26	875	32	1050	35	1136	36
1125	1945	2600	127	450	10	600	14	750	18	900	21.5	1050	23.5	1125	24 . .
1260	2370	3900	96	630	5.5	788	7.5	947	8.5	1104	10	1260	11		
1260	2300	2920	96	630	12	788	15	947	16.5	1104	18	1260	20		
1365	2120	2490	89	683	17.5	853	22	1024	25.5	1194	28	1365	30		
1365	2120	2490	89	683	24	853	29	1024	34	1194	39	1365	41		
1200	2090	2450	89	683	23	853	27	1024	30	1194	32				
1200	2090	2450	89	683	32	853	38	1024	42	1194	45				
1287	2180	2370	87	736	28	920	34	1103	37.5	1287	39				
1183	2090	2120	84	788	38	987	44	1183	46						
1300	2160	2730	95	593	39	792	48	990	53	1188	55.3	1385	55.3	1583	52.5
				450	16.9	600	21.8	750	26.7	900	27.9				
				500	7.47	667	8.74	833	9.33	1000	9.56				
1300	2420	3070	104	632	5.78	947	7.56	1263	9.28	1578	8.80				
1705	2470	2900	95	682	5.6	1023	8.2	1364	9.3	1705	10.0				
2100	3250	3100	110	840	8.8	1260	13.2	1680	15.6	2100	16.0				
1314	1735	2640	104	526	6.0	788	9.6	1050	12.0	1314	12.6				
1578	1975	2500	95	632	10	947	15.6	1263	17.6	1578	18.0				
1300	1905	2420	95	632	16.5	947	23.4	1263	26.0	1578	24.0				
1400	1975	2320	95	682	20	1023	27.4	1364	31	1705	29				
1364	2220	2580	95	682	35	1023	42.5	1364	48						
1263	1800	2280	95	632	33.6	947	44.4	1263	50						
1416	1860	1575	95	1416	75.2										
	2050	2820	89	735	18	875	21.6								

mely small, data possibly inaccurate.

TABLE II.—*Continued.*

Values of m., etc., from equation (1). Cast-iron pistons.

$r =$	$d.$ in inches.	$d^2s = d3r =$	$m.$ in lbs.	$d^2s/m =$	$\sqrt{d^2s/m} =$
	2.5	27.3	3.08	8.87	2.98
	3	47.3	4.22	11.22	3.35
1.75	3.5	75	5.8	12.93	3.60
	4	112	8	14	3.74
	4.5	159.5	10.7	14.9	3.86
	5	218.8	14.1	15.5	3.94
	2.5	31.25	3.13	10	3.16
2	3	54	4.3	12.6	3.54
	3.5	85.8	6	14.3	3.78
	4	128	8.2	15.6	3.95
	2.5	39.1	3.22	12.15	3.49
2.5	3	67.5	4.5	15.1	3.89
	3.5	107.2	6.2	17.3	4.17

TABLE IIIa.,

showing comparison of calculated with actual mass of reciprocating parts from the formula :

$$m = 0.05\ d3\ (1 + 0.15r) + 1.5 \text{ in lb.} \qquad (2)$$

(For pressed steel pistons.)

$d =$ inches.	$s =$ inches.	$r = s/d.$	$d^2s =$	$m.$ actual.	$m.$ calculated.
3.13	4.75	1.52	46.5	3.43	3.38
3.15	4.73	1.5	46.9	3.53	3.41
3.38	4.25	1.26	48.6	3.38	3.8
3.35	5	1.49	56.1	3.5	3.8
3.75	4.5	1.2	63.2	3.63	4.6
3.75	4.5	1.2	63.2	3.88	4.6
3.94	4.73	1.2	73.5	5.3	5.1
3.94	5.12	1.3	79.3	5.52	5.15
3.94	6.3	1.6	97.8	3.91 ?	5.28
4	5.5	1.38	88	4.87	5.12
4	7	1.75	112	5.5	5.53

TABLE III.,

showing comparison of calculated with actual mass of reciprocating parts from the formula :

$$m = 0.08\ d3\ (1 + 0.15r) + 1.5 \text{ in lb.} \qquad (1)$$

(For cast-iron pistons.)

d – inches.	s inches.	$r = s/d.$	$d^2s =$	$m.$ actual.	$m.$ calculated.
2.44	4.33	1.78	25.8	3.2	2.97
2.56	4.33	1.69	28.4	3.24	3.18
2.95	4.73	1.6	41.2	4.6	4.04
3.13	4.5	1.44	44.1	3.72	4.48
3.15	3.94	1.25	39.1	4.63	4.47
3.50	4.5	1.29	55.2	5.25	5.6
3.56	4.75	1.33	60.2	5.5	5.8
3.74	4.73	1.27	66.2	8.1	6.5
4	4.38	1.1	70.1	7.25	7.5
4	5	1.25	80	6.72	7.6
4.25	5.25	1.24	94.8	9.12	8.8
4.33	5.52	1.28	103.5	9.25	9.25
4.5	5	1.11	101.2	9.5	10
4.5	6	1.33	121.4	9.5	10.25
4.63	5	1.08	107.3	13.12	10.75
4.73	5.12	1.08	114.5	9.87	11.35
4.75	5	1.05	112.6	12.22	11.4
4.92	5.92	1.2	143	13.8	12.7
5	5.25	1.05	131	10.5	13.1
5	5.13	1.03	128.2	12.25	13
5.12	5.52	1.08	144.4	13.2	13.8
5.92	7.08	1.2	248	30.4	21
7.25	7.5	1.03	394	32.5	36.7
8	8	1	512	54	48.6
12	8	.67	1152	151	153.5

These tables accompany the paper on "Horse-power Rating," read before the Institution of Automobile Engineers by G. A. Burls, and reported on pages 295-298.

TABLES OF ENGINE DATA REFERRED TO IN

(See

No. of Engine	Bore in Inches = *d*.	Stroke in Inches = *s*.	Stroke-bore Ratio, *r* = *s*/*d*.	No. of Cylinders. N.	Volume Ratio of Compression.	Inlet and Exhaust Valves. Diameter and Lift of Inlets. δin. × λin.	Diameter and Lift of Exhausts. δin. × λin.	Mean Velocity through Inlets at Maximum H.P. = *v*, ft. p.m.	Value of δ given by $.29d \sqrt[4]{\frac{d^2s}{m}}$ inches.	Mass of Reciprocating Parts in lb. = *m*.	d^2s =	$\sqrt{\frac{d^2s}{m}}$ =	η *p* at Maximum B.H.P., lb. per sq. in.
1	3.94	5.12	1.30	4	4.42	1.97 × .296	1.97 × .296	1700	2.04	7.63	79.3	3.22	69.3
2	3.50	4.02	1.15	4							49.3		59.3
3*a*	4.92	5.0	1.02	6							120.8		72.6
4	4.73	5.0	1.06	4	4.73	2.0 × .31	2.0 × .41	>1870	2.39	12.22	112.0	3.03	66.3
5	4.13	5.0	1.21	4	4.9	1.75 × .22	1.75 × .31	>1980	2.17	7.88	85.3	3.29	57.3
6*b*	3.74	5.32	1.42	4	4.0	1.97 × .315	1.97 × .315	1840	1.96	7.00	74.3	3.26	79.6
7*c*	3.78	5.12	1.35	4							73.2		
8*c*	4.88	5.12	1.05	4							122		
9	3.35	4.33	1.29	4							48.7		72.7
10	4.0	4.53	1.13	4							72.5		75.2
11	4.17	4.57	1.10	4							79.2		71.2
12	4.73	5.12	1.08	4							114.7		73.3
13	4.88	5.12	1.05	4							122		86.7
14*a*	4.73	5.00	1.06	6							112		68.2
15	4.25	5.00	1.18	4	3.52	1.75 × .375	1.75 × .375	1450	2.26	8.06	90.3	3.35	77.3
16	4.5	6.00	1.33	4							121.5		59.0
17	3.75	3.75	1.0	4	3.78	1.875 × .31	1.875 × .31	> 975	1.77	7.56	52.8	2.65	69.8
18	4.5	5.0	1.11	6							101.2		66.7
19*d*	12.0	8.0	0.67	4	3.18	5.0 × .81	5.0 × .81	>1375	5.78	151	1152	2.76	62.5
20*d*	8.0	8.0	1.0	4	3.07	3.38 × .88	3.38 × .88	>1850	4.08	54.0	512	3.08	59.7
21	3.28	3.95	1.20	4		1.38 × .31	1.38 × .31	>1860			42.4		59.4
22	3.82	4.33	1.13	1	5.31	1.50 × .28	1.50 × .28		1.69	6.44	63.3	3.14	
23	4.5	5.12	1.14	1	4.54	1.81 × .28	1.81 × .28		2.03	9.69	103.7	3.27	
24	3.82	4.33	1.14	2	5.0	1.50 × .28	1.50 × .28				63.3		
25	3.35	4.33	1.29	4	4.74	1.46 × .28	1.46 × .28		1.46	5.25	48.7	3.05	
26	3.82	4.33	1.14	4	5.0	1.50 × .28	1.50 × .28				63.3		
27	3.38	4.00	1.19	4	4.8	1.50 × .31	1.50 × .31	1715	1.77	4.25	45.7	3.28	85.8
28	3.38	4.00	1.19	6	4.8	1.50 × .31	1.50 × .31	1715	1.77	4.25	45.7	3.28	85.8
29	3.53	4.75	1.35	4	4.8	1.63 × .38	1.63 × .38	2050	1.86	5.38	59.2	3.32	72.3
30	4.0	4.92	1.23	4	5.0						78.7		78.3
31	3.23	4.33	1.34	4	5.0						45.2		
32	4.88	5.12	1.05	4	4.75						122		
33	3.15	4.33	1.38	4	4.0	1.18 × .28	1.18 × .28	>1825			43.0		76.4
34	3.74	4.33	1.16	4	4.0	1.42 × .28	1.42 × .28	>1730			60.6		81.2
35	4.33	5.12	1.18	4	4.2	1.73 × .34	1.73 × .34	>1880			96.2		83.2
36	4.73	5.52	1.17	6	4.0	1.73 × .34	1.73 × .34	>2290			123.6		65.0
37	4.5	6.0	1.33	1	3.5	1.75 × .44	1.75 × .44	>1985	2.47	9.5	121.4	3.57	65.7
38	4.5	6.0	1.33	2	3.5	1.75 × .44	1.75 × .44	>1985	2.47	9.5	121.4	3.57	65.0
39	4.5	6.0	1.33	4	3.5	1.75 × .44	1.75 × .44	>1985	2.47	9.5	121.4	3.57	66.7
40	4.5	6.0	1.33	6	3.5	1.75 × .44	1.75 × .44	>1985	2.47	9.5	121.4	3.57	67.0
41	6.0	8.0	1.33	4	3.5	2.50 × .56	2.50 × .63	>1765	3.03	31.5	288	3.03	56.3
42	3.75	4.5	1.20	4	3.5	1.75 × .31	1.75 × .31	1725	1.96	6.0	63.2	3.25	47.8
43	4.5	5.0	1.11	4	3.78	1.75 × .44	1.75 × .44	>2070	2.35	9.5	101.2	3.26	58.2
44*e*	4.5	5.0	1.11	6	3.78	1.75 × .44	1.75 × .44	>2070	2.35	9.5	101.2	3.26	51.8
45*f*	4.0	7.0	1.75	4	4.26	2.25 × .63	2.25 × .63	>1440	2.47	5.5	112	4.52	92.3
46	2.95	4.73	1.60	4	4.5	0.95 × .23	0.95 × .26	>3890 ?	1.48	4.6	41.2	2.99	55.3
47	3.74	4.73	1.27	4	4.5	1.26 × .25	1.26 × .28	>2780	1.83	8.1	66.2	2.86	66.8
48	4.33	5.12	1.18	4	4.5	1.73 × .25	1.73 × .28	>2140	2.20	10.2	96.2	3.07	65.6
49	3.54	4.73	1.34	2	3.23	1.75 × .25	1.75 × .25	1380	1.77	6.8	59.3	2.96	64.7
50	3.54	4.73	1.34	4	4.77	1.77 × .30	1.77 × .30	1350	1.83	5.94	59.3	3.16	63.5
51	3.94	5.12	1.30	4	4.42	1.97 × .296	1.97 × .296	1535	2.06	7.63	79.3	3.23	74.2
52	2.44	4.33	1.78	4	4.1	1.02 × .19	1.02 × .22	2065	1.19	3.2	25.8	2.84	53.8
53	2.56	4.33	1.69	4	4.5	1.02 × .19	1.02 × .22	2280	1.28	3.24	28.4	2.96	57.7
54	2.95	4.73	1.61	4	4.5	1.18 × .23	1.18 × .26	2470	1.50	4.43	41.2	3.05	55.2
55	4.13	4.73	1.15	4	4.84	1.75 × .31	1.75 × .31	1820	2.11	8.38	89.8	3.09	74.7
56	4.53	5.12	1.13	4	3.95	1.69 × .38	1.69 × .38		2.22	12.88	105.1	2.86	84.2
57	3.54	4.73	1.34	4	4.43	1.50 × .234	1.50 × .234	1950	1.76	6.88	59.2	2.93	66.2
58	3.25	5.0	1.54	4	5.0	1.50 × .344	1.50 × .344		1.56	7.13	52.8	2.72	86.0
59	3.5	5.0	1.43	4	5.0	1.50 × .344	1.50 × .344		1.72	7.56	61.3	2.85	84.5
60	4.0	5.0	1.25	4	5.0	1.50 × .344	1.50 × .344		1.99	9.13	80.0	2.96	86.7
61	4.25	5.0	1.18	4	5.0	1.75 × .344	1.75 × .344		2.14	10.0	90.2	3.01	83.8
62	3.15	4.73	1.50	4	4.0	1.58 × .276	1.58 × .276	1490	1.63	4.69	46.9	3.16	76.3
63	3.74	5.32	1.42	4	4.0	1.97 × .315	1.97 × .315	1820	1.96	7.0	74.3	3.26	80.5
64	3.5	4.5	1.29	4	4.76	1.625 × .22	1.625 × .25	1730	1.83	5.25	55.2	3.24	59.3
65	4.13	5.0	1.21	4	4.9	1.75 × .22	1.75 × .31	1860	2.18	7.88	85.4	3.30	59.1
66	4.75	5.0	1.05	4	4.73	2.0 × .31	2.0 × .41	1880	2.4	12.22	112.6	3.04	72.8
67	4.75	5.0	1.05	6	4.73	2.0 × .31	2.0 × .41	2350	2.4	12.22	112.6	3.04	67.2
68	3.15	4.73	1.50	4	4.0	1.42 × .28	1.42 × .28	1550	1.58	5.2	46.9	3.01	53.8
69	4.42	5.52	1.25	4	4.0	1.81 × .28	1.81 × .28	2190	2.33	9.8	107.6	3.31	52.8
70	3.56	5.12	1.44	4	5.1	1.25 × .38	1.25 × .38	2775	1.79	7.19	64.8	3.01	82.5
71	4.5	5.0	1.11	4	4.13	1.63 × .38	1.63 × .38	2220	2.27	11.1	101.2	3.02	87.3
72	5.0	5.0	1.0	4	3.95	1.63 × .38	1.63 × .38	2560	2.6	12.13	125	3.21	94.6
73	5.0	5.0	1.0	6	3.95	1.63 × .38	1.63 × .38		2.6	12.13	125	3.21	79.6
74	3.13	4.5	1.44	4	4.15	1.25 × .22	1.25 × .31	2345	1.69	3.72	44.1	3.44	54.5
75	3.56	4.75	1.33	4	4.15	1.38 × .22	1.38 × .31	2080	1.88	5.5	60.2	3.31	67.0
76	4.0	5.13	1.28	4	4.15	1.63 × .28	1.63 × .38	2575	2.1	7.63	82.2	3.28	50.0
77	3.56	5.13	1.44	6	4.15	1.38 × .22	1.38 × .31	2850	1.87	5.97	65.0	3.30	60.8
78	4.63	5.0	1.08	4	4.15	2.0 × .28	2.0 × .38	1700	2.27	13.12	107.3	2.86	66.2
79	5.0	5.13	1.03	4	4.15	2.0 × .28	2.0 × .38	1875	2.61	12.25	128.3	3.24	74.2
80	4.25	5.5	1.29	6	4.15	2.0 × .28	2.0 × .38	1655	2.27	8.68	99.2	3.38	67.7
81	4.0	3.0	0.75	4	4.22	1.38 × .31	1.38 × .31	2480	1.89	6.75	48	2.67	63.4
82	4.0	3.0	0.75	6	4.22	1.38 × .31	1.38 × .31	2110	1.89	6.75	48	2.67	74.6
83	3.13	4.75	1.52	4	3.1	1.63 × .25	1.63 × .25	1460	1.74	3.43	46.5	3.68	88.0
84	4.0	5.5	1.38	4	3.0	1.88 × .31	1.88 × .31	2490	2.39	4.87	88.0	4.25	73.6
85	4.13	5.0	1.21	1	4.17	2.0 × .22	2.0 × .25		2.21	7.31	85.3	3.42	80.2
86	3.35	4.0	1.19	4	5.0	1.44 × .30	1.44 × .33		1.66	5.25	44.8	2.92	95.7
87	4.0	4.38	1.10	2	4.0	1.63 × .32	1.63 × .35		2.05	7.25	70.1	3.11	85.7
88	4.0	4.38	1.10	4	4.0	1.63 × .32	1.63 × .35		2.01	7.81	70.1	3.0	90.2
89	3.5	4.75	1.35	4	4.09	1.38 × .31	1.38 × .31	2215			58.2		71.2
90	4.13	5.12	1.24	4	4.12	1.44 × .31	1.63 × .31				87.4		76.8
91*g*	5.0	5.13	1.03	4	3.85	2.13 × .38	2.13 × .38		2.29	20.5	128.2	2.5	75.2
92	4.25	5.0	1.18	2	4.24	1.88 × .31	1.88 × .31	1435			90.2		85.7
93	4.25	5.0	1.18	4	4.24	1.88 × .31	1.88 × .31				90.2		
94	3.42	3.94	1.15	4	5.12	1.38 × .25	1.38 × .28	1770	1.65	6.0	46.1	2.77	91.4

a Racing engine. *b* Sunbeam. *c* Knight engine, R.A.C. trial. *d* Marine engine; connecting rod centres 22in. and 20in.

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ge 295.)

Speeds.			Estimated Abs. Compression Pressure	B.H.P. by Test at Various Piston Speeds.											
σ at Maximum B.H.P. Recorded.	Limiting σ = 650 $\sqrt{d^2 s}/m$ ft. p.m.	n r.p.m. corresponding to the Limiting Value of σ.	$p v^{1\cdot 3}$ = const., lb. per sq. in.	σ =	B.H.P. =	σ =	B.H.P. =	σ =	B.H.P. =	σ =	B.H.P. =	σ =	B.H.P. =	σ =	B.H.P =
1620	2090	2450	101	510	20	767	26.3	1025	32.0	1275	37.0	1535	41.5	1705	41.5
1270				402	9	670	17	937	21	1205	22	1340	22		
1666				292	22	667	53	1000	76	1333	91	1500	96	1666	104
1333	1970	2360	110	292	12	747	37	1082	45	1333	47				
1417	2140	2570	116	290	8	500	18	833	28	1166	32	1417	33		
2040	2120	2400	89	798	23.5	1063	33.5	1330	40	1595	46.5	1770	50	2040	54
				1195	38.8										
				1025	54.3										
1082				722	16	866	18	1082	21						
1134				757	23.5	908	27.7	1134	32.5						
1140				760	27	913	30	1140	33.5						
1280				853	37	1024	43	1280	50						
1280				853	45	1024	55	1280	63						
1600				509	38	833	64	1165	80	1416	87	1600	87	1665	86
985	2175	2610	75	840	25.5	893	26.8	960	32.0	983	32.7	1035	30		
1320				584	24	900	35	1160	37	1320	37.5				
975	1720	2760	83	437	9.6	562	12.3	762	20.1	975	22.8				
983				550	31.7	750	42.7	913	45.2	983	47.3				
955	1790	1340	62	893	190	920	198.5	933	202	955	204.5				
1200	2000	1500	62	973	97	1053	105	1120	108	1200	109				
1316				790	12	1053	16	1316	20						
			120												
			104												
			119												
			110												
			119												
1350	2130	3200	112	533	9.0	800	19.5	1067	27.8	1333	31.5	1667	27.5		
1350	2130	3200	112	533	13.5	800	28.2	1067	41.7	1333	47.2	1667	41.2		
1750	2160	2740	112	712	15.8	948	25.0	1266	33.5	1580	37.5	1900	37.5		
1475			119	575	20	900	32.5	1230	40.8	1475	44				
			119												
			111												
1025			89	810	16.0	937	17.3	1025	18.5						
1000			89	867	25.6	953	26.3	1000	27.0						
1195			94	1023	41.0	1110	43.0	1195	44.4						
1223			89	1075	60.5	1215	61.8	1223	63.6						
1200	2320	2220	75	800	7.8	1000	9.0	1100	9.4						
1200	2320	2220	75	800	15.7	1000	18	1200	18.8						
1200	2320	2320	75	800	28.3	1000	34.8	1200	38.5						
1200	2320	2320	75	890	46	1000	53	1140	57.5	1200	58.2				
1225	1970	1480	75	960	50	1040	52.5	1145	56	1225	59				
1500	2110	2820	75	600	14.5	900	19.6	1200	23.8	1500	24.0				
1250	2120	2540	83	665	25.8	1000	33.3	1250	35.0						
1250	2120	2540	83	549	30.7	665	37	833	43.1	1000	46.7				
1820	2940	2520	95	1320	46.9	1480	51.7	1560	52.8	1820	63.9				
1575	1940	2460	104	790	11	950	13	1260	16	1575	18				
1260	1860	2360	104	790	19	950	24	1260	28						
1365	2000	2350	104	855	28	1025	34	1365	40						
1350	1920	2440	68	632	8	946	12	1262	13	1420	13				
1350	2050	2600	111	632	15	946	21	1262	24	1420	25.5				
1535	2100	2460	101	855	28	1020	31	1365	35	1535	42				
1445	1845	2560	91	723	7	867	8	1155	10	1445	11				
1445	1920	2660	104	723	8.5	867	9.5	1155	11.5	1445	13.0				
1580	1980	2510	104	790	10	947	13	1260	16	1580	18				
1300	2010	2550	11	946	31.2	1025	35.2	1183	38.2	1260	39.3	1340	39.3	1420	38.4
	1869	2180	87	853	35.0										
1400	1930	2410	102	1025	23.8	1183	26.0	1340	27.3	1420	27.6	1500	27.5	1575	26.8
	1770	2125	119	833	18										
	1850	2220	119	833	20.5										
	1920	2310	119	833	27.5										
	1960	2350	119	833	30.0										
1500	2050	2600	89	473	9.0	867	19.2	1105	23.3	1262	25.4	1420	27.0	1577	26.8
2018	2120	2395	89	797	24.0	1063	33.5	1328	40.5	1595	47.0	1860	53.0	2018	54.0
1500	2110	2810	111	600	14	900	20	1200	23.6	1500	26				
1333	2140	2570	116	667	24	1000	29	1333	32						
1333	1975	2370	110	667	33	1000	46	1333	52						
1666	1975	2370	110	667	54	1000	74	1333	86	1666	90				
1260	1955	2480	89	632	10	950	13	1260	16						
1470	2150	2340	89	735	20	1105	30	1470	36						
1366	1955	2390	122	683	16	1024	26	1366	34						
1167	1969	2350	91	657	29.5	1000	43.5	1167	49.0						
1083	2090	2510	87	667	37.5	1000	57	1083	61						
	2090	2510	87	667	59	1000	71								
1500	2240	2990	93	450	8	600	11	900	15.3	1200	18.2	1500	19.0		
1250	2150	2720	93	475	13	633	17.3	950	23	1265	25.2	1425	24.0		
1710	2130	2490	93	513	16.8	683	22	1025	29	1367	31.5	1710	32.5		
1710	2140	2500	93	513	19	683	27	1025	38	1367	46	1710	47		
1333	1860	2230	93	500	23	667	29	1000	38.2	1333	44	1670	37		
1200	2110	2470	93	513	27	683	36	1025	49.3	1194	53	1367	49		
1466	2200	2400	93	550	32	733	42	1100	56	1466	64				
1180	1735	3470	95	520	16.8	675	21.0	780	23.0	870	25	1000	27	1180	28.5
1010	1735	3470	95	438	21.2	600	28.5	675	31.5	750	33	900	39.5	1010	43
1585	2390	3020	63	317	4	633	13.5	950	21	1266	27	1585	32.5		
2200	2760	3010	61	367	14	733	30	1100	43	1466	51.7	1833	57.5	2200	61.5
	2220	2670	93	917	7.5	1167	9.5								
	1900	2850	119	867	22	933	23.8								
	2020	2760	89	912	14.5	1000	16.5								
	1950	2670	89	947	31.5	983	33.8								
1377			91	1030	24.7	1160	26.4	1377	28.5						
			92	768	24										
	1625	1900	85	770	34.4										
1125			95	633	13.9	842	16.3	1000	19.6	1125	20.7				
			95												
1150	1800	2740	122	460	10.9	888	23.2	1150	29.2						

c Marine engine. *f* Racing marine engine; connecting rod length 13in., specially low *m*. *g* Reciprocating parts unduly heavy.

No. of Engine	Bore in Inches $= d$.	Stroke in Inches $= s$.	Stroke-bore Ratio, $r = s/d$.	No of Cylinders. N.	Volume Ratio of Compression.	Inlet and Exhaust Valves. Diameter and Lift of Inlets. δin. × λin.	Diameter and Lift of Exhausts. δin. × λin.	Mean Velocity through Inlets at Maximum H.P. $= v$, ft. p.m.	Value of δ given by $.29d\sqrt[4]{d^2s/m}$ inches.	Mass of Reciprocating Parts in lb. $= m$	$d^2s =$	$\sqrt{d^2s/m} =$	ηp at Maximu B.H.P. lb. per sq. in.
95	3.75	4.5	1.20	2	5.0	1.50 × .31	1.50 × .31	1875	2.22	3.63	63.2	4.17	77.8
96	3.75	4.5	1.20	4	5.0	1.50 × .31	1.50 × .31	1875	2.18	3. 8	63.2	4.03	74.3
97	3.38	4.25	1.26	4	5.0	1.38 × .31	1.38 × .31	1700	1.91	3.38	48.6	3.79	72.2
98	3.35	5.0	1.49	4	4.0	1.58 × .35	1.58 × .35		1.94	3.5	56.1	4.0	
99	3.35	5.0	1.49	6	4.0	1.58 × .35	1.58 × .35		1.94	3.5	56.1	4.0	
100	4.5	4.5	1.0	6	4.0	2.0 × .35	2.0 × .35	1800	2.49	6.88	91.1	3.64	61.3
101	3.15	3.54	1.12	1	3.3	1.16 × .276	1.16 × .276	2210			35.1		70.6
102	3.15	3.54	1.12	2	3.3	1.16 × .276	1.16 × .276	2210			35.1		70.8
103	3.15	3.54	1.12	3	.3	1.16 × .276	1.16 × .276	2210			35.1		
104	3.15	3.54	1.12	4	3.3	1.16 × .276	1.16 × .276	2210			35.1		66.0
105	3.15	3.54	1.12	6	3.3	1.16 × .276	1.16 .276	2210			35.1		
106	3.35	4.33	1.29	4	3.5	1.58 × .276	1.58 × .276	1770			48.6		73.2
107	3.94	4.33	1.10	4	3.5	1.58 × .4 3	1.58 × .433	2180			67.2		71.3
108	3.94	5.12	1.30	4	3.7	1.85 × .433	1.58 × .433	2520			79.4		82.2
109	4.33	5.12	1.18	4	4.0	1.61 × .433	1.61 × .433	2490			96.0		75.5
110	4.73	5.12	1.08	4	4.0	1.61 .433	1.61 × .433	2970			114.5		76.2
111	5.0	5.12	1.02	4	4.0	2.36 .433	2.36 .433				128		
112	3.15	3.94	1.25	4	4.25	1.18 × .303	1.18 × .303	2340	1.56	4.63	39.1	2.91	67.8
113	3.54	4.73	1.34	4	4.25	1.50 × .315	1.50 × .315	>1760	1.78	6.62	59.3	3.0	66.4
114	4.33	5.12	1.18	4	4.25	1.89 × .394	1.89 × .394	>1790	2.24	9.47	96	3.19	62.3
115	5.12	5.52	1.08	4	4.25	2.36 × .473	2.36 × .473	>1520	2.70	13.2	144.4	3.31	67.4
116	4.88	5.12	1.05	4	4.8						122		90.3
117	3.78	5.12	1.35	4	5.0						73.2		
118	3.15	5.12	1.63	4	4.9						50.7		
119	4.0	5.0	1.25	4	4.0	2.25 × .438	2.25 × .438		2.16	6.72	80.0	3.45	
120	5.0	5.25	1.05	4	5.1	2.0 × .313	2.0 × .313	>1920	2.75	10.5	131	3.59	64.5
121	4.25	5.25	1.24	4	5.1	1.75 × .313	1.75 × .313	>1680	2.22	9.12	94.8	3.23	73.8
122	3.15	4.5	1.43	4	5.25	1.438 × .25	1.438 × .25	>1350	1.58	5.0	44.7	2.99	90.5
123	3.15	4.73	1.50	2	4.25	1.575 × .354	1.575 × .354	>1260	1.74	3.53	46.9	3.65	74.2
124	3.15	4.73	1.50	4	4.25	1.575 × .354	1.575 × .354	>1260	1.72	3.75	46.9	3.54	67.2
125	3.54	5.12	1.45	4	4.0	1.89 × .354	1.89 × .354	>1200	1.85	6.06	64.2	3.26	73.8
126	3.54	5.12	1.45	6	4.0	1.89 × .354	1.89 × .354	>1200	1.85	6.06	64.2	3.26	67.3
127	3.94	5.12	1.30	4	4.0	1.89 × .354	1.89 × .354	>1300	2.04	7.72	79.4	3.21	72.2
128	3.94	5.12	1.30	6	4.0	1.89 × .354	1.89 × .354	>1300	2.04	7.72	79.4	3.21	67.8
129	4.33	5.52	1.28	4	3.9	2.36 × .394	2.36 × .394	>1185	2.30	9.25	103.5	3.35	67.9
130	4.92	5.92	1.20	4	3.8	2.56 × .433	2.56 × .433	>1090	2.55	13.8	143	3.21	67.7
131	4.5	4.75	1.06	6	4.2	1.75 × .375	1.75 × .438	2150	2.37	8.75	96.2	3.32	59.2
132	4.25	4.5	1.06	4		1.375 × .375	1.375 × .375				81.2		
133	5.0	5.0	1.00	1							125		
134	3.94	4.73	1.20	1	4.5	1.42 × .26	1.42 × .26	2500	2.29	5.3	73.5	3.72	77.6
135	3.94	5.12	1.30	1	4.2	1.42 × .26	1.42 × .26	>3280	2.22	5.52	79.4	3.8	63.7
136*h*	3.94	6.30	1.60	1	4.7	1.10 × .197	1.10 × .197	>6720 ?	2.55	3.91	97.8	5.0 ?	82.7
137	2.60	3.94	1.52	4	4.5	0.985 × .185	0.985 × .185	>2200	1.26	3.74	26.6	2.67	59.7
138	2.95	4.73	1.60	4	4.2	1.10 × .264	1.10 × .264	2840	1.49	4.46	41.2	3.04	55.2
139	3.54	4.73	1.34	4	4.2	1.26 × .225	1.26 × .225	2560	1.76	6.95	59.3	2.93	67.2
140	3.94	5.12	1.30	4	4.2	1.65 × .225	1.65 × .225	2000	1.99	8.6	79.5	3.04	60.0
141	4.73	5.12	1.08	4	4.2	1.65 × .276	1.65 × .276	>2810	2.53	9.87	114.5	3.41	66.2
142	3.54	4.73	1.34	8	4.2	1.26 × .315	1.26 × .315	>2500	1.71	7.72	59.3	2.77	66.4
143	5.92	7.08	1.20	4	4.2	2.13 × .394	2.13 × .394	2740	2.99	30.4	248	2.86	63.8
144	4.0	4.38	1.09	4	4.0	1.38 × .18	1.38 × .18	>1840	2.05	7.0	70.1	3.16	68.0

h **Reciprocating parts abnormally light and valve**

TABLE 1.—Figures for Four Cylinders Only.

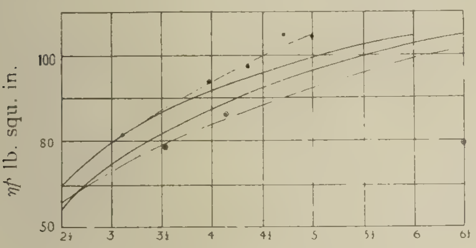
No.	$d =$	Compression Ratio.	$\sigma =$	H.P. –	$\eta p =$	No.	$d =$	Compression Ratio.	$\sigma =$	H.P.	$\eta p =$
52	2.44	4.1	1155	10	61.2	85	4.13	4.17	1167	38	80.2
53	2.56	4.5	1155	11.5	63.8	61	4.25	5	833	30	83.8
137	2.60	4.5	1050	12	71.2	80	4.25	4.15	1100	37.4	79.2
46	2.95	4.5	950	13	66.1	15	4.25	3.52	960	32	77.7
54	2.95	4.5	947	13	66.3	92	4.25	4.24	1000	39.2	91.2
138	2.95	4.2	947	15.6	79.6	121	4.25	5.1	1050	35	77.6
74	3.13	4.13	900	15.3	73	132	4.25		900	27.9	72.2
83	3.13	3.1	950	21	94.8	35	4.33	4.2	1110	43	86.5
33	3.15	4	937	17.3	78.3	109	4.33	4	984	49.8	92.8
62	3.15	4	1105	23.3	89.5	39	4.5	3.5	1000	34.8	72.2
104	3.15	3.3	1082	19.7	77.2	43	4.5	3.78	1000	46.7	96.8
122	3.15	5.25	900	21.5	101.2	71	4.5	4.13	1000	43.5	90.2
86	3.35	5	933	23.8	95 5	56	4.53	3.95	853	35	84
106	3.35	3.5	1082	25	86.7	100	4.5	4	1050	37.5	74.2
97	3.38	5	995	19.2	71	131	4.5	4.2	990	35.3	73.9
27	3.38	4.8	1067	27.8	95.8	4	4.73	4.73	1082	45	78.2
2	3.5	–	937	21	76.8	12	4.73	–	1024	43	78.8
64	3.5	4.76	900	20	76	14	4.73	–	1165	53.3	86.1
89	3.5	4.09	1030	24.7	82.4	36	4.73	4	1075	40.3	70.3
29	3.53	4.8	948	25	89	110	4.73	4	984	50.6	96.5
50	3.54	4.77	946	21	74.5	141	4.73	4.2	1023	42.5	78
57	3.54	4.43	1025	23.8	77.9	66	4.75	4.73	1000	46	85.7
113	3.54	4.25	948	20	70.8	116	4.88	4.8	1025	58	99.8
125	3.54	4	1024	25.5	83.7	3	4.92	–	1000	50.7	88.1
139	3.54	4.2	947	23.4	83	130	4.92	3.8	987	44	77.5
70	3.56	5.1	1024	26	84.3	72	5	3.96	1000	57	95.8
75	3.56	4.15	950	23	80.3	79	5	4.15	1025	49.3	80.9
107	3.94	3.5	1082	33.7	84.3	91	5	3.85	770	34.4	75.2
108	3.94	3.7	1108	37.4	91.5	111	5	4	984	59.5	101.6
127	3.94	4	1024	30	79.3	120	5	5.1	1050	45	72
135	3.94	4.2	1023	32.8	86.8						
10	4	–	908	27.7	80						
60	4	5	833	27.5	86.7						
81	4	4.22	1000	27	70.8						
84	4	3	1100	43	102.6						
88	4	4	983	33.8	90.3						
55	4.13	4.84	1025	35.2	84.5						

TABLE II.

Values of m., etc., from equation (1). Cast-iron pistons.

$r =$	d. in inches.	$d^2s = d^3r =$	m. in lbs.	$d^2s/m =$	$\sqrt{d^2s/m} =$
.75	2.5	11.72	2.89	4.06	2.02
	3	20.3	3.9	5.21	2.28
	3.5	32.2	5.3	6.08	2.47
	4	48	7.2	6.67	2.58
	4.5	68.3	9.6	7.12	2.67
	5	93.8	12.6	7.44	2.73
	5.5	124.8	16.3	7.67	2.77
	6	162	20.7	7.83	2.80
1	2.5	15.63	2.94	5.32	2.31
	3	27	4	6.75	2.60
	3.5	42.9	5.4	7.94	2.82
	4	64	7.4	8.65	2.94
	4.5	91.1	9.9	9.20	3.04
	5	125	13	9.63	3.10
	5.5	166.4	16.8	9.92	3.15
	6	216	21.4	10.1	3.18
1.25	2.5	19.5	2.98	6.55	2.56
	3	33.8	4.1	8.24	2.87
	3.5	53.6	5.6	9.57	3.09
	4	80	7.6	10.52	3.24
	4.5	113.9	10.2	11.16	3.34
	5	156.3	13.4	11.67	3.42
	5.5	208	17.3	12.03	3.47
	6	270	22	12.27	3.50
1.50	2.5	23.4	3.03	7.73	2.78
	3	40.5	4.14	9.73	3.13
	3.5	64.3	5.7	11.30	3.36
	4	96	7.8	12.31	3.51
	4.5	136.7	10.4	13.14	3.62
	5	187.5	13.8	13.58	3.68
	5.5	249.6	7.8	14.03	3.74
	6	324	22.7	14.28	3.78

rise as high as 95½ lb. Expressed in percentages, approximately 5.6 per cent. of the 124 engines give values above 90 lb.; 15.3 per cent. between 80 and 90 lb.; 28.2 per cent. between 70 and 80 lb.; and 21.2 per cent. between 65 and 70 lb. About 50 per cent. of the engines thus show mean pressures above 70 lb. The highest pressure was obtained in a cylinder of 3.35 inches diameter. Fig. III. supplies no evidence of increase of mean pressure with bore. Some makers obviously succeed in getting very high mean pressures from quite small cylinders and others do not. It is interesting now to consider the highest values of ηp given in these tests by cylinders of different diameters. Compare the two highest obtained at the highest brake horse-power recorded with cylinders of about 3 in., 4 in. and 5 in. respectively as follows:—

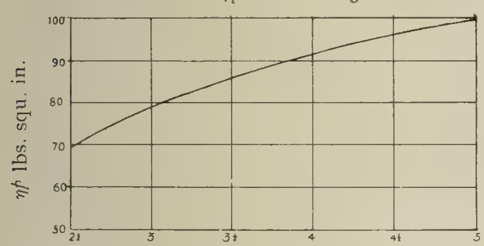


Cylinder Bore Inches.

Fig. IV.

Test No.	$d=$	ηp
122..	3·15 in. ..	90·5 lb. per sq. in.
83..	3·13 in. ..	88· ,, ,,
		Mean 89·2 lb. per sq. in.
45..	4·0 in. ..	92·3
88..	4·0 in. ..	90·2
		Mean 91·2 lb. per sq. in.
72..	5·0 in. ..	94·6
116..	4·88 in. ..	90 3
		Mean 92·4 lb. per sq. in.

Here an increase is shown of nearly 4 per cent. The value of ηp for about 3 in. diameter is 89 lb. per sq. in., and for 5 in. diameter 92.5 lb. per sq. in.

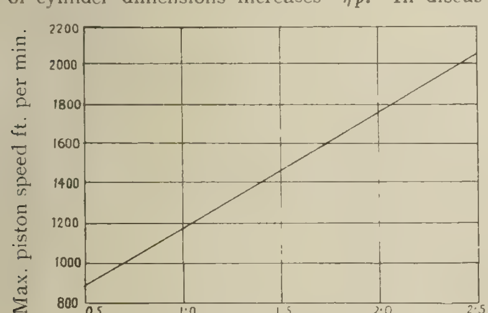


Cylinder Bore Inches.

Fig. V.

Tests by Messrs. White and Poppe cited by Mr. Burls show a large increase—over 25 per cent.; 3.15 in. diameter giving $\eta p=80.6$ lb. per sq. in., and 5 in. diameter $\eta p=102$ lb. per sq. in. See Fig. IV.

The experience of many makers also proves that in testing under similar conditions increase of cylinder dimensions increases ηp. In discussion many members of the committee give this increase as part of their ordinary experience. It is known also that in closed cylinder experiments the cooling loss in a given time is greater in a small vessel than in a large one; further, that in gas engine practice higher mean pressures are obtained under similar conditions of compression in large cylinders than in small; also that the mechanical efficiency of a large engine is usually greater than that of a small one. Taking into consideration all these matters and the results of many tests of petrol engines for cars, the committee is of opinion that it is necessary to allow for an increase of the value of ηp with the cylinder diameter, and they accept 68½ lb. per sq. in. for ηp when $d=2\frac{1}{2}$ in. and 99½ when $d=5$ in. The best current practice as to ηp is therefore taken as—



Cylinder Bore Inches.

Fig. VI.

$$\eta p = 130\ (1 - 1{\cdot}18/d) \quad \text{lb. sq. in.} \quad \ldots \quad \text{(a)}$$

After consideration of the evidence contained in the Data of 143 Petrol Engines collected by Mr. G. A. Burls, M.Inst.C.E., the Committee is of opinion that the rate of increase of maximum practicable piston speed with stroke-bore ratio can be adequately represented by the equation:—

$$\sigma = 600\ (r+1) \quad \text{ft. per min.} \quad \ldots \quad \text{(b)}$$

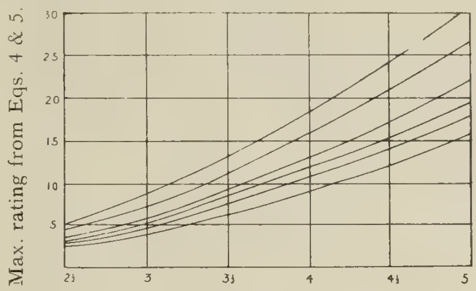
where r is the ratio of stroke to bore. This implies a maximum practicable piston speed of 1,200 ft. per min. for $r=1$, rising to 2,100 ft. per min. for $r=2.5$. See Fig. VI.

Proposed Formula.

The committee therefore propose a formula which includes an increase of ηp with cylinder diameter and increase of piston speed with stroke-bore ratio. It is:—

Max. b.h.p. rating per cylinder $= 0.464\ (d+s)\ (d - 1.18)$.

This formula may be considered to give the maximum practicable b.h.p. as determined by a bench test under onerous, but still safe, conditions for carefully designed and soundly constructed engines of from 2½ in. to 5 in. cylinder diameter and stroke-bore ratio up to 2.5.



Cylinder Bore Inches.

Fig. VII.

The exact formula for horse-power in terms of bore, d, and piston speed, σ is:—

$$B.H.P.\ per\ cyl. = 1/168000 \cdot d^2 \cdot \eta p \cdot \sigma \ \ldots$$

where d is in inches; ηp in lb. per sq. in.; and σ in ft. per minute.

In this equation substitute for ηp and σ the expressions (a) and (b), and the equation then becomes:—

$$\text{B.H.P. per cyl.} = 0{\cdot}464\ d\ (d - 1{\cdot}18)\ (r+1).$$

For purposes of computation the following equivalent form is better:—

$$\text{B.H.P. per cyl.} = 0{\cdot}464\ (d+s)\ (d - 1.18).$$

To simplify calculations, we recommend that the constant in the formula be taken as 0.45 instead of 0.464, that is the b.h.p. for an engine with N cylinders $= 0.45\ (d+s)\ (d - 1.18)$ N where d and s are in inches. The form of the formula if d and s are in mm. will be added later.

TABLE

Cf proposed Ratings by formula.

Max. b.h.p. rating per cylinder $= 0{\cdot}45\ (d+s)\ (d - 1{\cdot}18)$.

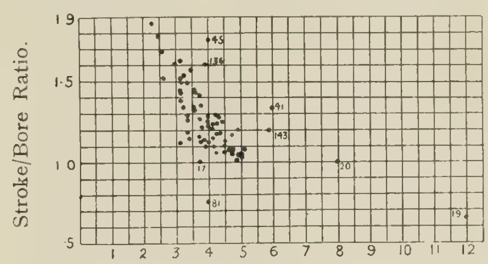
$r=$	$d=$ in.	$s=$ in.	R't'ng per Cyl.	$r=$	$d=$ in.	$s=$ in.	R't'ng per Cyl.
	2½	1⅞	2·6		2½	3¾	3·7
0.75	3	2¼	4 3		3	4½	6·1
	4	3	8·9	1·5	3½	5¼	9·1
	5	3¾	15·0		4	6	12·7
					4½	6¾	16·8
	2½	2½	3·0		5	7½	21·4
	3	3	4 9				
1·0	3½	3½	7·3		2½	5	4·5
	4	4	10·2	2·0	3	6	7·4
	4½	4½	13·5		4	8	15·2
	5	5	17·2		5	10	25·8
	2½	3⅛	3·3		2½	6¼	5·2
	3	3¾	5 5	2.5	3	7½	8·6
1·25	3½	4⅜	8·2		4	10	17·8
	4	5	11·4		5	12½	29·6
	4½	5⅝	15·1				
	5	6¼	19·3				

A Proposal for a Maximum Power Rating Formula for the Petrol Engines of Automobiles.

THE following paper, by G. A. Burls, Wh.Ex., M.Inst.C.E., is supplementary to the report just given, and the following symbols are employed throughout:—

- d = bore of cylinder in inches.
- s = stroke of piston in inches.
- r = s/d = the stroke-bore ratio.
- N = number of cylinders.
- n = number of revolutions per minute.
- σ = piston speed in ft. per minute.
- η = mechanical efficiency of engine.
- p' = "acceleration-pressure" in lb. per sq. in. on piston.
- p = mean effective pressure on piston, in lb. per sq. in. during the working stroke.
- m = mass of piston and connecting rod, all complete, in lb.
- ρ = crank length in ft.
- l = connecting rod length, between centres, in ft.
- v = mean velocity of gas through inlets, in ft. per minute.
- δ = diameter of inlet valve in inches.
- λ = lift of inlet valve in inches.
- b.h.p. = brake horse power.
- C = volume ratio of compression.
- P = compression-pressure in lb. per sq. in. absolute.

In the accompanying diagram (Fig. I.), values of the stroke-bore ratio are plotted against bore. No regular variation was, of course, to be anticipated, but the diagram indicates a steady general diminution in r with increase of d. Broadly, in the cases included in Fig I., r varies from 1.75 for $d=2\frac{1}{2}$ in., to 1.0 when $d=6$ in. Mr. Poppe holds the opinion that for large bores, from considerations of gearing strength, and comfort of occupants of the car, a value of r not differing much from unity is best; for small bores r may be 1½ or even more without much disadvantage; during the past two or three years the value of r has shown a tendency to increase. Few examples exist at present wherein the ratio has as high a value as 2; practice tending towards the use of



Cylinder Bore Inches.

Fig. I.

TABLE

illustrating roughly the variation of r with d in car engines of 1911.

Description.	Nominal h.p.	Bore m.m.	Stroke m.m.	Stroke / Bore
Small car engines:				
Calthorpe	15.0	75	150	2·0
Le Gui.....	10.0	65	130	2·0
Le Gui.........	15.0	75	150	2·0
Jackson	6.2	100	200	2·0
Delage.........	15.7	65	125	1·92
Opel............	14.0	64	120	1·88
Delage.........	9.5	62	110	1·78
Medium car engines:				
Sunbeam........	18-22	80	120	1·5
Talbot..........	20	80	120	1·5
Sheffield Simplex:	25	85	127	1·49
Napier..........	30	88	127	1·55
Deasy	18-24	90	130	1·45
Crossley	20	102	140	1·37
Large car engines:				
Dennis..........	40	126	130	1·03
Lanchester.....	38	102	102	1·0
Maudslay	35-45	127	127	1 0
Napier..........	65	127	127	1 0
Talbot...... ..	35	127	120	0.945
Imperia	56-60	150	140	0·933
Napier..........	90	156	127	0·814

as large a ratio as can be adopted without undue sacrifice of smoothness in running. For the car engines of 1911 the range appears to be from $r=2.0$ for $2\frac{1}{2}$ in. bore, to $r=1.0$, or even rather less for 6 in. bore; the short table at the foot of the last column illustrates generally the state of current practice.

The lowest value of r is found in the 20 h.p. Lanchester, with a bore of 4 in., and a stroke of only 3 in.; thus r is here only 0.75; in this respect the Lanchester engine is unique. A stroke of 3 in. also appears to be the shortest used in car engines; the longest, excluding racing engines, is about $7\frac{3}{4}$ in., though it is rare to find an engine with a stroke exceeding 7 in. Pistons are usually of medium, hard, close-grained cast iron, usually with three, but occasionally with four cast iron spring rings; an additional oil-excluding ring is sometimes fitted near the bottom of the piston. In normal car engines the piston is generally rather greater in length than the cylinder bore, the proportion averaging about 1.16 to 1. Pressed steel pistons machined all over are now being increasingly used, as with this material from 15 to 20 per cent. of weight can be saved.

Connecting-rods are of stamped steel, of I section; the ratio of length of rod to stroke of piston varies but little at present from the value 9/4.

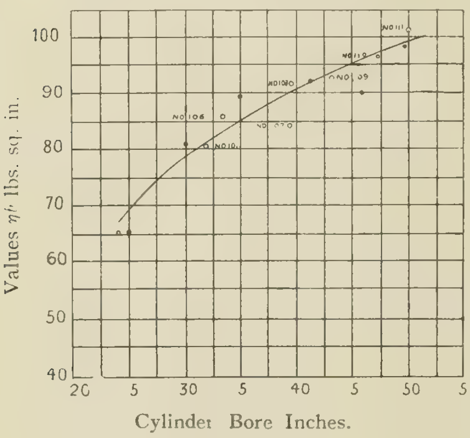


FIG. II.—Variation of ηp with Cylinder Bore.

In the S. M. M. T. report on horse-power formulæ, no actual values of m are given, but it is assumed from rational considerations alone that m can be expressed in the form :—

$$m=ad^2\,(d+bs)$$
$$m=ad^3\,(1+br),$$

where a and b are constants.

In the tables of engine data we have now the advantage of possessing a large collection of actual values of m over a range of engine sizes from $d=2.44$ in. to $d=12$ in.

The author has found it necessary to add a constant term to the above formula in order that it may adequately resume practice, and suggests the following expression :—

$$m=0.08\,d^3\,(1+0.15r)+1.5\ \text{lb.} \quad \ldots \quad (1)$$

for cast iron pistons. For the very light pressed steel pistons now being used, the few cases collected suggest the expression

$$m=0.05\,d^3\,(1+0.15r)+1.5\ \text{lb.} \quad \ldots \quad (2)$$

In the accompanying table II., values of m and of d^2s/m and $\sqrt{d^2s/m}$ are shewn, calculated from equation (1) for the whole range of present practice; and in Fig. III., the graphs of (1) and (2) are exhibited for several values of r together with plottings of actual values of m taken from the tables of engine data.

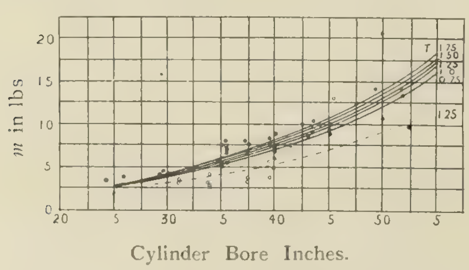


Fig. III.—Weight of reciprocating parts in terms of bore and ratio r.

In Tables III. and IIIA., actual and calculated values of m may be compared; it will be noted that equation (1) fairly resumes the facts from $d=2\frac{1}{2}$ in. to $d=12$ in.

In Fig. IV. values of d^2s/m from Table II. are plotted against d^2s for values of r, from 0.75 to 2.5 inclusive; the curves are hyperbolæ having horizontal asymptotes as indicated in the diagram. The asymptotes are rapidly approached, but unfortunately d^2s/m does not become approximately constant in value until d^2s exceeds 125, which is somewhat outside the range of ordinary sizes. Much attention is now being given to the reduction of the mass of the reciprocating parts, not only in order to lessen vibration arising from want of complete balance, but also to enable higher piston speeds to be normally used, and thus diminish the weight of the engine per horse power.

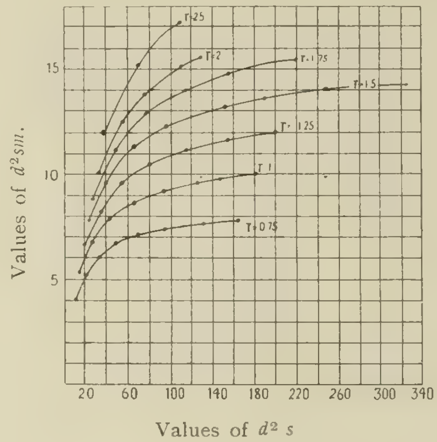


FIG. IV.—Values of d^2s/m for $r=0.75$ to 2.5; C.I. Pistons.
See Table II.

At the very high piston speeds now frequent in car engines, the inertia of the piston and connecting rod exercises an important influence upon the distribution of the driving effort at the crank pin during the working stroke. In the accompanying Fig. V., OP represents the crank, PM the connecting-rod of an engine; the piston stroke is AB; DQCE is the curve of piston acceleration obtained by any of the well-known graphical methods, for example, that of Prof. Klein. In the position shewn, MQ measures the acceleration of the piston to the scale for which OP measures the velocity of the crank pin.

The values of the acceleration when the piston is respectively at the top and bottom of its stroke are :—

at top $\quad AD=\omega^2\rho\,(1+\rho/l)$ ft. per sec. per sec. (3)
at bottom $BE=\omega^2\rho\,(1-\rho/l)$ ft. per sec. per sec. (4)

The force necessary to produce this acceleration is obtained by multiplying by the reciprocating mass. As the connecting-rod partly reciprocates and partly rotates only a fraction of its mass should be reckoned as reciprocating, the remainder being regarded as revolving with the crank pin.

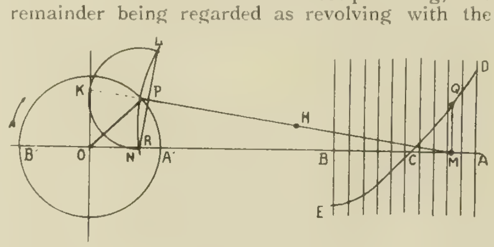


Fig. V.

OP$=\rho$ ft. PM$=l$ ft.
$l/\rho=9/2$
ω=angular velocity of P in radians per sec.

Herr Güldner considers that from 0.45 to 0.55 of the connecting-rod mass should be regarded as reciprocating; for car engines it will be sufficiently accurate to take the fraction roundly at the value 0.5. Denoting then by μ the mass, in lb., of the part regarded as reciprocating, we may in usual cases take $\mu=0.75\ m$ (*vide* equation (1), *ante*). Thus, if F. denote the force in lb. weight, necessary to produce the accelerations at the top and bottom of the stroke, we have

at top $\quad F_A=\mu\omega^2\rho/g\,(1+\rho/l)$ lb. (5)
at bottom $\quad F_B=\mu\omega^2\rho/g\,(1-\rho/l)$ lb. (6)

Now d denoting the piston diameter in inches, if p^1 be the pressure upon it in lb. per sq. in. which is necessary to produce the gross force F, then $F=\pi/4.d^2p^1$, so that $p^1=4\ F/\pi\ d^2$; thus from (5) we have, for the top end :—

$$p^1=4\mu\omega^2\rho/\pi g d^2\ (1+\rho/l)\ \text{lb. per sq. in.} \quad (7)$$

For practical calculation this result may be simplified; for if n be the number of revs. per min., $\omega=2\pi\ n/60$, while s being the stroke in inches, $\rho=s/24$. Also $\mu=0.75\ m$. and the ratio ρ/l departs but little from the value 2/9. Thus, on substitution and reduction, equation (7) becomes :—

$$p^1=.0000166\ .\ mn^2s/d^2\ \text{lb. per sq. in.} \quad (8),$$

in which d and s are in inches, n in revs. per min., and m in lb. It is of interest to note the large values attained by p^1 in some cases owing to the very high revolution speed, notwithstanding the small mass of the reciprocating parts. In the following short table some results from actual engines are given :—

Reference letter.	Bore. d. in.	Stroke s. in.	Revs per min. n.	Piston speed. ft. p. m	Mass. m. lb.	Value of p^1. lb. per sq. in.
A.	4 0	7.0	1560	1820	5.5	97
B.	2.44	4.33	2000	1443	3.2	154
C.	4.75	5.0	2000	1670	12.22	179
D.	3.53	4.75	2400	1900	5.38	195
E.	3.74	5.32	2300	2040	7.01	234
F.	3.15	5.12	2900	2475	3.7	266

Referring now to Fig. VI. Suppose the engine to be running and that the piston is at A, commencing the downward suction stroke. The pressure, in lb. per sq. in. of piston area, required to produce the necessary acceleration of the reciprocating parts is measured by AD, in this case about 266 lb. per sq. in. As, however, there is no pressure on the piston during this stroke, the necessary force must be supplied from the crank pin; hence from A to C the crank pin must drive the reciprocating mass. At C this mass has its maximum velocity, and is thereafter retarded until it comes momentarily to rest at B; thus from C to B the reciprocating mass drives the crank pin, restoring during this period the work given to it during the first period AC; hence at C there is a reversal of the pressure between the crank pin on the connecting-rod end. Similarly during the compression and exhaust strokes a reversal of pressure occurs at or close to C.

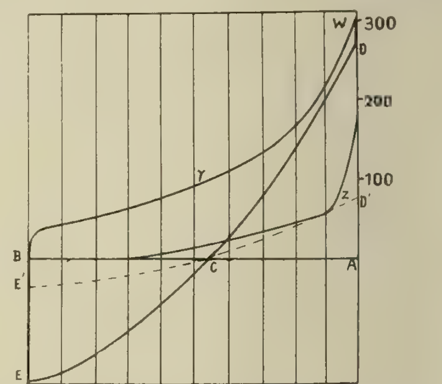


FIG. VI.—Effective piston pressures for Engine F. Revolutions, 2,900 per min.

In the working stroke ignition occurs, and at A a pressure represented by AW is suddenly created on the piston; a reversal of pressure immediately takes place, the reciprocating mass driving the crank pin, which condition continues in normal cases throughout the working stroke. The pressure available for producing driving effort at the crank pin is represented at any point of the stroke by the vertical height, at that point, of the diagram DCEBYW.

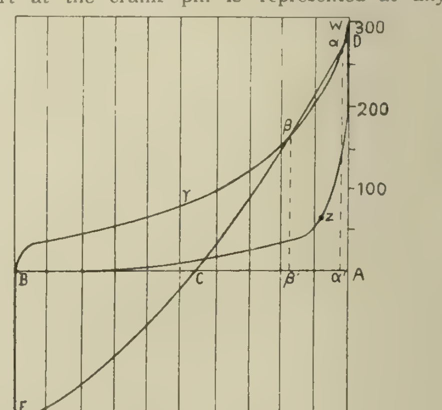


FIG. VII.
Effective piston pressures for Engine F. Revolutions 3,000 per min.

The highest speed of which the writer has good evidence at present, is that of engine F, which has on several occasions maintained about 2,900 revs. per min. in a car for twenty consecutive

minutes. Fig. VI. is drawn to scale for this case, using an actual indicator diagram taken at a somewhat lower speed by a mirror and diaphragm indicator. Here it will be noted that at the commencement of the downward working stroke, with the explosion pressure assumed, nearly all the pressure on the piston is absorbed in accelerating the reciprocating mass, only the small amount represented by DW being available for causing driving effort at the crank pin.

The resultant pressure increases during the working stroke as shown by the depth of the area as already pointed out, and thus the effect of inertia is to transfer the maximum pressure available for causing driving effort to the later portion of the stroke. The height of the acceleration curve DCE increases in proportion to the *square* of the number of revs. per minute; thus, at about 1,500 revs. in the case under consideration it sinks to the position indicated by the dotted line D^1CE^1, and the inertia effect then tends to equalise the driving pressure throughout the stroke. On the other hand, if the engine speed were increased to about 3,000 revs. per minute the acceleration pressure curve would cut the expansion line of the indicator diagram at the points α and β, as shewn in Fig. VII. Such cases appear to be very rare, but the condition is undoubtedly closely approached in many racing engines.

When the two curves cut there are *three* reversals of pressure at the crank pin during the working stroke, viz., at D, α and β. D α W and βCEBY are positive work-areas, while that between α and β is negative. This triple reversal would tend to set up objectionable vibration, and this consideration, coupled with the fact that this critical speed is occasionally approached but apparently not exceeded by certain racing engines, has led the author to propose that for practical maximum power rating purposes the highest speed of an engine shall be taken to be this critical speed. As will be seen shortly, this assumption furnishes reasonable values for the maximum speed of an engine, and also enables a rating formula in d and s to be constructed by which the performance of special racing engines may be satisfactorily approximated to for handicapping purposes.

The maximum explosion pressure attained in normal car engines appears to range from about 290 to 320 lb. per sq. in. above atmosphere, and this can be realized at speeds of up to 2,000 r.p.m. It is doubtful whether so great an initial pressure is obtained, however, at the high "critical speed," owing to the effects of wire-drawing and imperfect scavenging. Accordingly, for the present purpose 275 lb. per sq. in. above atmosphere is taken as roundly the maximum explosion pressure ordinarily realizable at the limit. If p^1 be taken at about 0.95 of this, the acceleration-pressure curve will in general cut the expansion line in two points in the manner indicated in Fig. VII., and the triple reversal of

TABLE

of maximum piston speed and revolutions from eqs. (9) and (10).

Cast Iron Pistons.

r	d in in.	σ max ft.p m.	n max. r.p.m.
0·75	2.5	1330	4260
	3·0	1510	4030
	3·5	1630	3730
	4·0	1700	3400
	4·5	1760	3140
	5 0	1800	2880
	5·5	1825	2650
	6·0	1840	2450
1·0	2·5	1525	3660
	3·0	1710	3430
	3·5	1860	3190
	4·0	1940	2910
	4·5	2000	2680
	5·0	2040	2450
	5·5	2080	2270
	6·0	2100	2100
1·25	2·5	1690	3250
	3·0	1890	3020
	3·5	2040	2800
	4·0	2130	2550
	4·5	2200	2360
	5·0	2260	2170
	5·5	2280	2000
	6·0	2300	1850

r	d in in.	σ max. ft.p.m.	n max. r.p.m.
1·5	2·5	1830	2930
	3·0	2060	2750
	3·5	2210	2530
	4·0	2310	2310
	4·5	2380	2110
	5·0	2430	1940
1·75	2·5	1960	2700
	3 0	2200	2520
	3·5	2370	2330
	4·0	2460	2110
	4·5	2540	1940
2·0	2·5	2080	2500
	3·0	2330	2330
	3·5	2490	2140
	4·0	2600	1950
2·5	2·5	2300	2210
	3·0	2560	2050
	3·5	2740	1890
	4·0	2850	1710

pressure at the crank pin will then occur. Hence the maximum value of p^1 for practical purposes is here taken as $275 \times 0.95 = 260$ lb. per sq. in. above atmosphere.

Substitute now, the value 260 for p^1 in equation (8), and we obtain on reduction, for the maximum revolution speed:—

$$n_{\text{max.}} = 3957 \sqrt{d^2/ms} \text{ r.p.m.} \quad . \quad . \quad . \quad . \qquad (9)$$

Also, if σ denote piston speed in feet per minute, $\sigma = ns/6$, so that (9) in terms of σ is:—

$$\sigma_{\text{max.}} = 660 \sqrt{d^2s/m} \text{ ft. per min.} \quad . \quad . \quad . \quad . \qquad (10)$$

for the maximum piston speed. It is to be noted that if the expression given in equation (1) is substituted for m in this equation, the maximum piston speed is given in terms of d and r only, viz.:—

$$\sigma_{\text{max.}} = \frac{660}{\sqrt{0.012 + 0.08/r + 1.5/d^3r}} \quad . \quad . \quad . \quad . \qquad (11)$$

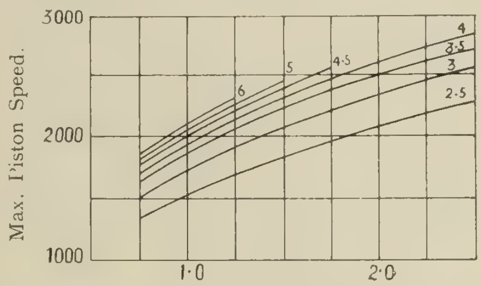
and thus the maximum practicable piston speed increases both with r and with d absolutely. In the table in the last column values of $n_{\text{max.}}$ and $\sigma_{\text{max.}}$ from (9) and (10) are exhibited for the range of sizes at present employed in petrol engines for car work, and it will be seen that the figures obtained are not unreasonably high, viewed in the light of recent racing performances. The increase of maximum piston speed with d and r is also shown by the curves in Fig. VIII., which have been plotted from this table.

Exact Formula for Horse-power.

For an internal combustion engine working on the Otto cycle, the brake horse-power is exactly given by the formula:—

$$\text{b.h.p. per cyl.} = 1/33{,}000 \times 24.\pi/4.d^2\eta p.ns. \quad . \quad . \quad . \qquad (12)$$

where d and s are the bore and stroke respectively in inches; η is the mechanical efficiency of the engine; p is the m.e.p. in lb. per sq. in. during the working stroke, and n is the number of revs. per minute.



Stroke-Bore ratio r.

FIG. VIII.

If σ denote the piston speed in feet per minute, then equation (12) becomes:—

$$\text{b.h.p. per cyl.} = 1/168000.d^2\eta p.\sigma \quad . \quad . \quad . \quad . \qquad (13)$$

Now, it is a matter of considerable difficulty and delicacy to obtain accurate indicator diagrams from a fast-running petrol engine, and hence p cannot usually be directly determined; η can be ascertained without much difficulty, but in general the value of the product ηp termed the "mean pressure corresponding to the b.h.p." is found from bench tests of the power by aid of equation (12). We have already obtained a formula for the maximum practicable piston speed (equations 10 and 11), and if the variation of ηp with size can be similarly expressed in terms of d and s it is apparent from equation (13) that the max. b.h.p. per cylinder will be given in terms of the stroke and bore.

For combustion chambers of similar form the surface to which the hot gases are exposed increases as d^2, while the cubic contents of the chambers increase in proportion to d^3, and hence the proportion of surface to volume varies as $1/d$, and accordingly diminishes as the bore increases. Thus, owing to smaller loss by cooling, higher mean pressures may reasonably be looked for in a large cylinder than in a small one, other things being equal. Higher compressions and richer mixtures were until recently used in conjunction with small bores, but of late increased experience and skill in designing have enabled builders to successfully use compressions as high, and mixtures as rich, in large engines as in the smaller sizes. Practice is very irregular in respect of the compression ratio as the accompanying short table for 1910 cars shows. The average ratio for the cases included is 4.3, and it will be seen that there is no observable tendency to any diminution in value with increase in bore from $2\frac{1}{2}$ in. to 6 in.

If C denote the compression ratio, and P the compression pressure in lb. per sq. in. absolute, then for practical purposes:—

$$P = 14.7.\ C^{\frac{4}{3}} \qquad (14)$$

The maximum compression pressure that can be successfully used in petrol car engines at present appears to be about 135 lb. per sq. in. absolute, corresponding to a compression ratio of about $5\frac{1}{4}$. Much difference of opinion has existed on the point of the variation of ηp with size of engine, and it may accordingly be well here to review some of the evidence available in support of

TABLE

illustrating the irregular variation of compression ratio in practice.

1910 *Car Engines.*

Bore in in.	Stroke in in.	Volume ratio of Compression.	Compression Pressure. lb. per sq. in. abs.
2.44	4.33	4.1	96
2 56	4.33	4.5	109
2.60	3.94	4.5	109
2.95	4.73	4.5	109
3.13	4.75	3.1	67
3.15	4.73	4.0	93
3.38	4.0	4.8	119
3.50	4.5	4.7	115
3.54	4.73	3 23	70
3.54	4.73	4.8	119
3.56	5.12	5.1	130
3.74	5.32	4.0	93
4.0	4.92	5.0	126
4.0	5.5	3.0	64
4.0	3.0	4.2	99
4.13	5.0	4.9	123
4.5	5.0	3.78	87
4.75	5.0	4.7	115
4.88	5.12	4 8	119
4.92	5.92	3.8	88
5.0	5.25	5.1	130
5.12	5.52	4.25	101
5.92	7.1	4.2	99

the view that an increase should be regarded as taking place with increase of bore:—

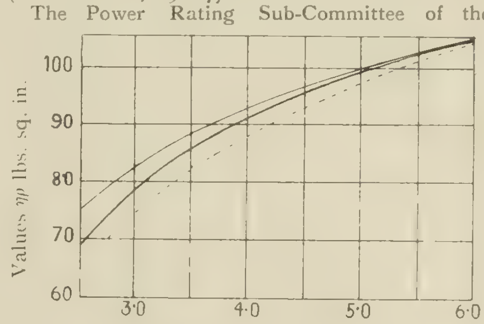
Mr. Remington says:—"By adopting suitable proportions of carburettor, induction and exhaust pipes and valves, it is possible to fill and scavenge an engine running at very high piston speeds and thus obtain as high a mean effective pressure as is usually associated with moderate, or even low, piston speeds." And again:—"From an examination of the normal tests of the Wolseley engines many lessons have been learned, conspicuous among them being the increase of brake mean effective pressure with increase of size."

Prof. Callendar from the results of some tests over a limited range was led to propose the relation:—

$$\eta p = k\ (1 - 1/d) \quad . \quad . \quad . \quad . \qquad (15)$$

as expressing the connection of ηp with d.—(Proc. I.A.E., 1906-7).

The Power Rating Sub-Committee of the



Cylinder Bore Inches.

FIG. IX.

TABLE

shewing increase of ηp with d on test of a series of engines built by the same firm.

d. in.	s. in.	n. r. p. m.	B.H.P. by Test.	ηp lb. per sq, in.
3 15	3·54	1675	18·7	80·6
3·35	4·33	1360	22.6	86·2
3 94	4·33	1360	32·0	88·2
3·94	5·12	1150	33·3	91·8
4·33	5 12	1150	40·8	93·0
4·73	5·12	1150	50·6	96·5
5·0	5·12	1150	59 5	102

Society of Motor Manufacturers, after consideration of a number of further cases, adopted an expression of this form in their report on horse-power formulæ issued in August, 1908. In a subsequent discussion of this report by the Institution of Automobile Engineers, Prof. Callendar said:—"The experiments of the A.C.F. on 96 engines ranging very equally from 65 mm. to 190 mm. (*i.e.*, $2\frac{1}{2}$ in. to $7\frac{1}{2}$ in.) in bore. . . . . are practically conclusive evidence of the increase of mean pressure with size." And further: "We cannot be far wrong if we take the formula $k\ d\ (d-1)$, which allows only half the rate of increase observed by the A.C.F. . . . . to represent the increase of ηp with d."

Mr. P. A. Poppe says: "We have now tested over 1,700 engines of bore from 3.15 in. to 5 in.; the formula we have arrived at for the calculation of brake horse power at a piston speed of about 1,000 feet per minute is:—

$$\text{b.h.p.} = 0.81\ (d - 0.79)^2\ .\ .\ .\ .\text{"} \qquad (16)$$

Comparing this with equation (13), we see that its form implies that

$$\eta p = 136\ (1 - 0.79/d)^2\ .\ .\ .\ . \qquad (17)$$

so that in these tests an ηp-increase with d is manifested. The following are results from equation (17):—

$d = 2\frac{1}{2}$ in.	3 in.	4 in.	5 in.	6 in.
$\eta p = 63\frac{1}{2}$	74	87	96	$102\frac{1}{2}$

The last table above shows the results obtained from tests of a series of engines built by the same firm; it will be noted that the value of ηp steadily rises as the engines increase in size. The piston speed in each case was about 1,000 ft. per minute; plotting the values of ηp against d we get the curve shown by the upper fine line in Fig. IX. The lower dotted line shows the variation in accordance with equation (17), representing Mr. Poppe's experience.

From the tables of engine data, Table I. has been constructed including 67 engines arranged in order of increasing bore, at a piston speed in the neighbourhood of 1,000 feet per minute. The volume ratio of compression is also tabulated in order to emphasise the absence of any observable connection with ηp; this is probably due in many cases to inadequate valve and piping areas and is also largely influenced by the valve setting. In Table I. we are clearly entitled to reject low values of ηp in any group. After consideration, the writer has thought it fair on the whole, to take the average of the three highest figures in each set; the result is shown in Fig. II., where the points are plotted down on the diagram, in black dots.

Excepting the anomalous result for $d = 4\frac{1}{2}$ in. to $4\frac{3}{4}$ in., a decided increase of ηp with d is manifested, and on trial it is found that the formula:—

$$\eta p = 130\ (1 - 1/0.85d)\ .\ .\ .\ . \qquad (18)$$

best resumes the points, as is indicated by the dotted curve.

The following values of ηp are furnished by equation (18):—

$d = 2\frac{1}{2}$ in.	3 in.	4 in.	5 in.	6 in.
$\eta p = 69$	79	$91\frac{1}{2}$	$99\frac{1}{2}$	104

Maximum B.H.P. per cylinder, from eq. (19).

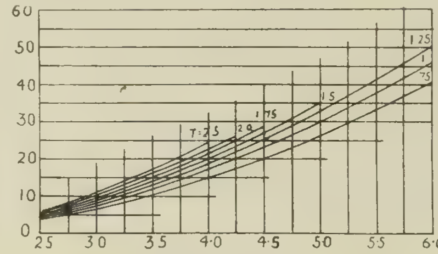
Cast Iron Pistons.

r	$d =$	Max B.H.P.	r	d	Max. B.H.P.
	2.5	3.33		2.5	4.6
	3.0	6.22		3.0	8.6
	3.5	10.0		3.5	13.6
	4.0	14.5	1.50	4.0	19.8
0.75	4.5	20		4.5	27.0
	5.0	26		5.0	35.2
	5.5	33			
	6.0	40.5		2.5	4.9
				3.0	9.2
	2.5	3.8		3.5	14.7
	3.0	7.0	1.75	4.0	21.2
	3.5	11.5		4.5	28.8
	4.0	16.5			
1.0	4.5	22.7		2.5	5.2
	5.0	29.5		3.0	9.6
	5.5	37.5	2.0	3.5	15.4
	6.0	46		4.0	22.2
	2.5	4.20		2.5	5.7
	3.0	7.9		3.0	10.6
	3.5	12.5	2.5	3.5	17.0
	4.0	18.3		4.0	24.4
1.25	4.5	24.9			
	5.0	32.7			
	5.5	41.2			
	6.0	50.6			

These are plotted down on Fig. IX. and furnish the middle curve shown by a full black line. The three curves are in substantial agreement; the writer herein adopts the mean, viz., equation (18), to express the variation of ηp with bore. Taking, then, equations (10) and (18) and substituting for σ and ηp in equation (13), we obtain the following rating expression for maximum performance:—

$$\text{Max. b.h.p. per cyl.} = \tfrac{1}{2}.d\ (d - 1.18)\ \sqrt{d^2 s/m}\ .\ .\ . \qquad (19)$$

The following table, calculated from this equation, gives figures for the range of sizes usual in car engine practice, the values of $\sqrt{d^2 s/m}$ being taken from the table. In Fig. X. also



Cylinder Bore Inches.

FIG. X.

curves are shown connecting max. b.h.p. per cylinder with bore, in accordance with equation (19). The results furnished by equation (19) are very high, and can only be approximated to at present for short periods of running and under the special conditions of racing where every refinement of practice is utilised. They appear, however, to be not unreasonable when compared with some actual racing engine performances, as the following six cases show:—

(1.) A four-cylinder 16-20 h.p. Sunbeam engine developed on test 54.2 b.h.p. at 2,300 revs. per minute; *i.e.*, 13.55 h.p. per cylinder. Here $d = 3.74$ in., $s = 5.32$ in., and $m = 7.1$ lb., whence, by equation (19):—

Max. b.h.p. per cyl. = 15.5.

Thus in this case the actual power was 87 per cent. of the rating according to equation (19).

(2.) The 100 mm. × 250 mm. (3.94 in × 9.85 in.) single cylinder de Dion racing engine was referred to in the *Autocar* for May 14th, 1910, as follows:—" It is claimed that this engine will develop about 35 horse-power, but it is probable that 30 will be nearer the mark."

Here:—

$d =$	$s =$	$r =$	$m =$	$\sqrt{d^2 s/m} =$
3.94 in.	9.85 in.	2.5	5.7	5.18

whence $\sigma_{\max.} = 3420$ ft. p.m., $n_{\max.} = 2080$ r.p.m., and from equation (19):—

max. b.h.p. = 28

which is in satisfactory agreement with the anticipated performance. This engine was fitted to the winning Le Gui car in the Coupe des Voiturettes competition of June, 1909; the course of roundly 280 miles was covered at an average speed of $47\frac{1}{2}$ miles per hour.

(3.) The 1908 Coupe des Voiturettes competition was won by a Delage car, fitted with a single-cylinder engine of 3.94 in. bore and 5.92 in. stroke. Using equation (2) we have $m = 5\frac{1}{4}$ lb.; the actual mass is said to have been very small, but the author does not know the exact figure. With $m = 5\frac{1}{4}$ we have $\sqrt{d^2 s/m} = 4.2$, whence, by equation (19):—

max. b.h.p. = 22.8.

The author is informed that the actual power developed exceeded this figure; this may well have been the case if the value of m were very low. The car covered the whole course at an average speed of just 50 m.p.h., so that the average horse-power, roughly estimated from the gross weight of car, and wind resistance, must have exceeded 20.

(4.) In the *Autocar* for December 17th, 1910, an illustration appears of a four-cylinder 130 mm. × 190 mm. Fiat engine nominally of 90 horse-power, and it is stated that this engine gives 143 horse-power on the brake, i.e., 35.8 b.h.p. per cylinder.

Here $d = 5.12$ in., $s = 7.48$ in., $\sqrt{d^2 s/m} = 3.66$; substituting in equation (19), we have therefore

max. b.h.p. per cyl. = 36.8,

a result practically identical with the stated maximum output.

(5.) In the *Autocar* for October 29th, 1910, there appeared an illustration and note on the performance of a 4-cylinder White and Poppe engine, 80 mm. × 130 mm. (3.15 in. × 5.12 in.), having an R.A.C. rating of 15.9. An "all out" test of this engine furnished the following results:—

r.p.m.	1900	2000	2200
b.h.p	38	44	47

Thus the maximum horse-power per cylinder was 11.75. The actual mass, m, of the reciprocating parts is not stated; if this be estimated by aid of equations (1) and (2) we get:—

For steel pistons ... ... $m = 3.44$ lb.

Hence $\sqrt{d^2 s/m}$ has in this case the value 3.84. Using equation (19) we have therefore for the maximum power rating:—

Max. b.h.p. per cyl. 11.9 with steel pistons

A result again in entire accord with the actual performance.

(6.) The 4-cyl. 3.54 in. × 4.73 in. Vauxhall engine in 1910 developed about 52 b.h.p. at 2,400 r.p.m. Writing in January, 1911, Mr. Pomeroy states:—"We have now got this up to 60 b.h.p. at 2,500 r.p.m., mainly by reduction of reciprocating weight and improvement in mechanical efficiency." Here by the proposed maximum rating formula, using m for cast-iron pistons, we have:—

Max. b.h.p. per cyl. = 13.5,

while with m for steel pistons:—

Max. b.h.p. per cyl. = 15.8.

the actual maximum powers being 13.0 and 15.0 respectively; the correspondence is here again very close.

From these instances it is clear that under racing conditions the ratings furnished by equation (19) are in satisfactory agreement with actual performances, and the writer therefore proposes the use of a formula of this form for rating purposes in "all out" contests for the future. To obtain the rating of any particular engine by the aid of this it would be, strictly considered, necessary to actually weigh the reciprocating parts in order to determine m. If this be considered impracticable, m could be inferred with a sufficient degree of accuracy from a formula of the type of equation (1) or (2) as soon as the bore and stroke were known. The great discrepancies in the power results frequently observed in engines of about the same size are largely due to the want of uniformity in practice in regard to valve diameter and lift, piping and valve setting. Some makers purposely fit small valves to prevent over-running of the engines; even with the same engine builder there is, however, frequently found to be considerable want of uniformity in valve proportions, considerations of economy in production sometimes resulting in the same size or lift of valves, or both, being used in several engines of differing bore and stroke, as the following series of figures illustrates. The upper line shows the actual valve diameters, the engines increasing in size from left to right; the lower line gives the calculated diameters from an equation to be given shortly:—

Actual δ :—	1.16	1.58	1.58	1.61	1.61	1.61	2.36
Calculated δ :	1.31	1.40	1.63	1.63	1.80	1.97	2.08

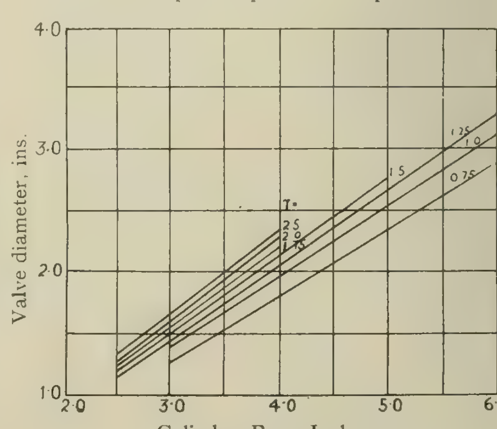
During the suction stroke $\pi/4.d^2 s$ cubic inches of mixture enter the cylinder through the inlet valve. If the engine makes n revs. per minute, there are, on the Otto cycle, $n/2$ suction strokes per minute, and hence $n/2.\ \pi/4.d^2 s$ cubic inches of mixture must pass the inlet valve per minute. If v'' denote the average velocity through the valve area $\pi/4.\delta^2$, in inches per minute, we have therefore:—

$$\pi/4\ .\ \delta^2 v'' = n/2\ .\ \pi/4\ .\ d^2 s$$
$$,, \quad = 3/4\ .\ ns/6\ .\ \pi d^2$$
$$,, \quad = 3/4\ .\ \pi\ .\ d^2 \sigma$$

whence, if v denotes the mean velocity in *feet* per minute, as $v = v''/12$, we get:—

$$v = \tfrac{1}{4}\ .\ (d/\delta)^2\ .\ \sigma\ .\ .\ .\ . \qquad (20)$$

where σ is the piston speed in feet per minute.



Cylinder Bore Inches.

FIG. XI.—Valve diameter in terms of Bore and r from Equation 21.

Values of v, calculated in this way are given in the tables of engine data herewith, and consideration of the figures thus obtained has led the writer

to propose that an average value of v be taken as *2000** feet per minute.

For attainment of the best power result, this should be reached at the maximum piston speed; substitute, then in (20) 2000 for v, and for σ the value given in equation (10), and we get on reduction :—

$$\delta = 0.287 . d . \sqrt[4]{d^2 s/m} \text{ in.} \quad (21)$$

This equation connects the valve diameter with the cylinder bore, and the function d^2s/m on the basis of maximum power rating herein proposed. Values of δ calculated from this equation are shown in the tables of engine data, and also in the accompanying table and the curves in Fig. XI.

Reference to the tables of engine data shows the great want of uniformity existing in current practice. In general, the values of δ furnished by equation (21), are somewhat in excess of actual sizes, though cases will be noted in which engines have valves fully equal to the calculated diameter. If λ be the lift of the valve, Fig. XII., a cylindric surface $\pi \delta \lambda$ is available for the passage of gas normally across it; if we equate this to the throat area, $\pi/4 . \delta^2$, we have, on reduction :—

$$\lambda = \tfrac{1}{4} . \delta$$

as the valve lift corresponding to diameter δ. So great a lift as one quarter of the diameter is, however, unusual in car engines on account of inertia; at very high speeds the tappet rod parts contact with the cam even when the strongest

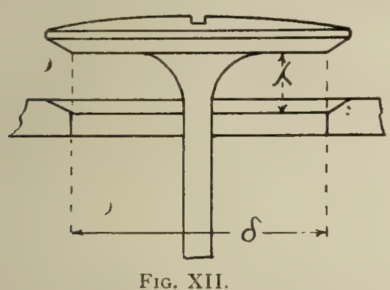


FIG. XII.

springs are employed. To overcome this difficulty and obtain sufficient valve area, some racing engines are fitted with a double set of valves to each cylinder.

*This figure is, of course, *8000* if the actual time of suction be taken; it makes no difference to equation (21).

Usually the exhaust and inlet valves are of the same diameter, so as to be interchangeable; usually, also, the lift is the same in each case, though in some cases the exhaust valve has 1/16 in. to 1/32 in. more lift than the inlet. Occasionally cases are met with in which the inlet valve diameter is greater than that of the exhaust, and there seems reason to think this may be good practice. The ratio of lift to diameter varies from about 0.13 to 0.30, the average value being about 0.2. The average valve setting for touring car engines, deduced from examination of forty different cases, is as follows :—

Inlet—Opens 12 degrees late. Closes 20 degrees late

Exhaust.—Opens 45 degrees early. Closes 6 degrees late.

Some makers do not vary the setting for high speed work, but if the valves are small a change of setting will often cause a marked increase in the output of power. This is well shown by the following test results from a 4-cyl, $4\frac{1}{2}$-in. × 5 in. engine with valves $1\frac{5}{8}$ in. in diameter :—

With Inlet opening 8 degrees late and closing 0 degrees late, and Exhaust opening 14 degrees early and closing 0 degrees late the *max h.p. was* 19.3 *at* 865 *r.p.m.*

With Inlet opening 13 degrees late and closing 22 degrees late, and Exhaust opening 39 degrees early and closing 10 degrees late the *max. h.p. increased to* 31.2 *at* 1,300 *r.p.m.*

The values of λ are somewhat higher for bores of 5 in. and above, but it is probable that a smaller lift would be found sufficient in these cases on account of the diminution in surface friction with increased valve diameter. The proposed maximum rating formula equation (19) for "all out" contests may be considered inapplicable to touring car competitions where the engines are designed with reference to normal running conditions with valves usually smaller in diameter than would be required by equation (21). In such an event we may deduce a rating formula based on the limiting velocity of gas through inlet value, assumed as 2,000 ft. per min. as follows :—

In equation (20) put $v = 2000$; then we have:—

$$\sigma = 8000 \, (\delta/d)^2 \text{ ft. per min.} \quad (22)$$

the maximum practicable piston speed being now determined by valve diameter; of course, if δ be designed in accordance with equation (21) we obtain for σ from (22) the same value as from equation (10).

In equation (13) substitute now for σ from (22) and for ηp from (18); then, after reduction, we have :—

$$\text{b.h.p. rating per cylinder} = 6 \, \delta^2 \, (1 - 1/0.85d) \quad (23)$$

a rating expression for touring car engines of simple form, involving the bore and inlet valve diameter only. It resembles the Marine Motor Association rating rule, but with the addition of a factor $(1 - 1/0.85d)$ for the ηp variation with size. The writer has not compared equation (23) with any actual test results so far, but it may prove to be worth consideration for the purpose suggested. In conclusion, it is hoped that the proposals set out in the above note, even if not found acceptable in their present form, may at least prove of assistance in enabling us to shortly reach a satisfactory rating rule for the petrol engines of automobiles.

TABLE *of valve diameter and lift, from Eq.* **(21)** . $\lambda = 0{\cdot}2\delta$.

$r =$	$d =$ in.	$\delta =$ in.	$\lambda =$ in.	$r =$	$d =$ in.	$\delta =$ in	$\lambda =$ in.
	3	1·29	0·26		2·5	1·21	0·24
0·75	4	1·82	0·36		3	1·52	0·30
	5	2·38	0·48	1·5	4	2 18	0·44
	6	2·87	0·57		5	2·77	0·55
	3	1·39	0·28		2 5	1·25	0 25
1·0	4	1 98	0·39	1·75	3	1·58	0·32
	5	2·57	0·51		4	2·22	0·44
	6	3·07	0·61				
					2·5	1.28	0·26
	2·5	1·16	0·23	2·0	3	1·62	0 32
	3	1·48	0·30		4	2·28	0·46
1·25	4	2·08	0·42				
	5	2·67	0·53		2·5	1·36	0·27
	6	3·26	0·65	2·5	3	1·68	0·34
					4	2·37	0·47

NOTES AND DISCUSSION.

Note on the Report.

BY DUGALD CLERK, F.R.S., M.INST.C.E.

THE formula recommended by the Committee gives a much higher b.h.p. rating for given cylinder dimensions than the well-known R.A.C. formula, and it is accordingly necessary to explain a little more fully than is done in the report that the new rating cannot be applied indifferently to motor car engines as ordinarily found at work upon the road.

The proposed formula gives the maximum b.h.p. rating which can in the opinion of the Committee be safely obtained from a petrol engine having the specified dimensions when run bolted down to a bench with every part working at its best. It by no means follows that this power is given by the same engine on the road, and it still less follows that any engine of the specified dimensions, whether old or new, could give the specified power. All that the formula means is that it is possible by carefully proportioning the valves by cutting down the weights of pistons, connecting rods and other moving parts, and by a careful adjustment of the carburettor and igniting arrangements to produce the power noted by the formula. The tests of 144 engines are given in the three tables of engine data, but out of these 144 only 124 are given with sufficient particularity to calculate the value of ηp and the piston speed at the highest tested brake horse-power. The lowest value of ηp found in these tests is 47.8 lbs. per square inch and the highest is 94.6 lbs. per square inch, with the tests in which the engines gave the highest recorded brake power. The value of ηp thus varied from 47.8 to 94.6. Of the 124 engines 62 gave ηp values of 70 lbs. per square inch and less. In 36 engines the value lay between 80 lbs. down to 70, and 19 engines gave 90 down to 80. Half the engines thus gave results of 70 and below.

It will be observed that the proposed formula requires the value of ηp to vary with the cylinder diameter, so that at $2\frac{1}{2}$ inches diameter $\eta p = 68\frac{1}{2}$ lbs. and at 5 inches $\eta p = 99\frac{1}{2}$ lbs. If the curve, Fig. IV. of the Report, be examined it will be found that the ηp value given for any particular diameter of cylinder is higher on the whole than the actual values obtained by 100 out of the 124 engines. This higher value has been adopted because the Committee consider that these particular pressures can be obtained in an economically working engine, although, as has been seen, comparatively few of the engines detailed in the tables of data have succeeded in obtaining these pressures.

It will be observed also that the formula is based on a piston speed varying with the stroke bore ratio from 1,200 feet per minute at $r = 1$ and to 2,100 feet per minute at $r = 2.5$. These piston speeds are also considered by the Committee to be safe piston speeds under certain conditions, but the tests prove that engines frequently fall below them and accordingly it is not surprising to find that engines at work on the road usually give much lower powers than would be deduced from the formula. To get an idea of the power usually exerted in actual work in a motor car upon the road by an engine of given dimensions, in my view the proposed formula should be multiplied by 0.6, that is, a little more than half the power indicated by the formula may usually be expected from the actual working engine in a motor car. This practically amounts to saying that the standard R.A.C. formula rating $= 0.4 \ d^2$ is correct where the stroke bore ratio is equal to 1, and this appears to me to be true, because in actual practice a piston speed of 1,000 feet per minute in ordinary work is but rarely met with, and an ηp value above 67 lbs. is not often found in actual work on the road, except under racing conditions. It is undoubted, however, that engines with longer stroke than bore do give higher piston speeds in ordinary work, and this is allowed for by using the proposed formula and multiplying it by .6.

When the R.A.C. formula was first proposed it gave with considerable accuracy the actual maximum brake horse power to be expected from certain cylinder dimensions. Notwithstanding the fact that the improvements made in engines in the last five years have made it possible to obtain much higher powers under carefully arranged conditions, it yet remains true that the R.A.C. formula gives with sufficient accuracy the brake horse power to be expected upon the road from the great mass of motor car engines now at work. If paraffin or petrol engines operating commercial vehicles be considered, the b.h.p. rating may be expected to prove to be still less. My present view is unaltered from that expressed by me in a paper read at the Royal Automobile Club on March 22nd, 1906. There I said: "Personally, I fear it is impossible to devise a rating rule which will enable one to accurately estimate the power of any engine from cylinder dimensions only. To obtain any such accurate rule would require uniformity of mean pressures, cylinder proportions, piston speeds and engine revolutions, which would tend in my view to impede progress rather than assist it."

The collection of data appended to the Report clearly proves that even to-day petrol engine constructors have not arrived at uniformity in mean pressures, cylinder proportions, or piston speeds, and accordingly that no rule can be formed which will accurately express the power which can be given by any engine on the road from bore and stroke alone.

The mean effective pressure which corresponds to the brake horse power developed by a petrol engine at a given piston speed has been called for some time ηp,—η representing the mechanical efficiency of the engine, and p the mean indicated pressure upon the piston. For the purpose of the Report it has been taken for the reasons given that ηp increases with the dimensions of the cylinder. In previous discussions this increase has been considered mostly due to diminution of cooling loss with increased cylinder dimensions, and Prof. Callendar, in his paper to the Institution some years ago, has discussed the change from

that point of view. It is clear, however, from the tables of engine data that the ηp variation depends on other things as well as cooling. For example, undoubtedly mechanical efficiency changes with increased cylinder dimensions. This was clearly proved in the Institution of Civil Engineers' tests with three gas engines of $5\frac{1}{2}$ in., 9 in. and 14 in. cylinder diameter, respectively. The mechanical efficiency of those three engines was respectively .84, .85 and .86. No doubt a similar change takes place with change of cylinder diameter in petrol engines. Probably the variation of mechanical efficiency for the range covered by the formula—$2\frac{1}{2}$ in. to 5 in.—is greater than in the case of the three gas engines, but sufficient experiments have not been made to prove that this is so. ηp will also vary with the total charge admitted to the cylinder, and thus with the same engine a higher value may often be obtained at a lower piston speed. This is clearly shown in the first table of engine data, where engines were tested at .9 of the maximum recorded brake horse power. In the case of engine No. 34, for example, ηp rises from 81.2 to 104.3 lbs. per square inch with a change of piston speed of from 1,000 to 700 feet per minute, and in engine No. 36 it rises from 65 to 100 lbs. per square inch with a change of piston speed of from 1,223 feet per minute to 715 feet per minute. Change in value of ηp therefore obviously depends on other things as well as change of cylinder dimensions. Considerable change can also be made by varying the compression ratio, although the tables of data do not prove this.

Another cause may produce variations in ηp. If an engine be worked with considerable excess of petrol so as to produce large quantities of carbonic oxide and other unburned gases in the exhaust, then ηp will be increased to quite a large extent. This is proved in actual practice and may be clearly seen from Dr. Watson's experiments, but it can be inferred from the nature of petrol. Petrol is so constituted that on complete combustion a certain moderate expansion occurs, whereas in coal gas a contraction takes place. If combustion be incomplete by reason of excess of petrol a still greater expansion occurs.

It is thus evident that there are several causes leading to variation in the value ηp, and it must not be taken that this variation is considered to be entirely due to cooling. If the mixture in a petrol engine, however, be adjusted for maximum economy, then ηp will vary in a more regular manner. If an engine maker, however, desires to increase ηp without reference to economical working he can do so even in a small cylinder by introducing more petrol than the oxygen can burn.

The Report and the tables undoubtedly prove increase of piston speed with increasing stroke bore ratio, but it should be noted that stroke bore ratio diminishes with increasing cylinder diameter, and the formula must not be taken as applicable to engines smaller than $2\frac{1}{2}$ in. diameter, or materially larger than 5 in. in diameter. The larger the diameter becomes in practice the more nearly does the stroke tend to equal the diameter.

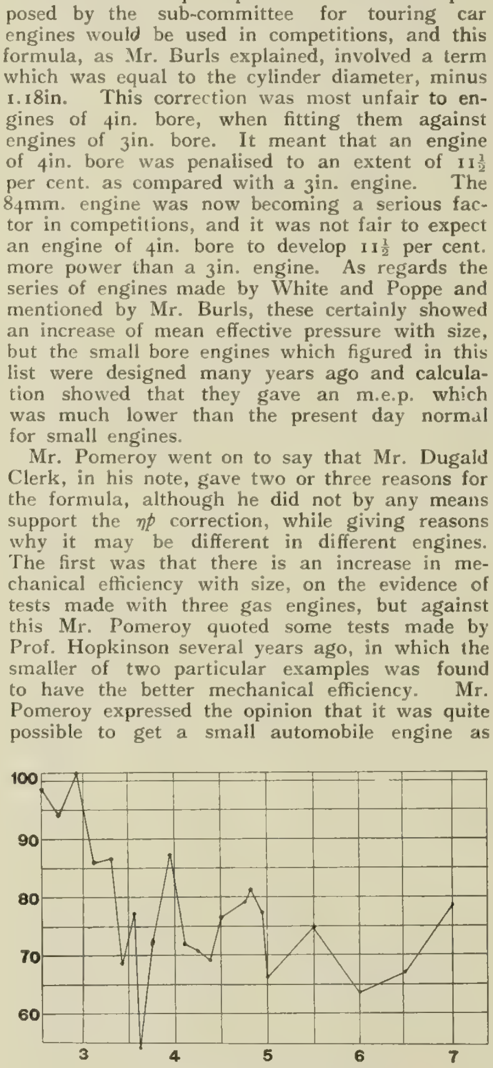
Discussion.

The discussion of the papers took place a week later than their reading, and there was a large number of speakers. Most of the points raised, however, were covered by Mr. Lawrence H. Pomeroy, and Mr. Lanchester, who in leading off said that the Callendar Correction was based on the assumption that, if a small engine was tested in comparison with a big one, it was impossible, owing to the more rapid cooling in the small engine, to obtain as high a mean pressure in that cylinder as in the larger one. It might be asked, why not put the compression up higher so as to get a bigger output of power? The reason why they had to stop where they did was because of pre-ignition occurring, and before that a hard ignition, which was due to the fact that as the combustion proceeded through the mixture it compressed the mixture in front, so that it detonated and caused a noisy explosion. That was what limited the compression. The cure was to drop the compression slightly. That was the limit they all worked to, and he asked, was that limit the same in big as in small engines? He had considerable evidence that it was not. There was also a theoretical reason why it should not be so. As the compression curve for the small engine dropped below that for the big one, so in the small engine the loss of heat during compression fell below the curve belonging to the bigger cylinder. Consequently, a smaller combustion space would serve before the pre-ignition point was reached. Still, the engine gave what would otherwise be, except for the extra cooling, much more power than the larger engine, and these two effects corrected each other. By employing a large compression it was possible to get a higher mean pressure in a small engine than with a large one, but there was a loss by the drop in the cooling curve on the small engine. He then gave examples from Knight engine practice in illustration of this theory.

He asserted, as the result of his experience, that he was convinced it would be utterly wrong to embody this cooling correction in any rating formula. On looking at Mr. Burls's painstaking and careful piece of work, one was apt to assume that all the engines were designed carefully, but he would remind them of a fact brought out in a paper by Mr. Mervyn O'Gorman, that the pistons of engines by different designers were not designed for maximum strength, but were made thicker from other considerations, and not with the greatest possible care and attention. No correction was valid unless it could be shown that the designers concerned were men of considerable ability, working independently of one another. It was well-known how few of them were working independently.

Mr. Pomeroy said that his ambition had been to dispose of the Callendar correction, but that it seemed this matter had been covered fully by Mr. Lanchester. After some further remarks in support of Mr. Lanchester's contentions, he said that it was quite possible the formula proposed by the sub-committee for touring car engines would be used in competitions, and this formula, as Mr. Burls explained, involved a term which was equal to the cylinder diameter, minus 1.18in. This correction was most unfair to engines of 4in. bore, when fitting them against engines of 3in. bore. It meant that an engine of 4in. bore was penalised to an extent of $11\frac{1}{2}$ per cent. as compared with a 3in. engine. The 84mm. engine was now becoming a serious factor in competitions, and it was not fair to expect an engine of 4in. bore to develop $11\frac{1}{2}$ per cent. more power than a 3in. engine. As regards the series of engines made by White and Poppe and mentioned by Mr. Burls, these certainly showed an increase of mean effective pressure with size, but the small bore engines which figured in this list were designed many years ago and calculation showed that they gave an m.e.p. which was much lower than the present day normal for small engines.

Mr. Pomeroy went on to say that Mr. Dugald Clerk, in his note, gave two or three reasons for the formula, although he did not by any means support the ηp correction, while giving reasons why it may be different in different engines. The first was that there is an increase in mechanical efficiency with size, on the evidence of tests made with three gas engines, but against this Mr. Pomeroy quoted some tests made by Prof. Hopkinson several years ago, in which the smaller of two particular examples was found to have the better mechanical efficiency. Mr. Pomeroy expressed the opinion that it was quite possible to get a small automobile engine as



A.C.F. test ηp against bore.

highly efficient as a large one, and went on to say that when Prof. Callendar proposed the ηp correction, it was proposed under two very rigid assumptions, the first being that the piston speed was constant and the second that the compression pressure was constant. The mixture strength was left out of the argument altogether. Therefore, he argued, the Committee had chosen a formula which included a factor depending upon piston speed being constant, although the formula as a whole ignored the piston speed. Further, one reason given in support of the ηp correction was the result of a series of experiments by the Automobile Club of France on 96 engines. Mr. Pomeroy was disposed to give great weight to these experiments, and showed a curve plotted from their results which is reproduced herewith. From this curve it seemed that the ηp correction could not rationally be included in the formula. Mr. Pomeroy then said that different results could be obtained from the tables of data according to the method by which they were plotted. He suggested that Mr. Burls had had a happy accident in connection with his method of plotting, and also suggested that the ηp correction should not be included in the rating formula, saying that the only effect could be to produce small bore engines with long strokes.

Continuing, Mr. Pomeroy expressed his strong disagreement with the form of the formula, which bore a resemblance to that suggested by Mr. Lanchester some years ago on the assumption that piston speed was a function of inertia stress. Working on this assumption, with reasonable piston weight and reasonable m.e.p., Mr. Pomeroy had found it would be necessary to run an engine at about 4,000 r.p.m. before the inertia stress on the neck of the connecting rod would exceed the explosion pressure on top of the piston and he thought that this speed was outside the limits which were worthy of consideration. If this idea of piston speed were excluded, it would seem quite a feasible scheme to compute h.p. on cylinder capacity only.

In conclusion, the speaker said that all engineers owed a deep debt of gratitude to Mr. Burls for collecting the data (given in the supplement published with this issue) which he considered would be of tremendous importance to the automobile engine designer in the future.

In replying, Mr. Burls said firstly, with regard to the increase of piston speed with stroke bore ratio, from considerations of gearing, comfort and so forth, the disadvantage of long strokes were more obvious in engines with large bores. Within the last two or three years an immense amount of attention had been given to engines of the 80 mm. x 120 mm. type, and he thought that, if an equal amount of attention was given to larger engines, they would exhibit a proportional improvement. As regards the 96 engines tested by the Automobile Club of France, Mr. Pomeroy had fallen into the error of plotting down a number of unreduced observations. It was not fair to take the maximum h.p., because there was a good deal of wire drawing, and tests ought to be taken before this occurred. Mr. Baillie had analysed the French engines' results, and Prof. Callendar had examined the analysis. The Committee attached great weight to the results, and in Prof. Callendar's view they found almost conclusive evidence of increase of mean pressure with size.

As regards Figs. 1, 2, and 3 in the Committee's report, these were not worth consideration, because the 96 A.C.F. test engines were also shown at Fig. IV. of the Committee's report, and though this curve fell a little below the White and Poppe curve, it was practically parallel to it. He did not think this could possibly be a mere coincidence.

Editorial Note.

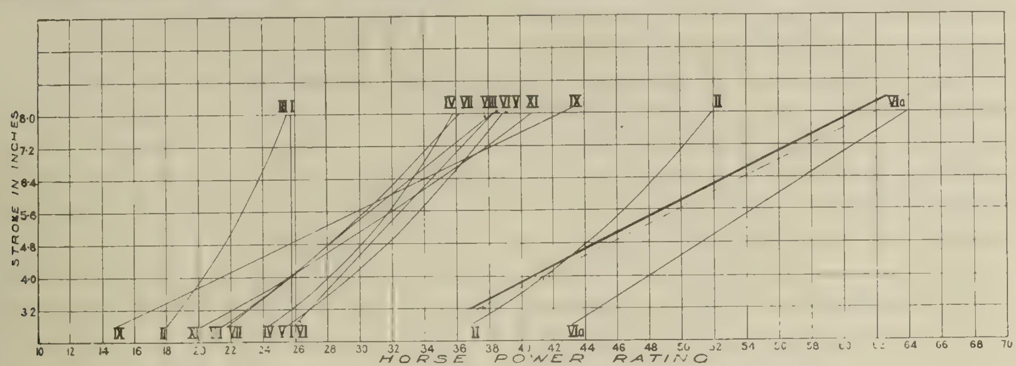
From the discussion, some of which is reported above, it appeared that many of the leading members of the Institution do not regard either the proposal of the committee or that of Mr. Burls as satisfactory. Undoubtedly, the work done is of great importance and the data gathered together by Mr. Burls is extremely valuable, but it may be questioned whether the committee have not given a little too much attention to proving the impossibility of a particular "impossible." Their aim has been to discover a reasonably simple formula which would give the power of an automobile engine in terms of bore and stroke alone, while being fairly accurate. The form of equation recommended is certainly not simple and is not extremely accurate, although it is more accurate than the majority of the other formulæ which have been suggested from time to time.

In the recommended formula the constant terms represent too much—One maker may make a series of engines in which ηp varies with size, while another may counteract this tendency by compression alterations. One may use cast iron pistons and another steel. One may use large valves and another small valves.—It would be possible to multiply these variations at great length, but it is not necessary to do so, because the point of importance is that these variations are mostly in existence for good reasons. The practice of one maker is not necessarily better than that of another, because so much depends upon the point of view. Had the ambition of the committee been to produce a *scientific* formula they could best have done so by means of a far more elaborate equation, which would take all the variables into account, and we believe that a formula which was accurate when every detail of

the engine was known would be useful, because it would enable the effect of small variations in valve area, and what not, to be studied. If, on the other hand, a formula is needed for general more or less rough and ready comparison of normal engines, there are simpler formulæ than the new one, which are not much inferior in all round accuracy. In the correspondence columns of *The Automobile Engineer* last month, there was a diagram showing the power of a four-inch engine with varying stroke as given by different formulæ. This is reproduced on this page with the committee's formula and another added to it. It will be noticed at once that these correspond in form with the Dendy Marshall, and obviously, if the constant in the latter was altered it could be made to very nearly coincide with the committee line. This is striking, because the Dendy Marshall is merely an elaboration of the R.A.C. formula to take in stroke. Its constant is a mere arbitrary figure and does not depend upon ηp variations, or indeed on any other variations. The heavy black line in the figure is the committee rating for a four-cylinder four-inch engine with stroke varying from 2.8 to 8 inches, and immediately beneath it there is a dotted line, which is obtained from the modification of the Dendy Marshall :—

$$\text{b. h. p.} = \frac{D^2 S.N.}{12} + 19$$

which brings the two curves together. Of course, it is not suggested that this is an actual formula, it is merely cited as showing that a simpler form might be almost as accurate as the official suggestion—quite accurate enough for ordinary purposes of comparison. It is practically the same as the contention put forward by Mr. Pomeroy that the power of normal engines varies nearly in proportion with their volumetric capacity.



Rating Curves for 4-inch Bore Engine by Different Formulae.

Formulae.

I. R.A.C. $\cdot 4.D^2.N.$

II. Automobile Engineer (Lanchester) : $1000\ D^2.N.\left(\frac{2R+1}{R+2}\right)$

III. Remington (Bourne) : $\cdot 2.D^2\sqrt[3]{S}.N.$

IV. Lanchester (a) : $\cdot 5\ \frac{D^2.}{\sqrt{\left(\frac{D}{S}+\cdot 3\right)}}.\ N.$

V. O'Gorman (Lanchester) : $\cdot 4.D^{1\cdot 6}.S^{\cdot 4}\,N.$

VI. S.M.M.T. : $\cdot 2.D.N.\ (D-1)\ .\ (R+2).$

VIa. ,, : $\cdot 33.D.N.\ (D-1)\ .\ (R+2).$

VII. Lanchester (b) : $\cdot 4.D^{1\cdot 5}.S^{\cdot 5}\,N.$

VIII. Henderson (Lanchester) : $\cdot 2.\ D.\ N.\ (S+D).$

IX. Dendy Marshall : $\frac{D^2.\ S.\ N.}{12}$

X. Thornycroft : $K.D^2.\ S^{1\cdot 5}.\ N.$

XI. Henderson : $\cdot 4\,D^{\frac{4}{3}}.\ S^{\frac{2}{3}}.\ N.$

CORRESPONDENCE.

No responsibility is accepted by the Editor for the opinions of correspondents, and the right is reserved to omit any portion of a letter if it is considered desirable to do so. Letters to be inserted with a nom de plume must be accompanied by the name and address of the writer. If publication in the next issue is desired letters should reach the Editor in no case later than the 20th of the preceding month.

DOUBLE ACTING ENGINES.

Sir,—Mr. Bertram C. Joy's suggestive article in your last issue touches an interesting problem in the possibility of the adoption of the double-acting internal combustion engine for automobile work. A two-cylindered double-acting engine such as outlined would certainly make a compact unit, as well exemplified by the diagrams ; in connection with which it is pleasing to think that the fashion of big bonnets seems to be passing (at least as regards automobiles), and compactness is now appreciated more at its real value. Firms like the Lanchester and N.E.C. have all along realised that unobstrusiveness was an excellent feature for a carriage motor ; and now there are new types put forward (such as the Banner and H engines referred to in your same issue, which claim compactness as one of their virtues), so that the double-acting type would be quite justified on this score

But as pointed out, the double-acting engine entails its problems, cooling and others, and it would be interesting to know which would give the best result of the two timings allowed, the crank throws being at 180°. In one case the impulses for the couple of revolutions would be—the first two in sequence on one crank (one impulse down and one up), followed by the final two similarly in sequence on the other crank. Or with the alternative setting—the first two impulses acting down (first on one crank then the other), and the final two in similar sequence acting up. The question of balance would be the same as with the usual two-cylindered engine, there being an unbalanced couple ; further, the increased mass of reciprocating parts entailed by the crosshead construction might tell against steady running at high speeds, though after all such construction is but following sound steam practice.

An alternative that, it seems to me, might offer greater feasability would be a tandem disposition of cylinders, which, while not being quite such a plain unit as a double-acting cylinder, would afford much the same compactness and simplicity, along with additional advantages. Cylinders so arranged could be both open ended, so that there should be no special difficulty about cooling ; piston rod and gland could be obviated by the adoption of external coupling between the tandem pistons ; and power would be delivered better, there being a constant downward pressure. Certainly a tandem engine offers more promise of commercial success than most of the ingenuous but impracticable alternatives to the present orthodox type that have been proposed from time to time.

There is a number of large gas engines built on these principles, such as the Crossley and Premier tandem engines, the Cockerill double-acting type, the Westinghouse and Nurnberg tandem double-acting engines, and others ; and the fact that these are regularly made and run well cannot but have some optimistic bearing on the matter. But this is a case where the greater does not include the lesser ; and these large engines cannot well be taken for data to apply to automobile motors, chiefly because of the comparatively low crankshaft speed and the greater facilities large construction offers for piston cooling. However, as some off-set to this, it may be borne in mind that, just because size does so modify design, there are mechanical possibilities which, while they are not applicable to large engines, may well be quite legitimate and advantageous in small types.

R. G. WELLS.

CARBURETTOR ACTION.

Sir,—I notice that in your February issue Professor Morgan has again seen fit to criticise an article of mine, this time dealing with Carburettor Action. Let us first clear the ground a little in remembering that Professor Morgan is interested in a certain carburettor based upon his theory and upon the results of some laboratory experiments he has made. Also it is well to bear in mind that I am not interested in any particular carburettor, and that my observations and the curves "Y" are based upon practical tests which I have conducted upon the road and track for the last five or six years, with various carburettors and cars. Professor Morgan says that he is not quite clear as to how my curves were obtained ; I thought I had indicated this in the article in question.

The instrument was used as a basis for determining a number of points on the chart and various experimental results marked in the ordinary way upon this chart and curves drawn through the points striking a mean value as far as possible. I do not think, therefore, that Professor Morgan need find any obscurity about the matter, nor need he imperil himself by argument.

I particularly stated that Unwin's formula was applied only to suctions up to 150 mm. of water, and it is a well-known fact that the efflux from a small jet is not proportional at very high suctions and air velocities. It is for this reason that experimenters have been at work for so many years upon the design of various devices to counteract this tendency for increased discharge. Referring again to the .90 mm jet, I have obtained results from a Claudel Hobson jet 0.95 mm. diameter of particular construction, which gives a considerably less petrol flow at high speeds than is indicated in my chart, namely 1.4 gallons per hour with an air velocity of 300 feet per second ; this jet gives also a flow of 1 gallon per hour with an air velocity of 200 feet per second. Such a curve as is characteristic of this particular jet will be practically a straight line in the Y group of curves.

Professor Morgan says that it is to be decided whether or not certain results must be rejected, and insinuates that mine are incorrect, as he says they depend on estimations which are nebulously outlined, etc.

In reply, I can only state that my experimental work shows that a great deal depends upon the design of the jet itself and the experiments which I conduct are on behalf of practical men to assist them in the solution of engineering problems.

ROBERT W. A. BREWER.

[This correspondence seems now to have reached its limit of usefulness, and will therefore be closed.—Ed.]

CASTELLATED SHAFTS.

Sir,—It is a long time since I read a more interesting paper than that of Mr. Larrard, on "The Strength of Castellated Shafts," as reported in *The Automobile Engineer* of last month, but while reading it I could not help thinking that, in the making of castellated shafts and their sliding sleeves, the question is considered from the wrong end, and as an outcome of this

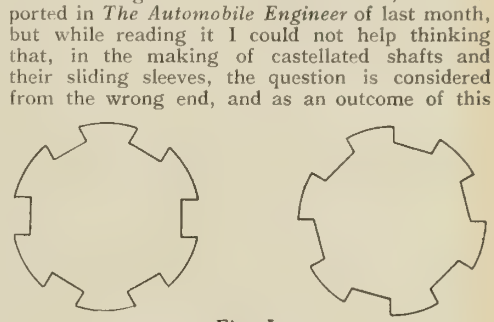


Fig. I.

fact the best results are not obtained. At present we take the major diameter of the castellated shaft as a starting point in manufacture, turn the shaft to this dimension, and then mill the grooves, adapting the sleeve to the result so obtained. This result gives sections as in Fig. I., in which the surfaces at the maximum radius from the centre follow arcs of circles, while those at the minor radius from the centre—the bottoms of the grooves—are flat.

Now, as a mechanic, and from the manufacturing point of view, I contend that this is wrong,

for the correct machining of the inside of a sleeve is undoubtedly the more difficult operation of the two (especially in sleeves of small diameter), and we ought therefore to subordinate our easier and more accessible process to the more difficult, if the best results are to be obtained.

To follow out such a method we should start from the minor diameter of the sleeve, bore that out in the usual way, and cut keyways. But in cutting these keyways we should naturally select the simplest form of groove that would introduce the least chance of inaccuracy. Such a groove would obviously have flat sides, and we should thus have to fill with our castellated shaft a space as in Fig. II., in which the curved and flat surfaces of the section are located in exactly the opposite positions to those in Fig. I.; in other words, the outermost surfaces are flat, and the innermost curved to circular arcs.

To produce the necessary shaft turning would then be unnecessary, the profiling being done by milling cutters of approximately Fig. III. cutting section.

It may be argued that such a section would give an unpleasantly sharp re-entrant angle on the sleeve at the point marked A, Fig. II., and so it does with only the three feathers I have shown, but increase the number of feathers, and such an objection rapidly vanishes. To emphasise this point it is only necessary to imagine one feather, of the size that would be required if there were six feathers to the shaft, and this

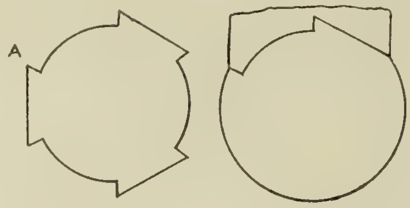


Fig. II. Fig. III.

shows the alteration of angle at A. Even with three feathers, however, I venture to think that the sleeve in its more advantageous momental position would, in most designs, contain ample material to counteract such an objection in practical working.

In making such a suggestion as the foregoing, I do so with the greatest respect for existing practice, and, in fact, expect to hear that my idea is far from new and has such serious objections as forbid its general adoption, for there are invariably strong reasons for every existing practice. I should like to hear from others, however, the objections against any such method as I have proposed.

A. R. HENDON.

ARRANGEMENT OF HUB BEARINGS.

Sir,—No doubt the arrangement of hub bearings has not had so much attention as the majority of the details of the modern car, and renewals have to be made more frequently than they ought.

I do not think that the condition of affairs is going to be altered by simply keeping the water out, but in many cases replacement would not have been made if thrust bearings had been fitted, notably on taxi cabs.

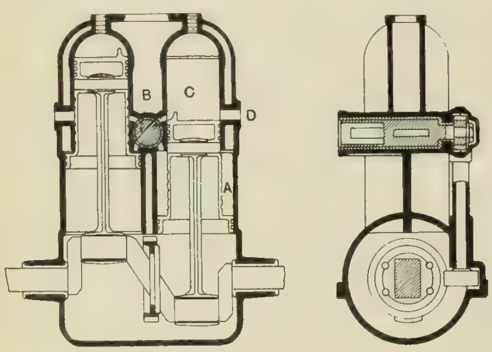
GEO. H. RODWAY.

RECENT AUTOMOBILE PATENTS.

By Eric W. Walford, F.C.I.P.A.

A Two-Stroke Engine.

The engine is provided with twin cylinders, the pistons of which are extended below in tandem fashion, so as to act as compressors in the spaces A. As one

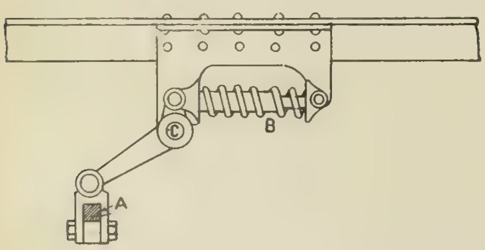


piston descends the space A is filled through a passage (not shown) in a rotary valve B, which communicates with the carburettor. On the upstroke the gas drawn in is compressed and passed through the transfer passage C in the rotary valve so as to blow against a baffle on the piston in the other cylinder and be directed to the top, forcing out the exhaust gases through the usual cylinder ports D. The rotary valve is driven by worm gearing from the crankshaft and, as stated, serves the two functions of supplying the pump and also causing the gas so drawn in to be forced across into the other cylinder element.

B. T. Hamilton. No. 5,873/10.

A Springing System.

This system consists in interposing between the back axle and the spring a lever system, such that, as the load increases,



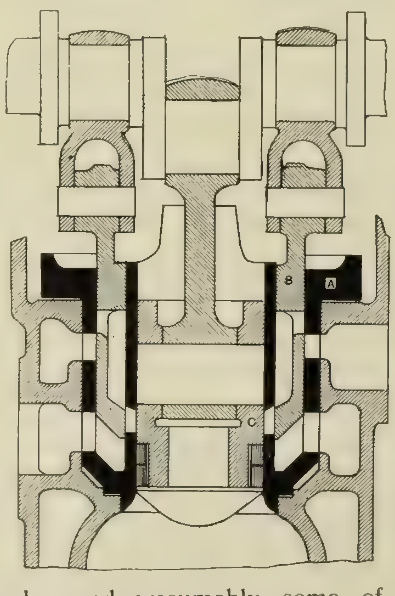
the effective length of the lever directly acting on the springs will be decreased, whilst the effective length of the other lever increases. The drawing shows a simple bell-crank lever connecting the axle A to a helical spring B, the lever being pivotted at C. As the longer arm of the bell-crank lever becomes more horizontal when the load increases, its effective length increases, whilst the reverse is the case with the shorter arm, and a decreasing spring compression is obtained.

M. M. Brophy. No. 570/10.

An Overhead Slide Valve Engine.

The top of the combustion chamber is bored out to receive a duplicated stationary guide member A, in the walls of which are ports, whilst intermediately of the walls works a valve sleeve B, reciprocated by means of cranks. Inside the inner wall slides a valve piston C operated by a separate crank on the overhead shaft, this piston serving periodically to cover and uncover the ports in the inner wall of A, whilst the outer sleeve acts as a distributing member uncovering one of the sets of ports in the outer wall.

It will be noticed that the piston C is exposed to the pressures within the

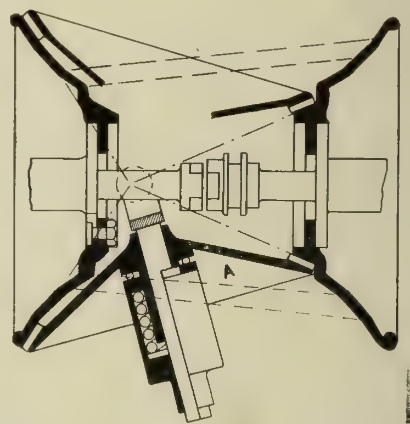


cylinder, and presumably some of the power developed in the engine is transmitted through this and the valve crankshaft. Further, the valve piston, by its movement in relation to the cylinder, varies the volume of the compression space.

J. Fielding. No. 48/10.

A Simple Change-Speed Gear.

This gear comprises two conical discs one on the driving and the other on the driven shaft, with a number of conical steps or tracks, the apices of each of which lie on the axis of the shafts, but are



at different points along it. Intermediate of the two discs are the adjustable cones A, which are mounted upon guide stems having inner ends free to move along the shaft. Thus as the guide stems are moved the discs A rock from one pair of tracks to another pair of different relative diameter. The intermediate discs A are spring engaged so as to provide the necessary frictional contact and for reversing purposes the two shafts are clutched directly together.

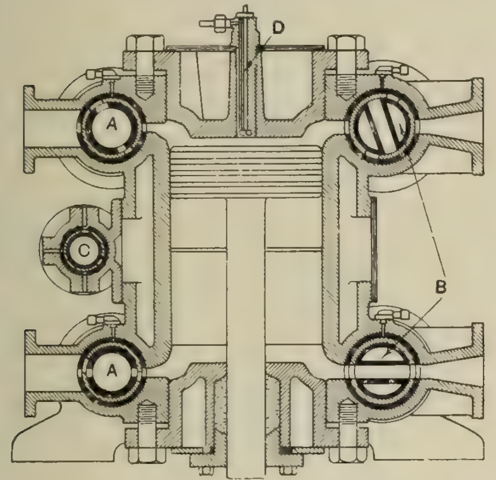
H. Austin. No. 974/10.

A Double-acting Two-stroke Engine.

The engine is provided with air inlet valves at A, and exhaust valves at B, the latter being cooled by the passing of water through the hollow valve members. Each valve is constituted by concentric sleeves which are oscillated for the most part in opposite directions to bring the ports in the sleeves into and out of line. At C is a fuel valve operating on a similar principle, the cycle being as follows:—Ignition takes place at one end of the cylinder, forcing the piston down. Near the bottom of the stroke the exhaust valve B opens and slightly past the dead centre the air inlet valve A is opened to allow a scavenging charge of compressed air to sweep through the cylinder. The exhaust

valve then shuts, the air valve closing a little later, whilst the air in the cylinder is compressed by the return of the piston. Either during compression or near the dead centre the fuel is injected through a valve D and the gas is fired.

A feature of the invention is that all the sleeves constituting the valve are actuated by a single pair of eccentrics, and that reversing is obtained by actuating the eccentric rods through Stephenson link motion. The fuel valve C is constructed to admit compressed air alone at starting, so that the engine operates at such times after the manner of a steam engine. As

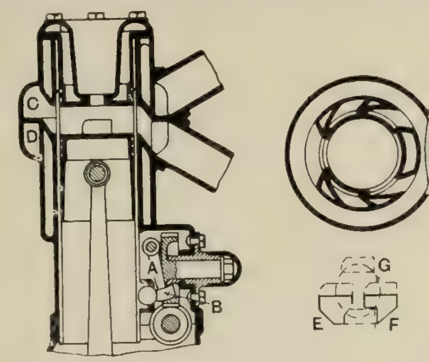


soon as the engine is under way, the valve C is adjusted to supply fuel as desired.

H. S. Johnson. No. 111/10.

A Slide Valve Engine.

This engine is of a type which has been previously described in these columns. It has a single sleeve with a port G, or set of ports, the sleeve having im-



parted to it a combined reciprocating and oscillating motion, a point on its surface travelling through practically a circular path. The sleeve is driven through a ball-ended crank A, attached to a driving gear wheel, so that the sleeve practically repeats the circular motion of the gear wheel, inasmuch as a heel portion B of the link engages the gear wheel.

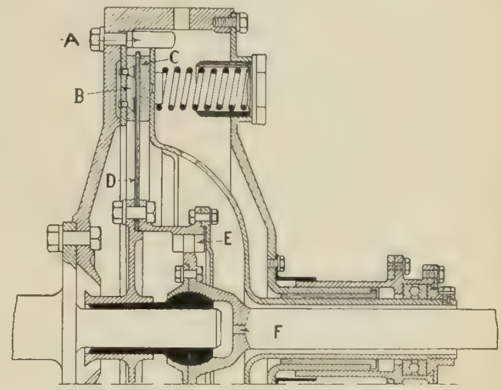
Around the top of the cylinder there are arranged two passages C and D. The former is the inlet passage and the latter the exhaust. Each of these has ports E and F respectively, arranged at the same level. The sleeve port is shown at G in the smaller view, and also its path of rotation, and it will be quite clear how, with the ports shaped as shown, first the inlet passage is opened up, then the sleeve port G is brought into the position illustrated for the compression and firing strokes, and is then caused to register with the exhaust port F, cut-off taking place when the port G is in the lower dotted position. The sleeve motion is so peculiar that it is not easy to describe, but it is made plain by the drawing.

C. B. Redrup. No. 291/10.

A Disc Clutch.

The driving member has attached to it by means of projecting bolts A a frictional surface B. The driving bolts engage holes in the movable driving member, which carries an annular frictional surface at C. These two frictional surfaces grip the driven disc D, which is attached to a sleeve having an internally toothed ring part E engaging a corresponding part on the driven shaft F. This has a spherical bearing upon a sleeve carried on the driving shaft, so that complete freedom of movement of the driven shaft in relation to the driver is provided.

The driven clutch shaft is formed hollow so that lubricant can be forced into the spherical bearing, whilst the whole



of the clutch may be closed by a cover which carries the spring abutments.

Albion Motor Car Co., Ltd., and T. B. Murray. No. 17,511/10.

INGENIOUS AND PECULIAR DETAIL DESIGNS.

At the recent Olympia and Paris automobile exhibitions seekers of the abnormal found very little to interest them, but particulars of a few peculiar designs which were noticed by the writer may be of interest, for even though some of them are by no means new, they are certainly not generally known. An excellent example of departure from the ordinary is the gate change system shown in Fig. I. This is fitted to Mass cars, and should certainly fulfil its purpose, which is to give a free action. The brake lever shaft, of course, does not slide, and neither does the tube outside it to which the change-speed lever is attached. All torsional stresses are taken by the tube—that is to say, the operation of moving the gears is performed thereby in the usual way, but

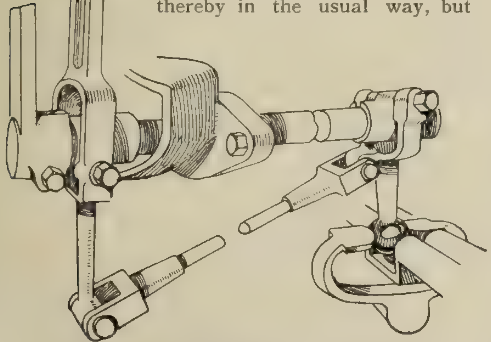


Fig. I. The Mass gate mechanism.

the angularly placed rod moves the ball across the strikers by swinging the striking arm. The arrangement is not very neat, but it is decidedly ingenious and very easy in action.

Another unusual design is to be found on the White petrol chassis, where the oil is forced to the main engine bearings, but instead of being carried to the big ends by way of drilled holes

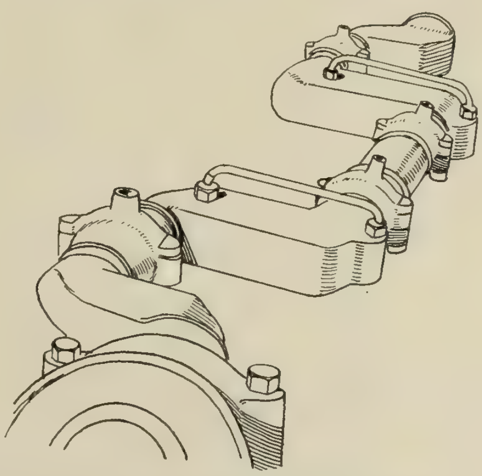


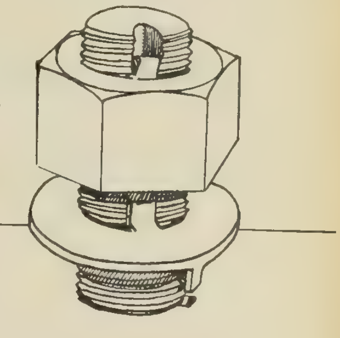
Fig. II. The oil leads on the crankshaft of the White petrol engine.

in the shaft and webs, there are external copper pipes as shown. The disadvantages are obvious, but the shaft is presumably weakened less—a matter of importance with only two main bearings.

A more useful device is shown in Fig. III., and this was mentioned in the December, 1910, issue of *The Automobile Engineer.* It will be seen that the bolt is slotted and a small peg hole is also made in the bed, while the washer has two tongues. The nut can thus be removed with the same ease as it can be put on, and an extra security can be got if required by bending over the part of the tongue which projects above the nut.

Fig. IV. shows a clever method of securing the valve caps on Delaunay-Belleville chassis, and is self explanatory. It seems a somewhat compli-

Fig. III. The Lamplough nut lock washer.



cated means to a quite simple end, but it has the advantage of cheapness, while it cannot very well fail to be effective.

Having mentioned retaining devices, two or three others were observed which are quite worthy of description, though they are not new.

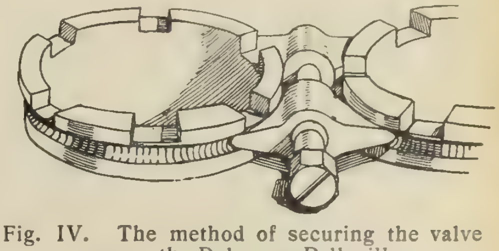


Fig. IV. The method of securing the valve cups on the Delaunay-Belleville.

One is shown in Fig. V., and is the stiff bracket arrangement employed to support the silencer on

the Imperia cars, while the other is the catch used to hold up the metal undershield on Benz chassis.

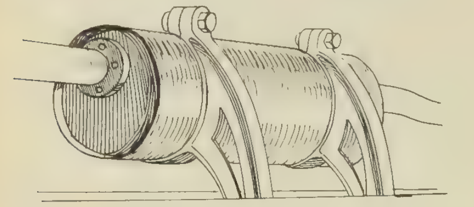


Fig. V. Method of attaching the Imperial silencer.

Fig. VI. makes the action clear, but it may be added that the amount of spring in the shield itself, as well as that in the steel wire loop, is sufficient to keep the shield pressed tightly

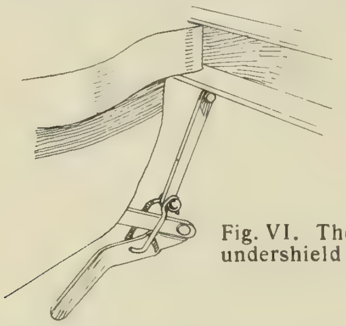


Fig. VI. The Benz undershield catch.

against the frame so that there is no suspicion of looseness or rattle, and, of course, the detachment is instantaneous as soon as the catches have been thrown over. It seems probable that

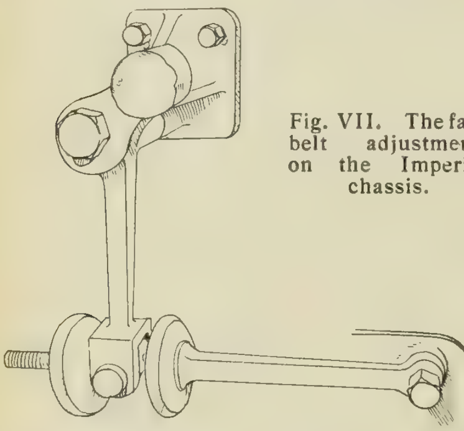


Fig. VII. The fan belt adjustment on the Imperia chassis.

this fitting would not be much affected by mud or water, as there is very little which could stick through rusting.

Much ingenuity has been given to the design of fan belt adjusters, and some extremely simple patterns have been devised, while others are distinctly elaborate. An example of the latter type is that fitted to the Imperia and shown in Fig. VII. Both the milled edged discs are adjustable, and the belt can be set to any desired tension very easily, but on the other hand the cost must be very much more than that of a spring-controlled device, and it is doubtful whether the latter form is not equally effective.

Fig. VIII. is a detachable wheel mechanism fitted to a few of the chassis at the Paris Salon, and is a sketch of the stub axle with the wheel removed. The usual bolts and flanges are used in the manufacture of the artillery wheels, only the bolts have cylindrical heads which register with the slots in the edge of the disc. There is a disc on each hub, of course, and those fitted to the back axle are combined with the brake drums. It is claimed that the slots give the same security as holes would do, while the bolt heads are not so liable to stick through rusting. Final locking is performed by the hub cap in the customary way.

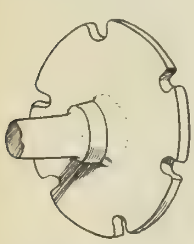


Fig. VIII. Stub axle for detachable wheel.

A part of a chassis which is often difficult to detach is the carburettor, and it is perhaps curious that there are so few easy attachments for this engine fitting, especially when the great variety of magneto clips are remembered. The form of fixing shown in Fig. IX. was also observed in Paris, and seems to be decidedly convenient. Each pipe ended in a collar or circular

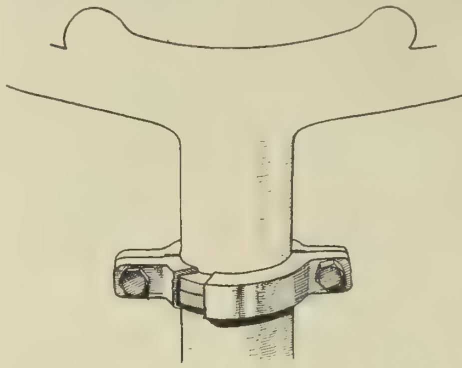


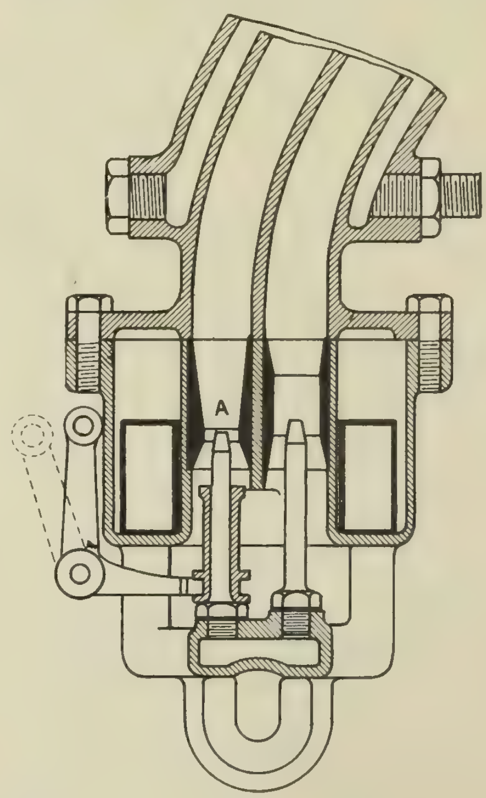
Fig. IX. A quick carburettor attachment.

flange, flat faced and coned slightly at the back so that tightening the clip also pressed the flat faces together. As shown, the clip was secured by bolts and nuts, but there seems no reason why it should not be joined by a hinge pin on one side and a hinged bolt and wing nut on the other side, precisely after the pattern of a domestic seltzogene. It is usually assumed that the idea of using "snap catches" of any kind is distressing to the engineering mind, but there is no reason why it should be so, and signs are not wanting that the newer industries are producing a new type of designer who is himself sufficient of a user to appreciate the very real convenience of adjustments which do not call for the use of any tools.

THE NAPIER CARBURETTOR.

THE Napier carburettor is a good example of the existing tendency to simplify all parts which are likely to need attention from the car owner, and to render any adjustment which may be necessary an easy and accessible matter.

Fig. I. is a sectional view of the carburettor, showing the only moving parts, and the absence of any spring-controlled valves or other delicate mechanisms likely to require attention is noticeable. The float chamber is arranged concentrically with the jet chamber, in a manner similar to that first adopted by De Dion Bouton. Petrol is supplied to two jets of different sizes, above which are tapered choke tubes that can be removed for alteration when the carburettor is first adapted to its particular engine. On the smaller jet a



sleeve will be observed, actuated by a bell-crank lever and capable of a sliding motion until impeded by the lower portion of the choke tube. In this position engine suction can only draw air between the inside of the sleeve and the external circumference of the jet, thus concentrating round the jet nozzle, which therefore delivers a greater quantity of petrol than it would supply normally. A considerable quantity of petrol flows down the jet sides and is retained by the top of the sleeve, forming in effect a small surface carburettor from which a very rich mixture is obtainable, and facilitating easy starting. Once the engine has been started, the sleeve can be lowered, and the jet operates in its usual manner, until it is cut off and the larger jet uncovered by the throttle position necessary for increased speed.

The double spray pipe casting is furnished with a hot water jacket, the inlet to which is seen in the drawing, the outlet pipe being placed slightly below the throttle, at a point where the pipe is branched to the cylinders. While the enriched mixture is necessary for starting purposes, it is conceivable that it might become too rich by this arrangement, should the engine require much swinging, but this is a drawback appertaining solely to the novice and easily overcome with experience. There seems but little which would cause trouble, and everything is particularly accessible, should trouble occur, while the general arrangement is both compact, neat, and well ordered.

REVIEW.

THE reports of proceedings of learned societies are too often looked upon as mere bookshelf ornaments, and it is improbable that the reports of the I.I.A.E. are used for reference to anything like the extent which they deserve. It is, therefore, worth while to point out that in the case of a society with a narrow specialised field there is little in the proceedings which is not of value to every member, and many a doubtful point of design has been debated lately by the I.I.A.E. For this reason every improvement in the record of the work of the Institution which makes for easy reference is to be welcomed, and many such are to be noticed in the latest volume.

The fourth volume of proceedings has been published during the last month and, although the contents are, of course, already well-known to the bulk of our readers, the discussions which have in several cases been communicated in writing make a reperusal of the papers desirable and interesting. The session 1909-1910 is dealt with fully, and the volume is very well indexed and arranged. It is thus convenient for reference, while the illustrations are so placed that there is no trouble in referring to them from the text, a state of affairs which is not universal with books of this class. Generally, regarding both the intrinsic interest of the papers and the vigour of the discussions, Volume IV. is distinctly an advance on either of its three predecessors, while it is also superior as regards arrangement and printing.

MISCELLANEOUS.

W. G. WALKER AND CO. inform us that in consequence of the article on the White and Poppe engine testing system, in the August, 1910, issue of *The Automobile Engineer*, they discovered Messrs. White and Poppe were unknowingly infringing Mr. W. G. Walker's Patent No. 2743 of the year 1904 in connection with the fan Dynamometers as mentioned in the article. This matter has now been amicably settled, but to protect themselves Messrs. Walker wish it to be known that any person or persons using fan brakes other than those supplied and manufactured by them (which bear the name of W. G. Walker, Westminster), they may commence proceedings for infringement of such patent.

AT THE MEETING OF THE I.I.A.E., on December 14th last, Mr. T. B. Browne took part in the discussion, but in our report thereof his initials were given wrongly as G. W. As there is another member of the Institution, Mr. G. W. Brown, confusion might easily have arisen, so we take this, the earliest possible, opportunity of correcting the misprint.

RUBERY OWEN AND CO., LTD., inform us that Mr. Rubery has lately retired from active participation in that business, which will continue to be carried on as heretofore, and with the original name, by Mr. Owen. We understand that the plant for pressing frames is constantly growing and that the firm are now able to turn out larger quantities than ever, while the last Olympia show was evidence of the popularity of the Rubery Owen pressed steel back axle.

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CONTENTS.

THE NAVIGATION OF THE AIR.

ELSEWHERE in this issue there is a considerably abridged report of the most interesting paper on problems relating to air-craft, presented to the Institution of Automobile Engineers on March 8th by Mr. Mervyn O'Gorman, the superintendent of the War Office Balloon Factory. It would be possible to criticise the paper by saying that there was little fresh information in it, but this fact does not detract from its value as a summary of the more immediate problems of air navigation by means of machines similar in broad principle to those which are now in existence. More particularly interesting perhaps, were the expression of the author's opinions as to probable developments of the aeroplane in the near future. In reading the paper it must not be forgotten, for a moment even, that the author is regarding the subject from the military point of view. He is concerned almost exclusively with the aeroplane as a weapon; hence his insistence on the troubles of landing on rough ground. To this extent there was an inconsistency in the paper, because Mr. O'Gorman voiced his belief in the theory that the safety of flight lies in speed, and speed in itself necessitates the provision of smooth alighting grounds quite free from all obstructions. Obviously the high speed machine would be of little service as an arm to a force moving over broken country, if it is relied upon for assistance in Europe only, dependence upon special landing places must reduce its value very greatly indeed.

It was doubtless owing to these facts that, in his introductory remarks, Mr. O'Gorman hinted at the possible evolution of a machine with the qualities of the aeroplane and the dirigible balloon combined. A machine which could rise from or land on any ground in a vertical direction and acquire horizontal velocity only when off the earth, while possessing the speed of the aeroplane and its compactness as compared with the balloon. Considering the rapid advance in aeronautical knowledge, it would certainly be unsafe to predict that either the aeroplane or the balloon will prove to be typical of future flying machines. It is entirely possible, though perhaps not too probable, that some totally different form may yet appear. Still, to leave such surmise quite out of the question, it seems unsound to base argument on the future provision of what have been called "aeroplane ports," or large spaces of land cleared specially. If the aeroplane comes to be used for the regular transport of passengers or goods between towns, then stations or "ports" would doubtless appear, because they would be a convenience, but the one man machine, or even the machine to carry half a dozen people will be restricted in scope to an enormous extent if it cannot rise and alight practically anywhere

It appears that Mr. O'Gorman thinks the machine with variable wing area, so as to be able to expand and fly slowly when near the earth, is not practicable because the atmospheric disturbances are at their greatest close to the ground, but seeing what can be done even to-day in the way of slow-speed flying, it must surely be possible to improve landing methods without increasing the speed of landing. Speaking from a mechanical point of view only, the landing chassis commonly used are of a most crude description. The skids have an extremely small area of contact with the ground, meaning that the load on them per square foot of area is very great. The wheels, when wheels are used, are very small, and the rubber springs employed are mostly of such a nature as to produce the maximum of bouncing. It should be possible to improve landing chassis to a very great extent indeed and still decrease their head resistance.

It is also to be regretted that space could not be found—or that the author for some other reason was unable—to deal with the materials of aeroplane construction. The Superintendent of the War Office Balloon Factory could give some extremely instructive particulars regarding the possibilities of metal for framework, and for most parts for which wood is now usually employed. Working in sheet metal is an old art, but it is also an entirely new science, and there are extremely few men who have any experience of working thin sheet steel as what might be called a "proportioned" material. It seems reasonable to believe that an aeroplane made entirely of metal would have less head resistance than a wood and wire structure, because a large steel tube made to a streamline form would cause considerably less atmospheric disturbance than a correspondingly strong wire-trussed timber framework. A metal wing surface could also be made to be accurate to calculated forms within one per cent. at most, while it is often doubtful if a canvas wing is accurate to fifty per cent. over the whole of its area. By the substitution of steel for wood either of two ends could be attained, machines could be made far stronger weight for weight and resistance for resistance, or they could be equally strong with much less weight and much less resistance. Of course, steel is slowly asserting its superiority for aeroplane work, as can be seen by anyone who has kept in touch with aeronautical construction for the past year or two, but it is still a very long way off from supremacy.

It is only when organised engineers commence to tackle aeroplane building as a serious business that we may expect to see rapid improvement in construction. Improvement in design

could not very well have been more rapid as things are. In such factories as that owned by our own war department we have engineering skill of a high order of merit brought to bear not so much upon the creation of new types of flying machines as upon the best possible making of existing types. It needs but little observation of aeroplanes to show that much could be done by taking a machine which is known to be a successful flier, leaving it entirely unaltered as regards general arrangement, etc., but re-designing the detail, and changing the material so as to build it lighter, stronger, and more accurately. That this could be done there is no doubt at all; that it is now being done quietly by several famous engineers is certain fact. Sufficient is now known of the aeroplane for it to be safe to say that its ease of handling and stability as regards controls depends to a very large extent upon the delicacy and ease of movement of the rudder, the elevator and the ailerons or their equivalent. Better construction must result in vast improvements in this respect, so making the art of flight less difficult to learn, and more easy to practice in disturbed air.

SPECIALISATION IN MANUFACTURE.

LAST month the question of the elaboration of automobile manufacture was discussed in these columns, and it is now proposed to give some consideration to a subject of a kindred nature, though of a somewhat less wide application. Last month it was pointed out that, if an automobile manufacturer proposes to produce not only private cars, but heavy vehicles and aeronautical machines as well, he is most likely to be successful if he sub-divides his business and treats each of these three branches separately, both from a factory and from a business point of view. If such sub-division is truly in accordance with economic principles it may be asked whether the specialisation of manufacture might not advantageously be carried still further by the limitation of any one factory to the production of not only one broad class, but to merely one or two types from that class.

It is an accepted fact that the Continental makers usually catalogue a large number of different models of car chassis, that British makers come next in order of individual variety of product, and that the American producers far more often confine their attention to one or two chassis alone. Which is the best practice? To answer this question satisfactorily it is necessary to examine it from several different points of view, and before doing so it may be interesting to give some figures showing something approximating fairly closely to the average number of types of chassis per manufacturer on the Continent, in Great Britain, and in America. Taking fifty representative makers from each of the three (the first class including France, Germany and Italy) it may be seen that the average number of types per head is :—The Continent 4.46, Great Britain 3.25, America 2.01. Amongst these makers some certain figures are :—Continental : One model 2, two models 3, three models 8, four models 5, five models 10, six models 3, seven models 3, eight models 6, nine models 1, ten models 1. British : One model 4, two models 5, three models 3, four models 9, five models 6, six models 1, seven models 2 ; while four is the largest number of types made by any of the American manufacturers.

Doubtless the averages are to some extent misleading, because many of the firms who catalogue a large number of types have simply added new models year by year without removing old types. Very many large cars are precisely the same to-day as they were three or four years ago, and it is to be presumed that only quite small numbers of many of the big cars listed are ever sold, or even asked for. The makers have the patterns and the jigs for their production and so could make them profitably if there was a demand. Therefore they remain in the catalogue, but they are more imaginable than real.

Thus if it was possible to obtain data concerning the number of cars which each of the makers is now actually manufacturing it would probably be found that the Continental and British average would fall very greatly, and would therefore approximate more closely to the American average, which is probably a true indication of the actual state of affairs in the United States. It is greatly to be doubted whether any manufacturer is selling quantities of more than two or three models. Makers who profess to supply six or more types are usually those with large resources and with organisations of such a size as to be exceptional for Europe. Some of the very large concerns in America which specialise on one model may also be ruled out because they likewise are exceptional, and owe their existence to an enormous and ravenous market. It is more than likely that some of these latter will find themselves in difficulties as the excess of the demand over supply dies its natural death.

This reduces the matter to a consideration of whether a factory of normal size is best employed upon the making of one chassis only, or whether it is better to have three or four types to strengthen the position of the sales department, but it is not necessary to regard the case from the commercial standpoint alone. Let it be assumed that the ambition of a manufacturer is to make the best possible cars at the lowest possible works cost, consistent with quality. Then, if the market is assured, it must be best to specialise on one chassis, because the larger the number of similar parts that are required the smaller will be the cost per piece while, if the designing staff have the behaviour of only one chassis to observe, they can be reduced in number without danger of affecting improvement from year to year. In this country it is possible to point to firms who have given the bulk of their attention to either large, medium, or small cars, but to very few who have never made more than one at a time, and this is easy to understand because if the principal product is a large car the maker does not want to feel that he is cutting himself off from all share in the trade with smaller vehicles, while if it is a small car that is turned out in the greatest quantities, then a maker likes to be able to offer something more pretentious to those of his old customers who require more power or more body room.

If, however, it is allowed that the maker with many models appeals to a wider class of customers than the maker with the few, and if this commercial argument only is considered, it is easy to elaborate a very strong case for multiplicity of products. On the other hand, if the quality of the chassis is made of paramount importance, as it should be when a manufacturer desires to build up or maintain a high reputation, it must be remembered that some of the most successful cars of recent years have come from factories where only one or two models were made, while many manufacturers have made their name by the superexcellence on one type over the average of their productions.

The large car, the car of medium size and the small car, all require special experience for their successful design, and it is hardly to be expected that any one man can study more than two types at most—and keep on improving upon them. Thus it is reasonable to assume that the smaller a firm and the smaller its staff, the less should be the number of types of car turned out. It is also perhaps well to remember that a one-model specialisation is not possible strictly, if varying gear ratios and wheelbases are supplied ; and it is both reasonable and right that this should be. Generally speaking the best chassis are those which are more or less specialities of their makers, but a really large firm can, at present, hardly expect to find enough demand to justify a single model programme.

THE PURPOSES OF CHAIN DRIVING.

With special reference to camshaft drives and chain driven gearboxes.

By J. R. Cantey.

THE use of driving chains on motor vehicles has always been a question arousing discussion among automobile designers, and, in spite of the controversies which have been carried on through the press on this subject, there are many points of interest to designers which are not well known. It is these points which it is proposed to deal with in this article, leaving the controversial matters alone as much as possible.

At present there are three types of application for the chain drive on motor vehicles, and in these both the roller and silent type of chain has been used. They are the final drive to the road wheels, the gearbox drive and the engine auxiliary drives : the last including camshafts, magneto, fan and pump drives. In the final drive the roller chain is apparently out of fashion on the private car and is having to fight to maintain its place on light traction vehicles. With the heavier commercial vehicles, the three to five ton petrol or steam lorries, the chain seems to be holding its own. There are many

reasons for this, the most important being—

(1) The saving of unsprung weight on the road wheels.

(2) The possibility of changing gear ratios to suit different conditions with small expense.

(3) The chain holds its high efficiency for a great proportion of its life. (This is true both of the roller and silent chains).

(4) The efficiency of the chain drive may be higher, under road conditions, than any other form of final drive. This is on account of its ability to stand mal-treatment of all sorts (especially that of bad alignment), with little loss of efficiency.

A point which seems to have been neglected by the controversialists, and one which is in many ways most important, is that the braking effect on the car puts more stress on the final drive than the driving effort. If designers had in the early days gone into the question of front wheel brakes or of all brakes on the back wheels there would have been less talk of the rapid wear of chains and the day of the chain might have been lengthened. It may be said that this point affects all forms of drive and in the United States, where the chain is as rare as in England and where brakes are of considerably more importance, a number of makers have for some years placed all their brakes on the back wheels. In a list of 1910 models of petrol cars, out of 212 manufacturers in the States all but 39 pinned their faith to brakes on the back wheel, while 7 out of the 39 had some models with one form and some with the other. This should dispose of the idea that back wheel brakes are insufficiently powerful.

The above may be points of purely academic interest to the private car designer, but they should certainly be considered by the designer of commercial vehicles and of very light cars where the maximum advantage must be taken of engine power. An idea which should be thoroughly investigated by designers of the last named vehicle is that which has been in use for some years by the makers of the Phoenix car, *i.e.*, to do away with the necessity for all universal joints (which if properly made are expensive in first cost) by means of an engine set crosswise in the frame and a chain drive from the engine through the gearbox to the differential back axle.

It would be easy to prove that a higher back wheel efficiency could be obtained on this type of car with an engine of given horse power and a given efficiency of gearbox than on any shaft driven car, taking, of course, an average of the two extreme conditions, one with the propeller shaft parallel to the crank shaft and the other with the maximum angle possible on the road. Also in the matter of silence a car thus driven with silent chains can only be surpassed by one which is driven by worm gearing.

The first chain-driven gearbox known to the writer was patented in 1898 for the use of silent chains on change speed mechanism for lathes, motor cars, etc. This has never been used to any extent, but change speed drives of similar type have been in use on lathes in various parts of the country for many years.

The first chain transmission in a gear box to come under the writer's notice was an American gear known as the Haynes-Apperson, constructed by one of the earliest automobile builders in the country. This vehicle used one or two chains of the Morse type in its gearbox, and was in use as early as 1904. There have been others using either roller or silent chains in this country; notably the Brooke box, with a roller chain. Later on there was the well-known attempt of the All British Car Co. to make what was an "all chain-driven motor bus." This was not exactly a chain-driven gearbox, but was a transmission with the changes of speed actuated almost entirely by silent chains. A chain driven change speed gear has been in use for the past three years on certain French petrol lorries, and this has given entire satisfaction, but it has not been taken up as extensively as it might be on account of certain structural difficulties in the vehicle itself. The latest practical development of the chain-driven gearbox seems to be that first actually used by the London General Omnibus Co. some time early last year. This box, as is well known, is in essence the well known constant mesh type gear box with the constant mesh gears separated and chains and chain wheels substituted for gears on the lay-shaft and forward speed drives, a spur gear still being used for the reverse. This idea is one which has been under discussion for some years, but makers have been deterred from taking it up, by fears of expense and trouble, until they were forced to do so by the police regulations in London.

The success of this box is to a large extent problematic as it has not yet been running long enough for an authoritative statement to be made. It should be noted however, that the conditions under which these boxes are running are exceptional as they are in the hands of skilled drivers and mechanics. Therefore, even though they are a success under these circumstances, it has not been proved that they would be suitable for traction vehicles generally, as these are often in unskilled hands and under no supervision.

It is, of course, well known that the top speed is in use for most of the running time, say from 75 to 90%, depending on power and service and type of vehicle. A thorough knowledge of chain driving would teach the fact that considerable wear takes place in a short chain when the chain is running under no load. For this reason it is very desirable to arrange to stop the lay-shaft when the vehicle is driven direct. It may be added that provisional protection has been applied for on an improvement covering the stopping of the lay-shaft, and pointing out as a further advantage that ball bearings on the lay-shaft will in this case not be so necessary as they would ordinarily, thus simplifying design and lessening expense. A further improvement, which has also been covered, is that of arranging for a certain amount of adjustment of the chains, say by allowing the lay-shaft to be mounted on a set of eccentric bushes. This would necessitate a pair of idlers in the reverse gear, but the extra expense would not be great, the main difficulty would be properly to keep the bearings in alignment, but on a lay-shaft with plain bearings the difficulty should not be insurmountable.

Leaving the question of gearbox design improvements the other points are those which affect any chain drive and these are :—

1. To obtain such a centre distance that three or four drives with different ratios may all join up without excessive slack.
2. To fix proper ratios of reduction and still conform with the first requirement.
3. To arrange for pinions under 23 teeth to have an odd number of teeth in order to obtain maximum service from the faces of the wheel teeth.
4. To eliminate the cranked link, as it is impossible to design it to be quite as reliable as the rest of the chain.

The reason for (3) is simply that with an even wheel and even number of pinion teeth the same combination of links always makes contact with any one same tooth and therefore only half of the tooth face is in work. The question of the chain-driven gearbox is really one of silence and the necessity for such a box depends on how acute the desire for silence becomes on vehicles where a fair amount of gear changing is necessary. Mileages are as yet unknown and will depend not only on the size of the chains used, but on attention paid to them, on the clutch used, and the quality of the driving. The main drawbacks to the box are its expense and the possibility of chain breakage without warning by the growth of undue noise. The first difficulty is not so great as it seems in these days of careful running-in of gearbox gears. The second is born of the great advantage in silence and efficiency owned by the chain drive, and it is a fact that until a chain begins to jump the teeth of the wheels the driver would have little or no previous warning either from loss of power or increase of noise. It is however, desirable to so design the boxes that if the chain breaks it may drop into a sufficiently deep sump so that there is no possibility of the wheels and chain fouling and breaking up the casing.

Camshafts and Auxiliary Drives.

It may be noted that such auxiliaries as the fan, the pumps and the magneto have been driven successfully for some years with light chains both of the roller and silent type. These drives will therefore only be discussed in connection with the newer application of the camshaft drive. The magneto driving by roller chains, notably on motor bicycles, has been wholly satisfactory, and has given no trouble from noise, but fan and pump drives have been largely special applications. The question of camshaft drives which is interesting designers at the present time, arose from the method of driving adopted by the inventor of a well known engine in which the valve gears are driven by a number of eccentrics and not by cams. In discussing camshaft drives designers appear in many cases to have overlooked the fact that driving the camshaft on the standard type of poppet valve engine is a different proposition to driving the eccentric shaft on this other type. If the designer were to lay out an effort diagram

of the two he would soon see this, especially where the tappet valve engine is of the two camshaft type, as in this case he would be obliged to compare one camshaft with the eccentric shaft driving eight or twelve eccentrics. For this reason the eccentric shaft engine will not be discussed, as it has really no bearing on the matter. Engine auxiliary driving by means of chains is by no means new and the steam engine governor, which requires most delicate regulation, is fitted with chains as standard practice by firms of high standing, while one of the large Lancashire steam engine makers has had a large vertical engine with chain driven Corliss valves in service for twelve years. Of course further examples might easily be mentioned and a number of gas engine applications of various sizes have been made.

In the past three years many camshaft drives have come under the writer's observation. Experiments have been made, some of them successful and some not, success depending on whether the designer was willing to make some allowance for the limitations of the chain in order to obtain its advantages, or whether he simply substituted a chain and pair of wheels for gear wheels. It will be seen from the discussion following that this drive is not in all ways ideal, but as an example of satisfactory service a particular case may perhaps be quoted from an article in "*La Vie Automobile*," by Mr. Charles Faroux.

"Referring to the chain drives for the camshafts, I have said that with this system there was another advantage besides silence. As a matter of fact on the Gregoire motor it has been found that the power obtained through this drive is considerably increased."

The article goes on to describe the action of the camshaft when driven by spur gears, with their necessary backlash, and contrasts with this the considerably more smooth running of the chain drive, coming finally to the conclusion that the chain drive is an advantage, not only from the point of view of silence, but of even flow of gas through the induction and exhaust pipes due to less chattering, and although this is doubtless an exaggerated view, it is very likely based on fact.

Having thus discussed camshaft drives in general terms it is advisable to go into the matter point by point. It is well known that the reason for taking up chain drives has been a desire for greater silence, and on the question of silence it may be said at once that the silent chain has shown itself to be an improvement on the roller pattern. On the other hand designers feel hampered by questions of expense and maintenance cost and are therefore desirous of using a chain drive in its simplest form, though as a matter of fact expense and maintenance cost are not so serious as might be anticipated: the crux of the matter lies in the care taken in designing the gear. The designer who goes furthest in recognising the limitations of the chain will be the one to reap its greatest advantages and for his benefit it will be well to point out both the advantages and limitations. Taking the advantages first—

(1) Silence is maintained throughout the life of the chain and one extremely potent noise is thus eliminated.

(2) The writer has been engaged for some time in working on efficiency tests of the silent chain and has been hampered by the fact that the efficiency through reasonable ranges of power, speed and wear is so high that accurate measurement is most difficult—these ranges of variation lying entirely above 96%. Bearing on this question the U.S. Agricultural Dept. Report of 1901, Bulletin No. 98, gives certain accurate tests on chain drives as applied to the bicycle. These tests show an efficiency for the bicycle chain up to 99.5%, and the writer has recently obtained further verification of the accuracy of these tests from the maker of them.

(3) For maximum efficiency the centre distance is not of so great importance as it is with gears. Accuracy in thousandths is necessary for gears to mesh properly, whereas with a chain a few hundredths of slack does not affect the efficiency.

(4) There will be a probable increase in the life of the valve gear apart from the chain. By the principle of the silent chain there is not only multiple tooth engagement, but the engagement of each tooth is without shock or sliding motion. Therefore, except for the imperceptible variation in speed owing to the polygonal motion, the chain may be regarded as being as smooth running as a belt. Chain drives have been substituted in many cases for short centred belts or for gear drives from electric motors. In both these cases the chain has effected a marked improvement in commutation, in fact bad cases of flats on the commutator have been cured by this means. In the second place the silent chain has a certain amount of flexibility which tends to smooth out shocks received from the driver before it is transmitted to the driven wheel, and this is aided, in a more or less horizontal drive, by the weight of the chain, though this latter effect is imperceptible on short centres.

From the foregoing it seems reasonable to suggest that the whole of the camshaft gear will wear better with chain drive than if the shocks due to the cyclic fluctuations in the crank shaft were transmitted direct to the camshaft, as is the case with spur gears. On certain cars the flywheel has been reduced to a degree which permits considerable cyclic variation to take place, and such variation is extremely severe on valve gear. In such cases the chain would tend to wear rather rapidly, but its replacement would be inexpensive and the saving on the rest of the gear would more than compensate for the outlay. Chain wheels will last much longer than the chain and if made symmetrical can be reversed, whereby their life is doubled without loss of efficiency or silence.

(5) The manufacturing cost is not serious, although the chain and wheels might seem likely to cost more than spur wheels. Thisi is hardly true, for chain wheels simply need accurate cutting and the chain is not excessively expensive. On the other hand with spur gears special pairing and running in with oil and emery is the rule rather than the exception. Also these gears are often cut on gear shaping machines in order to obtain correct tooth forms, while with the silent chain wheels the theoretically correct tooth form is obtainable from a standard rotary gear cutter.

(6) The maintenance cost depends on the life of the chain, but, as the replacement of a chain needs no special fitting and the chain itself is a small item, this point is not a serious one.

Having put forward such a formidable array of advantages, even though several are admittedly problematic, the limitations and possible disadvantages must be examined and these are principally—

(1) If the cam and crankshaft wheels are solid it is more difficult to adjust the relative positions of these shafts on account of the coarse pitch of the chain wheels (the smallest being ½in.), as the movement of one tooth makes a considerable difference in the angular position of the camshaft.

(2) Although the silent chain rises up its teeth and adjusts itself on the wheels for increase of pitch it cannot take up the increased pitch in the portion of chain between the wheels, wherefore the wheel must lag to the extent of the increase in pitch of the straight portion of the chain. For a given straight length of chain the angularity of this lag grows less as the size of the wheel grows larger; therefore it is advisable to use as large a wheel as possible. Fig. III. will perhaps make this more clear.

(3) The abnormal size of the chain necessary is not an advantage but it is more apparent than real, because the life of the chain is dependent on the area and number of its bearing surfaces. When the length of the chain is much below and the speed is much above the normal, as in the case of a camshaft drive, the bearing area must be increased by increasing the width of the chain.

(4) The life of the chain may be short, although this has not yet been proved, but the life will always depend largely on how far the designer will go to meet the limitations of the chain, and even if the life is short it may be much more than compensated for by other advantages.

Design of Drives.

Having set forth the points pro and con the next thing is to examine the matter with a view to weighing up the advantages and disadvantages. Commercial considerations may prevent the use of some of these proposals, but they are worthy of study.

In the foregoing remarks on camshaft driving it has been assumed that ½in.

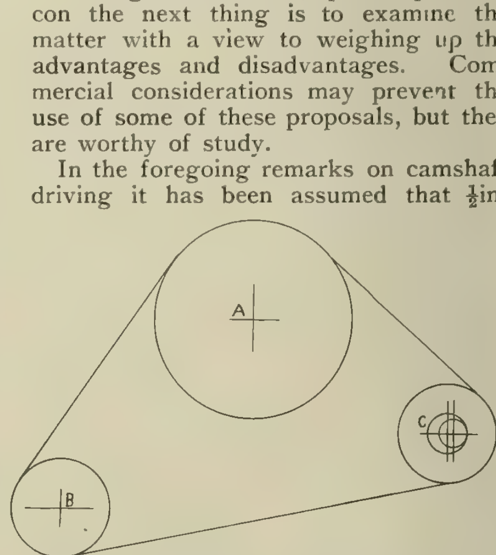


Fig. I.—A camshaft. B crankshaft. C magneto

pitch chain will be used, because

(1) The power to be transmitted is small.

(2) The linear speed will necessarily be high.

(3) Silence is the most important factor.

and to reduce noise a large number of teeth should be used in the pinion. On the other hand the linear speed should be kept down as far as possible and to do this a small pitch chain should be chosen.

(4) The ½in. pitch is the smallest which is at present commercially satisfactory for the purpose, though on

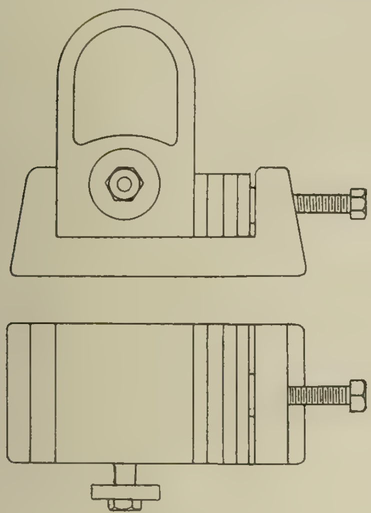


Fig. II.

exceptionally slow speed engines, such as certain traction vehicles, ⅝in. pitch chain may be used.

The proper width of chain depends entirely on the conditions of bore, stroke, speed and adjustment, and also on the chain length and number of wheels over which the chain is to run. Also an odd number of pitches should be avoided, for it is impossible to make a cranked link quite as reliable as a straight one.

In order to obtain the maximum life from a chain, together with the best of running, adjustment should be provided. In so short a drive a small amount of slack will take away from the smoothness of the drive and tend finally to produce snatching and possible breakage of the chain. In some cases the slack may reduce the number of teeth in engagement on the pinion and thus accentuate the trouble. Adjustment m usually be provided by an eccentric bush, or by some other well known method, and by arranging that the camshaft drive shall be triangular with the adjustable wheel on the magneto shaft. An arrangement of this sort is shown in Fig. I., and Fig. II. shows a possible method of allowing for adjustment which has the advantage of being self-aligning.

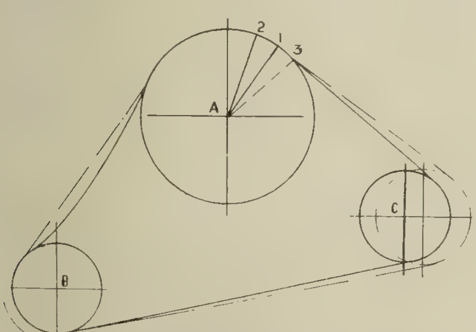


Fig. III.—A camshaft. B crankshaft. C magneto.

Though an eccentric bush may be used it is difficult to arrange for sufficient adjustment and still keep the magneto nearly enough in line with its driving shaft, 2 mm or 3 mm of adjustment being hardly sufficient. Eccentric adjustment does not however in any way re-set the timing to its original position, in fact it may be seen from Fig. IV. that as slack accumulates the chain will allow the camshaft to lag behind the original timing point. When adjustment for slackness is made in the form of drive shown in Fig. III. the effect of taking up the slack is to give the camshaft lead. Both these points will be fairly evident from the diagrams. If the direction of rotation on Fig. III. be changed the conditions are somewhat improved, as with the rotation in a clockwise direction the main driving effort is transmitted straight from the driver to driven wheel, and not around the magneto wheel. One way in which the first mentioned form of drive is perhaps advantageous is that with contra clock-wise rotation adjustment does not reduce the lag and bring the valves to a leading position for the time being, in other words, the first method allows the wear to take place on both sides of the correct position, whereas with contra-clockwise rotation the lag is the same whether the chain is adjusted or unadjusted. The next most important consideration is that of adjusting the camshaft wheel for the lag mentioned, and also for slightly varying the timing of the valves. This may be done in several ways, two of the simplest being shown in Figs. V. and VI. In both cases the camshaft wheel is a loose ring on a flanged boss, and in the

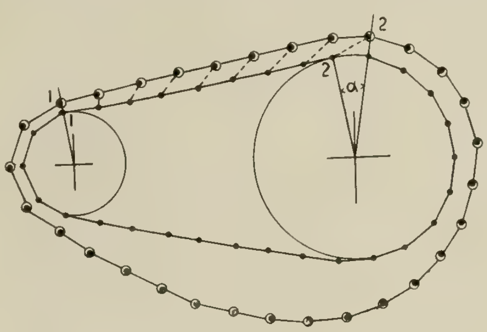


Fig. IV.

first case the method of adjustment is evident, while in the second case the holes in the ring and flange are drilled vernier fashion, being, say, ten in the flange and eleven in the ring. This could be duplicated as shown in order to make a more mechanical job and the adjustment could be very fine. Other simple and cheap methods could no doubt be found.

Leaving the two main considerations, the next in importance is that of wheel size. In most cases the wheels should be as large as possible for several reasons, the first is to minimise the lag effect, the second is to decrease the motion of rotation of each chain bearing as the chain comes into gear, the third is to reduce the pull on the chain and the fourth to reduce noise, as already noted. For usual practice it is advisable to use 21 and 42 teeth, though pinions down to 17 teeth may be used under certain circumstances where other considerations become more important than that of obtaining the best conditions possible for a chain drive. An odd number of teeth should be used in the pinion in order to get maximum life from the wheel tooth faces. Caution is necessary, not on account of gearing difficulties at high speed, but on account of the centrifugal tension set up in the chain at higher speeds, this tension increasing as the square of the linear velocity of the chain and being a very large portion of the total load at speeds of 2,000 ft. per minute and over. Steel wheels should in all cases be used, and these wheels may be hardened to give the maximum of life,

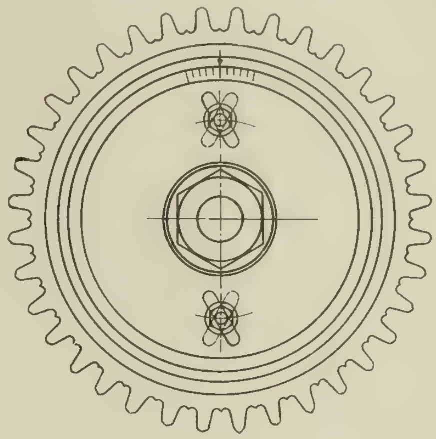


Fig. V.

the noise being so small that it is not worth while to keep the wheels unhardened on this account. There is no purpose in a large wheel of phosphor bronze, as all noise emanates from the pinion.

Jockey pulleys against the back of the chain are not to be recommended, because they are liable to burr up the backs of the chain links and cause serious trouble, though a soft leather-faced pulley might be used for experimental purposes. If a chain wheel jockey is found necessary (the magneto being driven by other means) care should be taken that at least three teeth are in proper engagement with the jockey, as a silent chain wheel cannot be made to gear with a silent chain running as a rack. Triangular drives should be avoided where there is no adjustment, as the arc of contact is necessarily smaller to start with than an ordinary drive, and the trouble arising in a two-wheel drive would simply be accentuated in such a case. Thus if it is desired to drive two camshafts without adjustment two separate drives should be used, and if a non-adjustable magneto is to be driven it would be advisable to drive it separately either from the camshaft or direct from the engine shaft.

Turning to the question of chain drives for overhead valve gear; these are not likely to be successful if the drive is

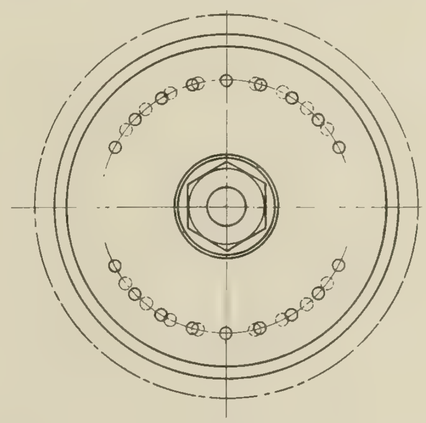


Fig. VI.

taken directly from the engine shaft to the camshaft, as the distance between the two is so great and in consequence a very small amount of wear would allow

the camshaft to lag considerably. Also this same small amount of wear would, especially when starting up, tend to make the chain mal-gear with the pinion, accentuating bad timing and perhaps causing the chain to spring the shaft or miss a tooth and spoil the timing. The only feasible method for driving an overhead camshaft seems to be one similar to that in use on the Germain car, which has an intermediate shaft with adjustment, which shaft may conveniently drive the magneto. In long drives of this sort a considerable amount of adjustment is necessary, and an eccentric bush would hardly provide sufficient latitude, especially as there are two chains to be adjusted. If ample adjustment is provided for the two drives at such an angle that if their wear be equal they will both adjust equally, there will be a very small amount of lag and, except for adjusting the timing, it is quite probable that a solid camshaft wheel might be used.

In conclusion, it may be said that for experimental work almost any size of wheels and chains may be used, but it is not to be expected that the results obtained will be a criterion of those obtainable from carefully designed gears.

THE BALANCING OF SHAFTS AND FLYWHEELS.

An elementary account of the differences between static and rotational balancing of mechanisms, and a description of some machines for correcting errors in balance.

It has several times been remarked in these columns that the most noticeable imperfection of the present day automobile engine is the vibration to which it is liable at high speeds and the matter is of ever increasing importance, because rates of revolution have risen so greatly and are still rising. It is not long since fifteen hundred revolutions a minute was regarded as a very high speed indeed, but to-day many standard 80mm. engines run at two thousand as a regular thing, and it is to be anticipated that even three thousand revolutions may be a common speed in the course of the next year or two. This being so, every effort ought to be made to obtain improvement in smoothness of running, and it may be useful to catalogue the methods usually adopted towards this end.

Balance of Reciprocating Parts.

Pistons and connecting rods are now almost invariably weighed against each other, or against a standard weight, so that the set for one engine shall only differ by some fraction of an ounce.

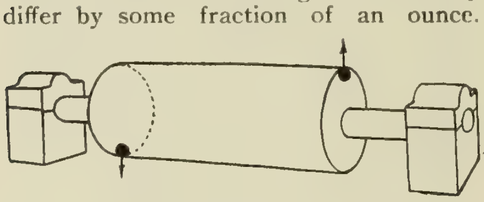


Fig. I.

Further, the ambition to obtain high revolution speeds has led to the reduction of reciprocating weight, and it is certain that one or two manufacturers have produced pistons in which the limit of possible lightness has been reached. While the average weight is still far in excess of these special cases, it is not likely that much improvement in balance can still be made by researches in this direction.

Undoubtedly (from the point of view of reciprocating balance only) the four cylinder vertical engine is inherently imperfect, and either the six cylinder vertical, the Vee engine, or an engine with two crankshafts rotating in opposite directions (like the Valveless) is much to be preferred. Each of these constructions has disadvantages peculiar to itself, of course, but it is certainly within the bounds of possibility that the four cylinder will become less common because of this imperfection.

Mention of the six cylinder leads to the next cause of vibration which is crankshaft oscillations, and these were discussed fully in the January issue. Bending or sagging of the crankshaft causes vibration by disturbing the normal accelerations of the pistons and so upsetting the normal balance, but this is an entirely mechanical difficulty curable by the use of a very stiff shaft or by various other means which have been published from time to time.

Rotational Balance.

All argument concerning the balance of reciprocations is based on the assumption that the flywheel and crankshaft of an engine would run with perfect smoothness if the connecting rods were removed and the shaft rotated by some steady form of drive. If it is assumed that the workmanship is perfect and if the crank has the throws properly balanced, then smooth running under the conditions stated above means that the materials of which both shaft and flywheel are made are perfectly homogeneous, and it is common knowledge that this is never the case. To overcome troubles arising from lack of homogenity it is usual to test rotating parts between dead centres to see whether they will stand in any position in which they are placed. Thus with a flywheel it may be found that one point always comes to the bottom and then it is necessary either to cut away metal at this point or to add more at the opposite side. Similarly with a crankshaft it may be necessary to grind away a little from one or other of the throws or to drill out a little from a pin.

Even when this is done however, the parts have been balanced only with respect to one line, which is the centre line of the shaft. That this is insufficient for the removal of all possibility of vibration can be seen by Fig. I., which shows a cylinder carried on a shaft running in a bearing at each end. Suppose that the two black spots are extra heavy parts but of equal weight. Then the

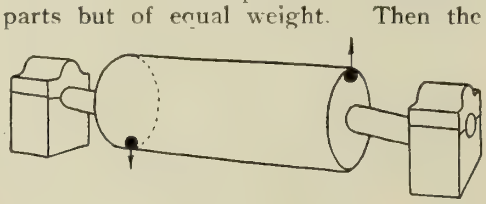


Fig. II.

cylinder would be in perfect standing balance and mounting between dead centres would not disclose the presence of the heavy spots. Let the cylinder be rotated, however, and each spot acquires a centripetal acceleration, the result being a load on the bearings rotating at the same speed as the shaft. Now the load caused by either spot will be in a direction opposite to that caused by the other, so there must be a twisting action on the whole cylinder. Supposing that the shaft was light and the bearings rigid, this would tend to cause distortion of the type shown diagramatically in Fig. II. To render the rotational balance correct it would therefore be necessary to drill out the heavy spots or to balance them separately by means of adding weights in the same vertical planes as the spots and at such radii as to create a force equal and opposite to that caused by the spot. Putting the matter in other words, it amounts to the fact that in the case of a cylinder as described above, if it is to be in rotational balance, every part of it must be in standing balance. Or, if we imagine the cylinder to be cut up into an infinite number of thin discs each disc must be in perfect standing balance.

To consider a slightly more complicated case than that shown in Fig. I., we may imagine that indicated in Fig. III., where there are four equal masses distributed proportionally on opposite sides of the cylinder. Considering firstly A and B alone. When rotation commences A and B endeavour to so twist the cylinder that they shall lie in

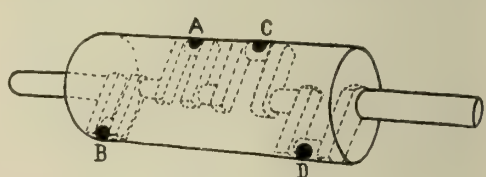


Fig. III.

the same plane at right angles to the shaft. C and D must obviously do likewise so the cylinder tries to take the form indicated in Fig. IV. (Incidentally this is exactly the condition of working of a four throw crankshaft with only two bearings). Now if the cylinder is stiff it will be able to resist the tendency to bend and there will therefore be no general disturbance of the balance of the piece as a whole because if there is sufficient stiffness the two couples caused by A B and B C balance each other. If, therefore, we have a crankshaft with a centre bearing or very great natural rigidity its rapid rotation will not give rise to vibration. Affairs can however be improved by using disc crank webs or by extending the rectangular webs as shown in Fig. V., for then the disturbing weights are the weights of the crank pins only, the webs being balanced in their own planes of rotation, and it would be possible to balance the pins as well by slightly expanding the ends of the web balancing pieces as indicated by dotting on the end view in Fig. V.

However, assuming that a shaft has sufficient stiffness to resist the bending

action of otherwise balanced couples, it may still be out of balance by reason of an unbalanced "heavy spot," and this can only be found out by experiment in a special machine. One of the best known machines is made by the Norton Manufacturing Company, and is handled in this country by Alfred Herbert, Ltd., to whom we are indebted for certain of

Fig. IV.

the particulars mentoined below. This machine, and the method of using, will be described later on, and we may now turn from the consideration of crankshafts to that of flywheels

Flywheel Balancing.

A flywheel is, of course, a cylinder rotating about its normal axis, and it is well known that even with small cast iron wheels much metal has often to be removed from one side before a standing balance can be obtained. Fig. VI. is a diagram of a wheel with three heavy spots A, B and C, which are assumed to be equal. If such a wheel was rigged up on dead centres, B and C would, of course, fall to the bottom, but in removing metal to obtain balance everything would depend upon whether B or C were cut away. If C was removed the wheel would then be in both static and rotational balance, but if the wheel was attacked at B the rotational balance would be much worse after standing balance had been obtained than it was before anything had been done. As a flywheel probably has dozens of heavy spots and light spots in its composition, it can easily be seen that the rotational balance might be very bad indeed without there being any possible means of ascertaining the fact except by trial with a proper apparatus.

Apart from the crankshaft and flywheel there are many other parts on a transmission gear which ought to be properly balanced, as for instance the clutch, because it is easy to see that an accurately balanced flywheel would be of little value if the clutch which it contains was badly out of balance. It is of no use to attempt to balance clutch and flywheel together, because a subsequent alteration of their relative positions would alter the condi-

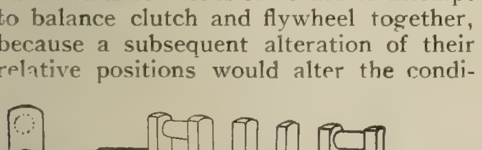


Fig. V.

tions, thus it is necessary to firstly balance the flywheel with all the clutch parts which it carries, and secondly to balance the male portion of the clutch by itself. A leather cone clutch is, of course, easily rendered correct by the addition of a few copper rivets, but a disc clutch is much less easy to deal with. Luckily the latter type is less likely to be out of balance in the first instance. Probably the most difficult type of clutch to deal with would be the internal expanding pattern, and the most easy the metal to

metal cone, but an instance of a particular engine which came under the notice of the writer may be given as emphasising the importance of this clutch balance. On a certain car great trouble was experienced from clashing in the camshaft driving gears, and it was observed that this was not noticeable except when the engine was in the car. Under load on the test stand it behaved quite satisfactorily, and after a great deal of unsuccessful experimenting, the trouble was found to be simply a matter of bad clutch balance, the clashing ceasing directly the clutch was altered and corrected.

Balance of Transmission.

Behind the engine, every shaft which runs at engine speed or nearly engine speed, ought to receive as much consideration as the crankshaft, and obviously the most important pieces are the main gear shaft, the layshaft and the propeller shaft. As regards the main shaft this in itself will usually be sufficiently true, but

cardan are of much less importance than the engine shafts and flywheel, because they seldom run at as high a speed as the engine. That is to say, the highest speeds of engine revolution usually occur when one of the lower gears is in use, and the major por-

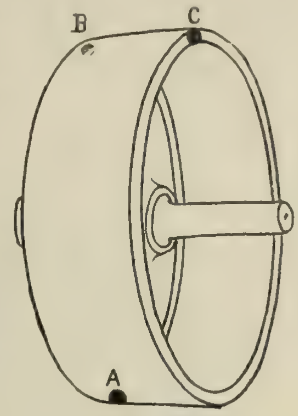


Fig. VI.

tion of the transmission is then moving comparatively slowly. In general therefore, it appears that it is advisable to balance carefully the crankshaft, the flywheel and clutch, and the front end of the main gear shaft set, while to balance

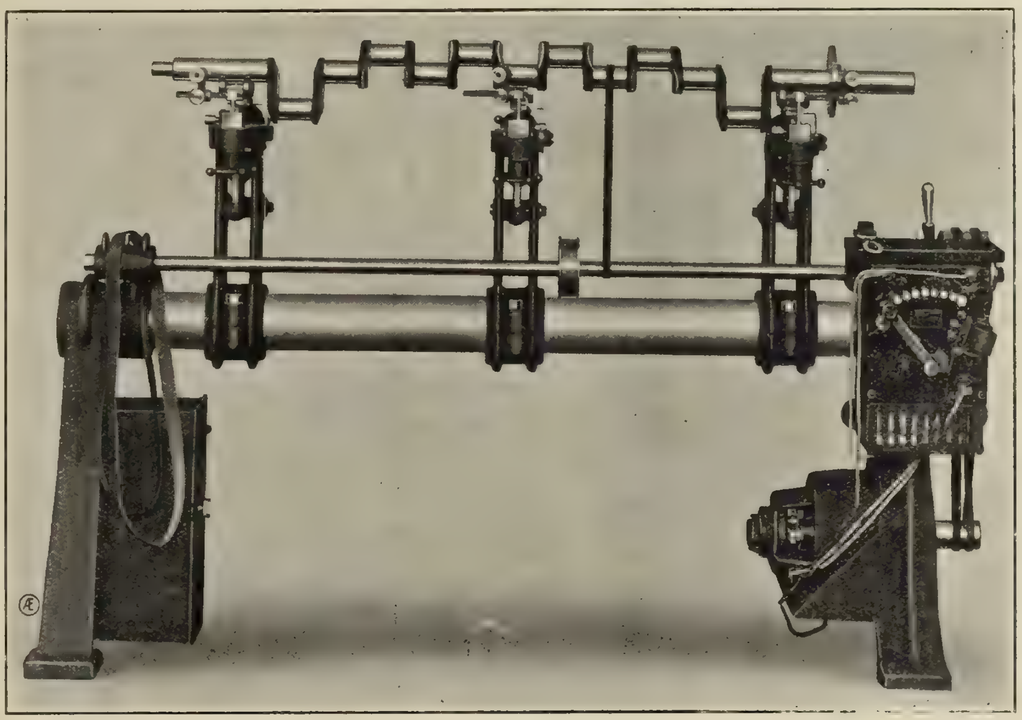


Fig. VII.

the gears which slide upon it may show considerable vibrative disturbance. It might be pointed out that it is not sufficient to balance the shaft, complete with gears in position for two reasons. Firstly, the gears are in different relative positions according to which is in use, and secondly, if a balance is obtained for both wheels taken together they must not be displaced radially afterwards. That is to say, if gears were so balanced and one of them was taken off the shaft and turned round partially before remounting, then the balance would be upset. For these reasons each mainshaft gear ought to be balanced separately on a standard shaft. For the layshaft, as the gears always retain their original relative positions this trouble does not arise, and the whole piece can be balanced at one operation.

To balance the propeller shaft is probably quite unnecessary, because the balancing of universal joints in all positions is practically impossible, and the vibratory stresses arising from flexion of the joints must be taken as an almost necessary evil of that form of transmission. However, both the gear shafts and the

the layshaft as well is perhaps worth while, because it is not difficult and it is certain that the better its balance the less noise is likely to be made by the layshaft drive, especially when the top or direct speed is in use and the layshaft is running light at a high speed.

A Running Balance Testing Machine.

Probably the best-known machine for testing and correcting rotational balance is the Norton, which we have already mentioned. One of these machines is now being used by Crossley Motors, Ltd., and we are indebted to them for many of the particulars of working which are given below. A general view of the tool is shown in Fig. VII., and it will be seen that it is very simple, consisting of a plain round bed with two sliding heads which can be locked in any position. The heads are simple hollow castings, and they are each provided with a rubber plug some four inches in diameter by two inches thick, which fits in a cylindrical socket, open top and bottom. The piece to be balanced rests on eight rollers (four at each end), and the roller carriages are at-

tached to a rod, which passes through a hole in one of the rubber plugs and rests in a cup at the lower end. This cup is observable in Figs. VIII. and IX., and is adjustable for height by means of the hand wheel on the opposite side of the head. It will be seen therefore, that the roller carriages are free to vibrate in any direction, being kept vertical only by the rubber. At

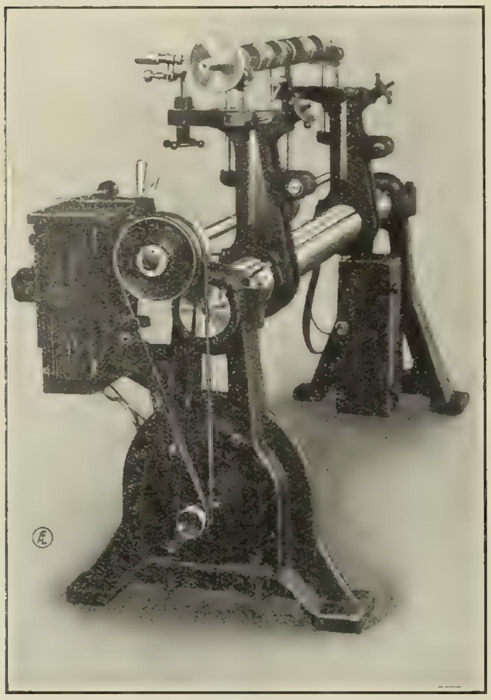


Fig. VIII.

the end of the bed there is a variable speed electric motor and a forward and reverse fast and loose belt gear, so that the test-piece can easily be revolved in either direction at a large number of speeds. The actual drive from countershaft to test-shaft is by another belt, the pulley for which can be slid along the countershaft to any desired position by means of gentle tapping with a block.

In Figs. VIII. and IX. the indicating needles are shown, and the actuation of these is as simple as is the rest of the apparatus. Attached to the vertical roller supporting spindles are a pair of arms, adjustable for height, and the indicating pointers rock on pivots fixed to the cast-iron heads. Each of the pointers is in reality a bell crank lever, of which the short end is kept in contact with the end of the arm on the corresponding supporting spindle by a light spring. Thus, as the spindle vibrates, the oscillation is magnified by the pointer, which quivers as indicated in Fig. VIII. by the blurred outlines seen against the sides of the heads.

The actual operation of putting a crankshaft into balance is conducted somewhat as follows :—

The shaft having been mounted is driven at approximately 500 r.p.m., and colour is then applied at the end where vibration (as indicated by the needle) is greatest. This is done by means of a roller supplied with the machine, and the position should be between the rollers of the carriage at that end. The scriber is then advanced until it just touches and, as soon as it has marked, is withdrawn, when the direction of rotation is changed by the reversing belt gear already mentioned. Then, as soon as the shaft has again acquired about the same speed of rotation as before, a second mark is made, and the heavy spot will be diametrically opposite the mean point between the two scriber marks. That is to say, the scriber marks the light side and not the heavy side. At very low speeds the heavy side may be marked, but above a certain speed it is always the light side, and change over in the nature of vibration can be observed in a badly balanced shaft by watching the pointers carefully. This may be explained by saying that if a part which is not in properly running balancing is revolving round its centre of mass it is obvious that the lighter side will run in a circle larger in diameter than will the heavy side. It is interesting to learn that in the experiments at the Crossley Works, the greatest difficulty with crankshafts has been owing to their whip. That is to say, it is impossible to run them at high speeds without bending, and directly bending takes place it becomes impossible to do anything further with regard to balancing. In order to guard against this, the Norton Company advise the use of as many supporting heads as there are main journals on the shaft, but the fact that whip does take place at comparatively low speeds of revolution, with the shaft running light on extremely free bearings, suggests that crankshafts might well be made of stronger section, because the Crossley cranks are already rather on the large than on the small side as compared with average dimensions.

Of course, once the scribing has been performed, the next stage is to cut away metal directly opposite the marked mean point, and to continue the tests until satisfactory steadiness is obtained. It is generally found that one end of a shaft is very much more out of balance than the other end, and in this case balancing the bad end will more often than not cure the whole shaft.

The extremely large amount of metal which can be removed from a crank in certain cases is really astonishing, in fact, it is so great that Mr. Reeves, who has conducted the Crossley experiments, says that this firm will probably cease to machine crankshaft webs, so that the work saved on the webs may be put into correcting

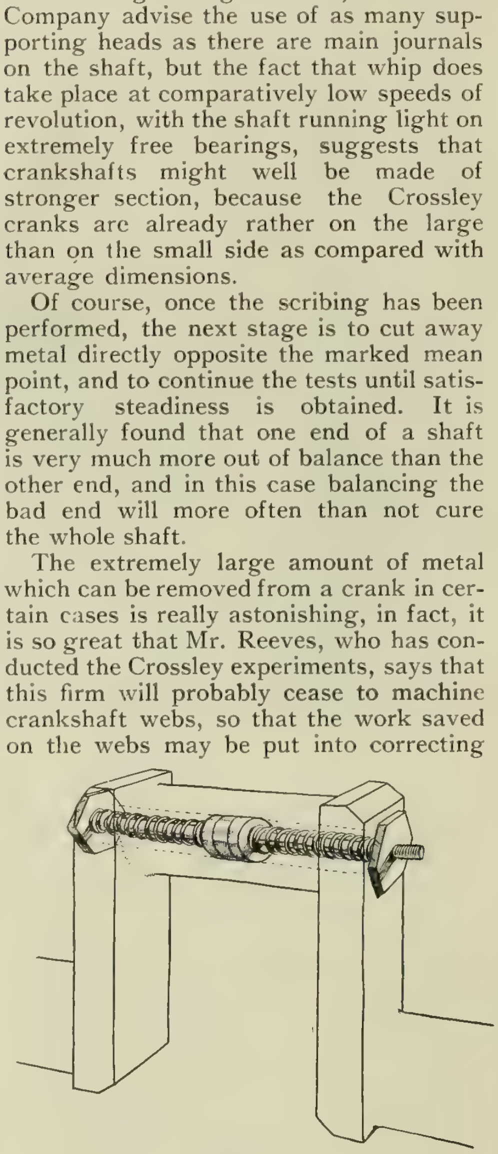


Fig. X.

running balance. That is to say, it appears scarcely worth while to pay much attention to symmetry of form as an aid to running balance.

Balancing of flywheels is rather less troublesome than that of shafts, and in this case it is usual to obtain a rough balance by adding weight at the light spots by placing a lump of putty inside the rim. When a balance has been obtained in this way the putty can be removed and a corresponding weight of metal drilled out from the other side. Similarly with crankshafts rotational additions can be made to the weight of the shaft for preliminary trials, before cutting off any portion, and the Crossley system of doing this is illustrated in Fig. X. It will be seen that the crank pins are hollow and a device similar to the one shown is fitted to each pin. The springs, the nuts and the spindle of each set are carefully proportioned so as to be of precisely similar

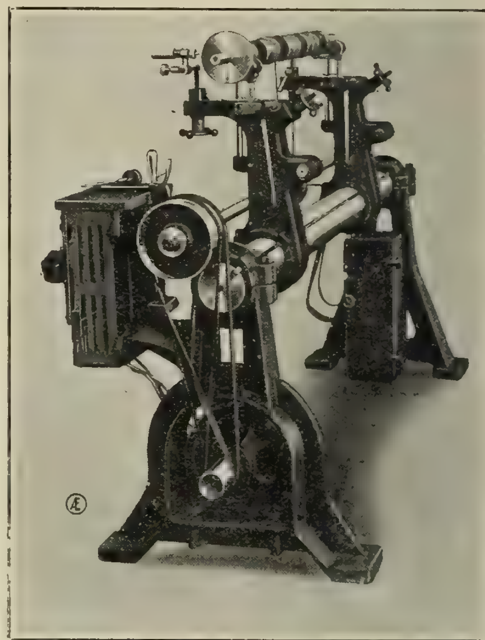


Fig. IX.

weight, and balancing is performed by changing the little weights seen at the centre between the two springs. A whole series of these weights is used, and are changed until the balance is correct, when the difference between opposite pairs indicates the amount of metal to be cut away.

In conclusion, it is interesting to quote Mr. Reeves' opinion on the use of the machine with which he has now been experimenting for about a month. He says—"It is, of course, an axiom that any part in running balance at one speed must be in running balance at all other speeds. Of course this will not be so if there are couples tending to distort the parts from centrifugal action, but there is another factor to be considered, and this is the possibility of the occurrence of large vibrations due to synchronizing between the shaft and its supports.

This action appears possibly to be owing to the fact that at the lower speeds the restraining action of the shaft supports is of sufficient value to partially overcome the unbalanced force of the shaft, and thus cause vibration to occur about a centre which is neither the centre of the figure nor the centre of mass. When the speed goes up, the unbalanced force becomes so great that the restraining effect of the supports becomes more or less negligible."

If, therefore, the shaft is balanced at too low a speed it may be found it is out of balance at high speeds, and it is at the high speeds that troubles from whip begin to occur. For this reason the use of a number of supports, as recommended by the Norton Company would seem to be advisable.

We hope to continue this article shortly, adding the description of another balancing machine.

MOTOR CYCLE AND CAR DESIGN COMPARED.

A study of the present state of motor cycle engineering, together with a brief consideration of what it has learnt from, and what it has taught, the car designer.

By Octavius.

IN the rapid development of the modern motor vehicle it is interesting to observe the effect which the motor cycle engine and its larger prototype, the car engine, have had upon each other. At first sight it would appear that each has developed separately, because the problem which confronts the respective designers is in no way the same. But on further observation it will be found that a very close connection exists, and this has been accentuated by the introduction of a class of car engine which will give a very high power for its cylinder dimensions, as this is also the principal object in motor cycle engine design.

Taking the most pressing problem of present-day designers, namely, smooth running in an engine from which great power is expected, we find that car influence is gradually forcing itself upon the motor cycle designer, and that in almost every case attempts have been made to render the 1911 models sweeter in their running, while retaining the power before given out. In a few instances very material progress has been accomplished in this much-to-be-commended direction.

In this case, however, the motor cycle designer has a very much harder set of conditions than those which face the car engine maker, for, while a certain amount of power may successfully be sacrificed by the latter in order to obtain sweeter running, no such allowance can be made by the former, because motor cycles are at present only compared on a power basis.

Touching now on the methods by which the high power obtainable from engines is produced, one finds that the motor cycle has taught many lessons to the car. A few years ago car engines were usually fitted with heavy cast iron pistons which deadened their whole life, and only rarely was a lightened piston obtainable. Motor manufacturers hesitated to fit pistons of reasonable weight, as far as their touring engines were concerned, though racing practice had certainly already taught them the value of a lightened piston, and it was only when light pistons had become common on standard touring motor cycles that car makers began to give serious consideration to the matter. Meanwhile, motor cycle racing pistons had reached the ultra-limit-point of endurance, and have since returned to a more reasonable weight (still less in proportion than usual for car engines), while the lessons taught by over-lightened pistons have been of use because they have shown the best and safest methods of manufacture. Following the reduction in piston weight came naturally improvements in the engine speed, and this was before the date of the voiturette racing car engines, which are so nearly akin to motor cycle engines. Compression also rose rapidly till a point was reached sufficient to cause overheating, although productive of excessive power during short runs. Then the pressures were lowered until the overheating effect disappeared and left an engine of high power for size and greater softness. It appears that car engines are as yet climbing towards higher compression, and experiencing the same troubles, so it seems a study of motor cycle history would be useful to some car designers in this respect.

Valve Mechanisms.

Next in order come valve gears and valve improvements; here great difficulties have arisen and have to some degree been overcome; rapid action cams, strong springs and well radiused valve heads are some of the results of past experience in the struggle for more and more power. Here, however, by reason of the lower average efficiency of car engines, the influence of the motor cycle is not so pronounced, for car progress has been along the lines of silence as well as power, and certain sacrifices have been made thereto. However, it is not too much to say that the present day car valve gear owes much of its design to the earlier high speed motor cycle engines, and it would be well to add that present car engine valve mechanisms are well worthy of study by those responsible for motor cycle design.

While on the subject of obtainable power a point suggests itself in the difference between racing and touring practice for the two classes: it will be found that motor cycle manufacturers more readily adopt improvements discovered during the production of racing machines than do car manufacturers, many of whom have racing cars with points which are well worth copying on touring chassis, and yet do not become standard. Much of this may be due to the difference in the character of the classes for which both are catering, yet it would seem that motor cycle makers are following a better policy.

In lubrication, curiously enough, we find an unexpected change of sides, since one would anticipate that the makers of the engine with the highest speed would develop the best lubrication system, while, as a matter of fact, car engines are very far in advance of motor cycles in this direction. Forced lubrication is most uncommon for motor cycles, although it has a foothold, and signs are not wanting of the influence of more modern car methods. It has been found that, notwithstanding the necessary increase in price on engines so fitted, forced lubrication is better, and the prevailing price of motor cycles is such that this system would add little in comparison with other details, while the argument that it is less fool-proof or simple does not hold, when one considers those machines which are at present so fitted and the extraordinarily few parts necessary to effectually pump-lubricate a motor cycle engine.

Carburettors.

In considering the bearing which the design of either machine has upon the other's carburettor, the subject must be split under two headings, for, while the car has had a great and beneficial effect on the levers which control the amount and quality of the mixture, yet no sign of any influence exists when the actual carburettor is examined, and it is here, if anywhere, that a gap arises consequent on the nature of the two vehicles under consideration. Control levers were present in a suitably placed and readily accessible form on the car steering wheel at the time when motor cycle drivers were still groping for levers, the actuation of which very often entailed an unusual and totally unnecessary amount of gymnastic exercise. After many had complained—pointing to car practice as a desirable example—a sudden change came, and the row of nickel-plated, be-ratchetted lengths of rod disappeared, leaving a neat, effectual, wire-operated control, placed conveniently to the hand.

In the attempt to get still higher power from the engines, motor cycle carburettors have taken many forms, and have almost universally arrived at an exceedingly simple, if not always effectual, design, consisting in the main of a jet with a large slide exposing a considerable orifice for extra air, a similar slide being operated to control the quantity of gas. Now, had car practice influenced this unit, there would have been a marked tendency to operate the air admission by automatic means, to filter the incoming fuel and to provide means whereby the aforesaid large air orifice should be constrained not to take in much dust as well as air. None of these things being common, it is safe to remark that neither machine has influenced the other, for, except in the means provided for rapidly detaching the float chamber lid, no influence of the motor cycle can be traced to the car. Perhaps it would have been better had car practice had its say in the general arrangement of a motor cycle vaporiser, since the latter is too often a rough piece of mechanism, fulfilling its purpose it is true, but doing so only with a considerable amount of skill on the part of the driver and a trained sense of sympathy with his engine only to be acquired by considerable experience.

Car designers might still learn much concerning the rapid detachment of the jet, for change of size or cleaning purposes, as this can be done in a few seconds on the motor-cycle, while, in many cases, the opening of a door in the under-shield giving access to a cramped space (in which a spanner has to be operated) is necessary to remove the jet or jets from a car carburettor. In other cases there is overmuch inlet piping and many small flanges to be detached before this important portion of the engine can be exposed.

Quietness.

Silencing is another feature in which no trace of influence exists on either side, as both seem to have approached the same problem by totally different roads. In some respects this is the result of a motor cycle engine being nearer akin to a racing machine in characteristics than the car, and in others it is due to the allowance of

power which car manufacturers can give in exchange for silence. It would be well could the motor cycle designer take close note of car practice, for the average motor cycle emits far too much noise and loses much power if it is silenced.

In time it will be realised that the ordinary citizen does not like this exceedingly unpleasant noise, and is apt to class the rider of a noisy machine with a certain vanishing and unpleasant type of cyclist; to the consequent harm of the motor cycle industry. It is known by car practice that a considerable length of exhaust pipe is one of the best silencers, and as long pipes have been tried on motor cycles with excellent results, it seems unreasonable to continue using a two-inch bore steel pipe some twelve inches long and a small, very inefficient silencer, provided with a large cut-out. The cut-out may be necessary, but the remainder is not.

By long experience car designers have discovered that silence is a very paying feature and a selling point worth the few drawbacks it may entail. Although there are regrettable exceptions in the large bonnet, small engined cars of racy appearance, yet there are signs that even these are ceasing to emit quite so unpleasant a din. Therefore, it behoves the motor cycle designer to obtain the same result, or very nearly the same result, from his own type of engine. The car is there to copy, let him take his opportunity!

Ignition.

Turning now to the fitting which brought the motor cycle from an unreliable machine to the extremely reliable machine it is at present, it will be found that the high tension magneto was used largely on motor cycles some time before it was in a similar position in regard to cars, and it is curious that the lower-priced machine should here lead the way to the more expensive car, especially as the magneto has done more for the car than any other single fitting. Even as with the motor cycle, so with the car, the disappearance of the accumulator from the position of chief importance to that of stand-by has improved the reliability to an enormous extent and, although it did not mean the difference between complete disappearance, and an excellent sale, it is safe to say that it increased the number of possible buyers to a very considerable extent.

Concerning the actual attachment of the magneto, on the other hand, the motorcycle has now much to learn from the car, for a motor cycle magneto can usually only be adjusted or moved by detaching four bolts, placed in a position where mud speedily renders them invisible, and moreover, being just above the silencer, it is in anything but a comfortable position to adjust. This, however, one is glad to see, is disappearing, the magneto being placed out of harm's way at the rear of the engine, though the bolts are still inaccessible. The car magneto invariably has a neat form of clip removable by a thumbscrew, whereupon the whole magneto can be detached, and it is to be hoped that this practice, in a suitable form, will be followed by motor cycle makers, there being no insurmountable difficulties in the way. Adjustment for timing purposes is also a point worth copying, as too much has to be removed before a motor cycle magneto can be altered either when chain or gear-driven, compared with the two nuts which are all that are necessary on a car.

In rapid terminal detachment the car learned much from the earlier motor cycles, while in weather protection of those terminals it has still much to learn, for a bonnet is not altogether as excellent a covering as would be expected, cases being known of considerable trouble caused by the engine-cover being raised in rain, during the completion of some minor adjustment unconnected with the ignition, whereas a motor cycle magneto will now stand any quantity of wet without sign of ignition troubles consequent thereon.

The long stroke engine has not made itself felt to the same degree in motorcycles as it has in cars, for although the stroke/bore ratio has exceeded unity for some time, there are considerations which limit the possible length very greatly, as it is necessary to keep the stroke short enough to enable the cylinder to be removed without the necessity for altering or removing the tank. Frequent cylinder removal is rendered necessary by the need for constant removal of carbon deposit, in turn necessitated by the present crude lubricating system. Conditions cannot very well alter while the arrangement of motor cycle tanks remains as at present, though whether it will do so is an open question admitting, as yet, of no satisfactory reply.

Car influence is again seen in the present fashion of providing usefully large filling orifices to both the oil tank and that containing petrol, though straining apparatus is most usually absent.

A practice in connection with extra lubrication devices which has been followed to a certain extent by car designers, is the admittance of an extra supply of air which has already traversed the engine base chamber, consequently becoming impregnated with oil. Although this cannot be said to be in any way general practice in either case, yet it was noticeably present at the recent Olympia Show, while it was fitted to motor cycles quite two years ago.

Cooling and Details.

Water and air cooling are so very much a question of engine size that they can hardly afford a subject for inference in connection with the influence of the motor cycle, but some signs are present that water-cooling may be attempted on motor cycles perhaps not altogether happily, as it involves a certain amount of extra mechanism composed in part of thin pipes ill able to withstand vibrational stresses such as are encountered on motor cycles; and neither rubber joints, nor water, are particularly desirable adjuncts to this type of vehicle.

How much air cooling is likely to influence car design it is impossible to say definitely at present, but it seems largely a question of valve-chamber cooling, and even then this is satisfactory to be suitable only for low powered cars. At any rate, it is a problem which only the future can decide.

Valve covers have in one instance appeared on a motor cycle as a result of motor car practice, perhaps carried too far, as valve springs get over-hot when exposed without the addition of a cover to assist them in this undesirable habit, though, as a prevention of wear in the valve guides with all the attendant troubles thereto, there is much to be said for the practice. While on the subject of valve guides, it would be well to say a word in favour of the general practice on cars, which lays down that these should be of great length and, if possible, detachable, for on motor cycles they are often too short, wear badly, and are incapable of removal. Many do not know the host of troubles which may lie in store for the unfortunate possessor of a machine whose valve guides cannot either be rebushed or replaced, and the few who do, seem unable to affect the manufacturers.

The starting from rest problem reveals the influence of the car, for the very tardy appearance of free engines and clutches on motor cycles and the design of these clutches is nothing if not a copy of car practice. Although the space consideration complexes the design to a very considerable degree, yet it is to be hoped that even greater advances will be made, since starting a machine which may easily weigh two hundred pounds is a performance needing a skilful and athletic rider.

One point which it is to be hoped will not be copied is the foot clutch control, as it is far easier and safer to use a hand-clutch lever actuated preferably from the handlebars themselves, than to use that provided for foot operation, and the difficulties and intricacies of design so caused are to be surmounted by the exercise of reasonable thought and engineering knowledge.

Again, in the matter of gear changing, although the two cases are not in many ways similar since sliding gears are of little use to motor cycles in their present form, much has still to be learned from the car.

Epicyclic gears for motor cycles are, in the main, more unsatisfactory than those used in car work, though the influence of the latter can be felt throughout the present designs, and, it is to be hoped, will be more so. It is not intended to imply that the present type of gear is utterly useless —although many complaints so worded have come from motor-cyclists—it is merely that these gears are not yet as good as those which are fitted to cars of reasonable weight, and the space difficulty, although great, can, and will be, overcome.

At one period motor car influence cast itself on the motor cycle brake-gear, but has departed, as it was not then realized that the two problems were not the same. Band brakes and internal expanding shoes are excellent in practice as a brake, and are indeed much softer in action than the modern rim or pulley brakes; but they possessed disadvantages which rendered them useless. In the first place they were too small, and much too open to the grit and mud ever present on them, while in the second place there was the rapid removal of the wheel, and the replacement of worn parts to consider. It is far easier to remove a wheel with the pulley rim brake as at present fitted, than with a car pattern brake, and the replacement of the fibre or leather block is a very easy matter.

The question of wheel detachment is of far greater importance than could at first be realized, and wonders have had to be performed to render this an easy operation, since it is the sole method of tyre or tube replacement. Thus it will easily be seen that there are quite sufficient things to be removed already, without

the brake gear, and that there is no likelihood of altered design.

In drives, again, we have different positions to consider, and the belt is likely to hold its own, because a very great point in its favour is the ease with which a gear may be altered, a factor which has greater influence than might be imagined. Shaft drive seems an ideal, perhaps not altogether a happy one, with a single cylinder, but should the care and thought apparent in car shaft drives be embodied in one suitable for a motor cycle, who shall say what could not be achieved? Certainly it would be better than either chain or belt, though the gear change, unfortunately, disappears.

The Two-Stroke Engine.

Turning now to the two-stroke engine, we find that great strides have been made in a subject which is really only just commencing to interest car manufacturers. Many motor cycle makers are experimenting quietly with these engines, and have the advantage of a great experience over the car manufacturer. They are undoubtedly engines from which, notwithstanding their present crude form, much is to be expected. The difference between driving a two-stroke motor cycle and one with the more usual engine, is near akin to that between a two-cylinder and six-cylinder car, omitting the questions of great power and great speed, and judging solely from the actual feel of the engine. Gradually these engines are improving, and it will be strange indeed if the first thoroughly powerful two-stroke engine design does not come from the hands of a designer whose experience is based on efforts connected with the motor cycle. A vast amount of time, energy, and money has made the Otto cycle engine for motor cycles what it is to-day, and should an equivalent amount be gradually expended on the two-stroke, a successful engine is likely to result, and it would bring the motor cycle the one thing necessary for widely extending its present scope.

Unfortunately motor cycle making has so far not received as much attention as it deserves from an engineering point of view. The motor cycle has suffered greatly on account of the extremely small number of the designers with a sound engineering training who have been, or are even now, connected with the industry. Even if it were assumed that cycle building is not engineering in the usual sense of the word, it is surely a fact that the construction of a power-driven vehicle must be an engineer's job. An excellent example of the retardation of progress that has taken place owing to motor cycle manufacturing methods is to be found in the evolution of the twin-cylinder engine. It will be remembered that the automatic inlet valve was unknown on cars literally years before the mechanical valve was at all common on motor cycles. The automatic valve was first banished from the single-cylinder machines, but with regard to the two-cylinder engines there was a curious piece of misplaced reasoning, it being said in earlier times that the mechanical inlet valve was unavoidably absent because the expense necessitated by the altered valve gear was too great. Yet many owners of twin-cylinder engines with suction valves complained about them, and often a machine of some 5 h.p. so fitted would prove itself less powerful than a single-cylinder mechanically-valved engine of approximate $3\frac{1}{2}$ h.p. Still no alteration was made, but when a certain machine, the twin-cylinder engine of which had mechanical valves, was demonstrated to be infinitely better than the older type a change occurred abruptly, every machine being hastily fitted with more or less efficient mechanical valves. It was then found that the cost was very little, and in some cases really brilliant ideas were conceived to lessen the number or the complexity of moving parts.

The Cycle Car.

At the present moment there is an interesting link between car and motor cycle which, although not by any means common, yet shows signs of becoming a factor in modern motor transport. It takes the form which, for lack of a stricter definition, can be called a four-wheeled motor cycle. Generally it is of extremely crude construction, having a small motor cycle engine, a rough wood frame, four wire wheels and some cheap form of belt drive. Seating accommodation for two may sometimes take the form of a canvas stretching between two supporting tubes. Even this crude machine has qualities which render its success likely in the matter of winter driving and the transport of two passengers, but its chief advantage lies in the extreme lightness of its prime cost.

Should this form of machine become a standard model with motor cycle manufacturers in days which are to come, it is probable that it will influence the motor cycle proper, by the attention which it must draw to car practice. Thus it is to be hoped that the car and motor cycle, instead of attempting to run on lines differing greatly from each other's practice, as at present, will become fused into a single branch of the engineering trade, working solely to the end that both are now really trying to attain, but with the help of each other's knowledge, creating thereby a mutual goodwill, the lack of which has had a considerable share in retarding the progress of types of automobiles.

TOOLS AND FIXTURES FOR MACHINING PISTONS.

Of all the component parts of a petrol engine, or indeed of any kind of high speed engine, no part requires greater care in its manufacture than the piston, as smooth working and efficiency depend upon it to a very large extent. For this reason a piston should be subjected to a very rigorous inspection after each machine operation, as the castings are very thin and fragile, and are very liable to get cracked in chucking operations, also, for various other reasons the percentage of wasters is generally rather high.

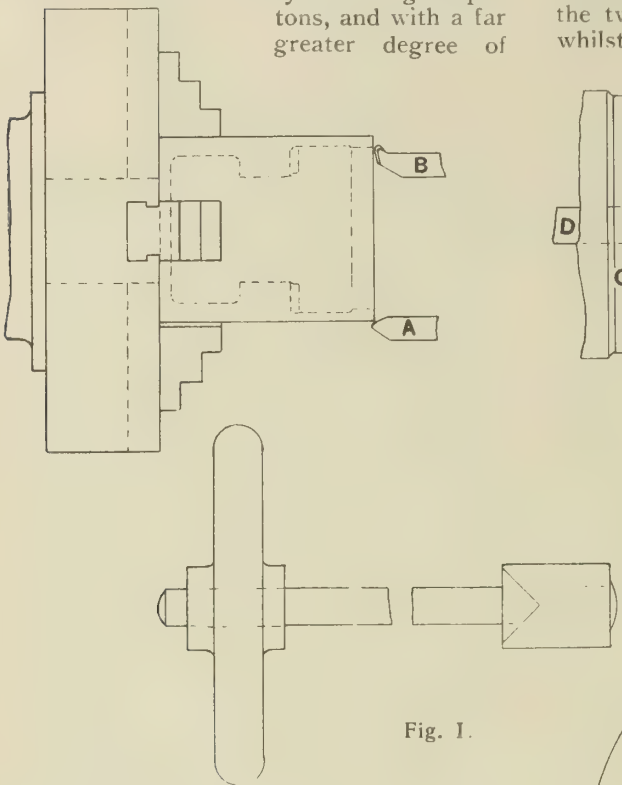
The system about to be described is one of the most modern methods of machining pistons, and it embodies quite the latest practice. It includes some very ingenious fixtures to facilitate rapid and accurate production, and novel methods of performing the operations. Fig. I. shows the first operation, for which the castings are gripped in a three-jaw universal, or a four-jaw independent chuck, and needless to say it is typically a turret lathe job, though two types of machines offer alternatives, namely, a flat turret lathe with a cross sliding head-stock, or a rigid capstan lathe with automatic power feed and traversing cut-off rest. The bottom end is first faced up with the straight tool A, the small shoulder, which is shown dotted, next being bored out by the boring tool B, so as to fit accurately on the turned spigot C.

This spigot is screwed on the lathe nose in place of the chuck and, of course, is turned dead true in position. For additional rigidity the piston is further secured in position by a long bolt D (also shown at Fig. I.), which passes through the hollow spindle and has, secured on the front end, a steel tee-piece with a vee-cut in each side, which grips on the top of the two bosses surrounding the gudgeon pin hole inside the piston as shown dotted in Fig. I. The handwheel shown at the back end is used for tightening the work without the aid of a spanner. While rigged thus the piston is faced on the end and turned over the top, a roughing and finishing cut being required with a tool held in a holder carried in the turret. Next the three grooves for the rings are cut, all at the same time, by three spaced cutters held in a special holder, a finishing cut being necessary for this operation also. The front cut-off rest carries the roughing tools and the back rest the finishers, while about 0.008in. to 0.010in. should be left on the outside to allow for grinding to a finish.

After being inspected for turning the pistons are now mounted on a special angle bracket fixture, as shown in Fig. II., this fixture being secured to the turret lathe spindle, either by screwing it on or by securing it to a face plate through a small spigot at the back, and four bolts. The bottom flange is bored or recessed out about 3/16in. deep, to locate the piston, while the two bolts clamp it down by the swinging clamp. In the same illustration the method of locating the casting in the correct position for boring the gudgeon pin hole central is shown dotted, and here E is a steel plunger with a vee planed across the top, which fits accurately in the milled sides of F, the latter being drilled as shown to receive the shank turned solid with E. The two springs force E upwards which locates the piston by the set-screw lugs on each side of the boss: of course, the operator drops the piston in position with the lugs approximately accurate. The hole is now bored with a twist drill, is then opened out with a three-lipped chucking drill and finally is reamed to size.

One of the most intricate operations, and at the same time one of the slowest on work of this description, is the facing of the inside bosses to width with facing cutters, on a drilling machine, and to obviate this the rig shown in Fig. III. was designed to be used on a plain milling

machine. It completes the operation in five minutes each, on 20 h.p. four-cylinder engine pistons, and with a far greater degree of



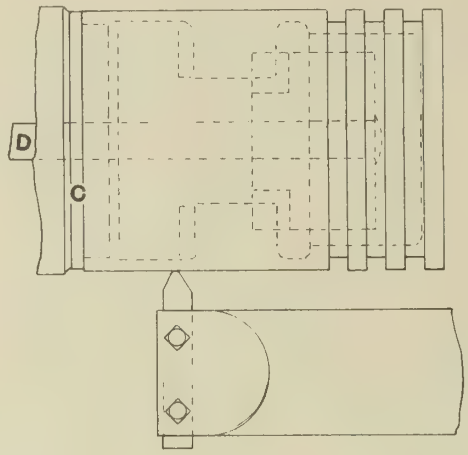


Fig. I.

accuracy than by drilling machine methods. As it is a novel arrangement it may require a little explanation: H is a malleable iron casting, also shown separately at the side, and this is bored out to clamp on to the top arm of the milling machine. A driving shaft I takes the place of the cutter arbour and has a short keyway milled in it to carry a feather key which drives the small gear enclosed in H. This gear transmits the power to the two milling cutters J, through the train shown. The gears are all of steel, while the lower of the two has a short arm at each side turned solid with it and screwed at each end for the attachment of the cutters. A loose plate fits on one side of H, to protect the train of gears and avoid overhung bearings, while also acting as a guard for the operator, and the plate is secured in position by eight cheese-head screws, the holes for which are shown. This attachment is a very good example of modern tool room practice and, for pistons of $4\frac{1}{4}$in. diameter and upwards, is a great labour saver. The piston is held on the milling machine table by the jig K, the two levers L clamping the castings, whilst the two parts M are spring plungers which fit accurately in the previously bored gudgeon pin holes, and locate the pistons positively in position. When thus secured the vertical feed is put on and the

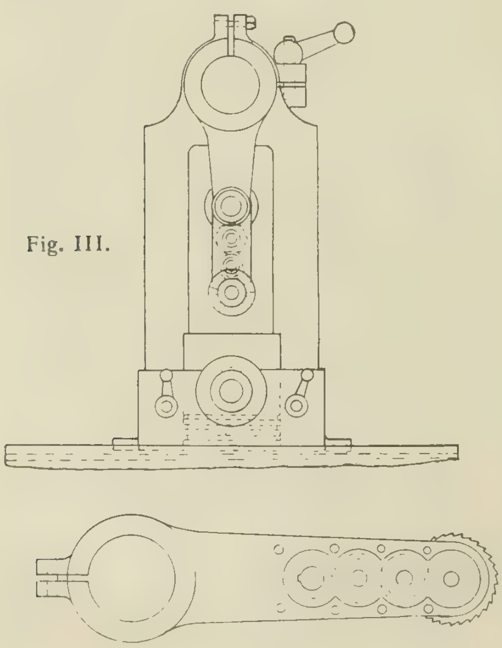
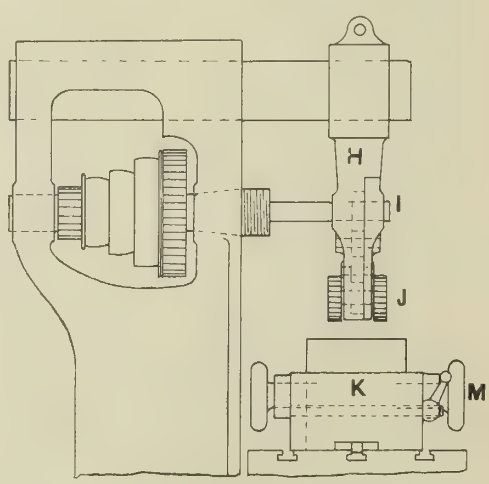
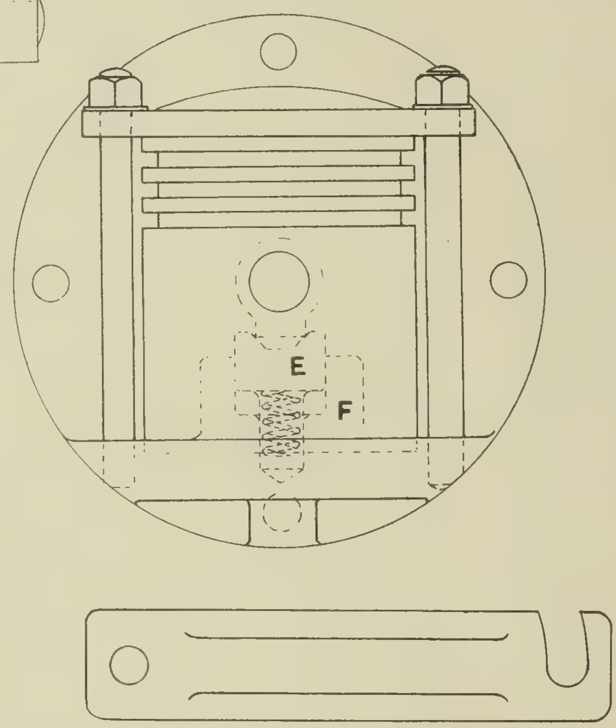


Fig. III.



cutters J mill the inside faces to width at one cut. The plungers M are then pulled back, and the levers slackened, when the piston can be taken out of the jig.

The next operation is to drill and tap the two set-screw holes inside the piston for securing the gudgeon pin in position, and there are various ways of performing this. A convenient method is shown in Fig. IV., and it will be seen that it consists of a simple casting carrying two hard steel bushes through the long bosses at each side for that purpose, a flange being turned to fit in the bored out end of the piston. A long lug projects down inside the piston and drops between the milled faces of the bosses, a springing pin P with two saw slits milled in at right angles being passed through the gudgeon pin holes and the hole in the boss, thereby locating the jig in position for drilling the two set-screw holes central with the pin. This operation is often done on a high speed sensitive drill and the tapping of the holes can also be done on the same machine by one of the automatic reversible tapping attachments, now usually supplied for use with these drills, with which the act of raising the drill spindle reverses the tap and backs it



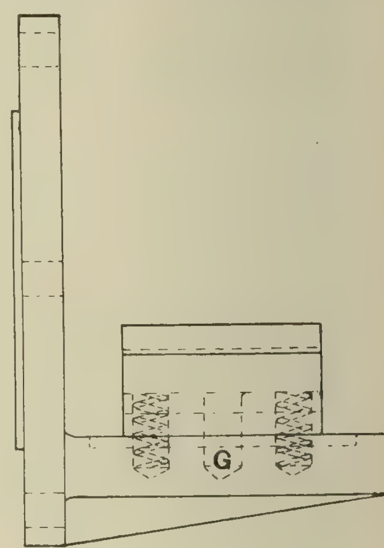
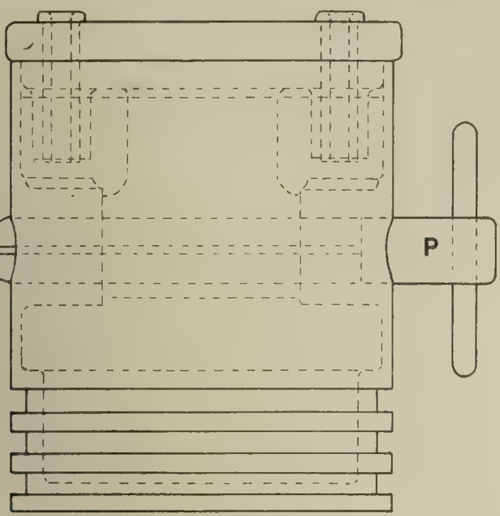


Fig. II.

out. In this case, however, a special long tap would have to be used, as the device would not go down the inside of the piston. Alternatively these holes are often tapped by hand. It should be pointed out that, if the faces on these bosses are machined, they are faced in the turret lathe with a facing tool at the first operation when the end is bored out.

The remaining operation, and one of the most important, is the grinding to size on the outside. For this the piston is secured in position on the grinding machine on a spigot and drawn up by a bolt in the same manner as shown in Fig. I. I have seen good results obtained by grinding the pistons wet, with Norton and carborundum grade 60m. wheels. To give the best results, pistons should not be ground perfectly parallel, but should taper larger towards the bottom or open end, that is to say, a piston to work in a ground cylinder $4\frac{1}{4}$in. diameter should be ground 4.240in. at the top, where it comes in contact with the flame, and it should taper to the bottom, where it should be about 4.246in. diameter. Thus the top of the piston will be about 0.01in. smaller in diameter than the cylinder and the bottom

about 0.004in. smaller, as the top will expand more than the bottom. The allow-



ances here given will be found to give good results in practice, providing always that the lubrication is good.

The times for completing a piston of this size, about $4\frac{1}{4}$in. diameter, should be

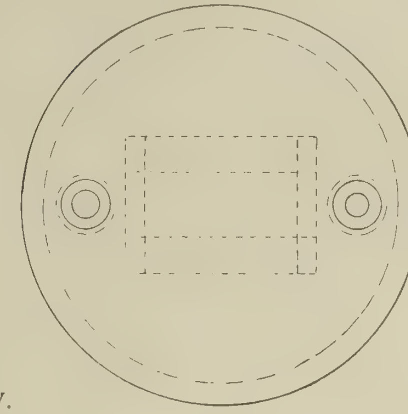


Fig. IV.

as follows for the various machining operations :—

First operation on turret lathe to bore, face, turn and cut ring grooves complete, should amount to about forty minutes.

Second operation, also on turret or capstan lathe, bore and ream gudgeon pin hole, about fifteen minutes.

Third operation, mill inside of gudgeon pin bosses to width on milling machine with special fixture, about five minutes.

Fourth operation, drill two set-screw holes in gudgeon pin bosses, five minutes.

Fifth operation, grind to size outside, thirty minutes to thirty-five minutes.

In cases where a small peg hole is drilled in each of the three ring grooves, to prevent the rings working round, this operation, which is performed without any special jigs or fixtures, will take about three minutes. Therefore the total time, exclusive of assembling, is one hour thirty-eight minutes each. These times allow for first-class workmanship, and include all setting up times in the various machine sections, but the quantities required will have to be not less than fifty on one order.

D. WALTERS.

AN INVESTIGATION OF BENDING STRESSES IN CONNECTING RODS.

By W. H. B.

THE advent of the quick revolution internal combustion engine has brought about a decided change in the design of the connecting rod. The reason for this is twofold. In the first place, the designer of an engine for traction, aeroplane, or dirigible work, in which weight per b.h.p. is such an important consideration, soon finds that, in his horse-power formula, the factor denoting revolutions per minute is going to be of great assistance in the ultimate reduction of overall sizes and consequent weight. Now, owing to the effects of inertia, the great number of accelerations

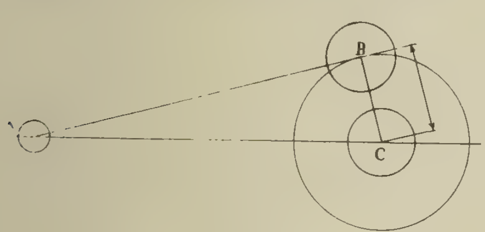


Fig. I.

and retardations of moving masses, which occur in a comparatively short time in a quick revolution engine, is only permissible where the weight of these parts has been cut down as much as possible in order to obtain an engine which will run smoothly at a high rate of revolution and cause the least possible amount of vibration on the engine bed and bearers. The second reason for the importance of care in the design of reciprocating parts is, that the frequent reversals of motion cause considerable stresses to be set up in the parts themselves, therefore the amount of material available for their construction must be put to the best possible advantage, and the metal itself must be of first-class quality. It is a fact that, in a badly designed rod, metal may be judiciously cut away so that, although the weight and cross sectional area are reduced, the actual stress per square inch when in use will be less, due to the fact that the forces acting on the rod caused by its own inertia have been considerably diminished. To determine whether the section of a connecting rod is the most economical, and if the material is being allowed a sufficient factor of safety, a thorough investigation into the various stresses set up is necessary, and the writer thinks that the following method, though not original, is not so well known as it should be, as it appears to give a very clear insight into the somewhat complicated stresses which arise, without involving the use of higher mathematics, and may be solved entirely by graphics giving results which will convey at a glance the magnitude and direction of the loads under which the rod might fail.

In the diagram Fig. I. the centre line of the crank and connecting rod are lettered *BC* and *BA* respectively. The angle *ABC* is a right angle, this position of the crank relatively to the rod giving the maximum of angle *BAC*. The direction of revolution of the crank is clockwise, and the piston is assumed to be on its firing stroke. With the piston in this position the acceleration in the line of stroke is nil, both piston and gudgeon pin having their maximum velocity at this point. What may be termed the vibratory acceleration of the connecting rod is, however, at its maximum, the angle *BAC*, after having increased from $O°$ at top dead centre, being just about to commence decreasing till bottom centre is reached. It is the bending stress produced by this acceleration which we have first to examine. It will at once be seen that the vibratory acceleration of the rod at a point coincident with the crank pin centre will be the same as the centripetal acceleration of the crank pin itself, and will therefore be equal to $\frac{v^2}{r}$ feet per second, where v = circumferential velocity of crank pin centre in feet per second, and r= radius of crank in feet. Proof of this formula may be readily found in any good text book on engine design, and there is no need to enter further into that in this article. We have already seen that the gudgeon pin centre has no acceleration at this point of the stroke; it therefore follows that the acceleration of that portion of the connecting rod attached to the gudgeon pin will be zero. We can therefore use Fig. I. as an acceleration diagram by making

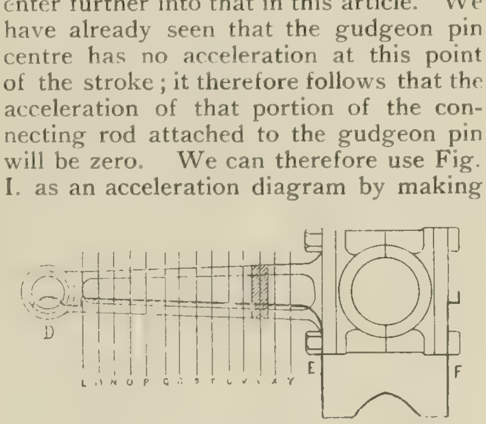


Fig. II.

the line *BC* represent to a suitable scale the quantity $\frac{v^2}{r}$, the line *AB* being to some other scale, the length of the connecting rod terminating at the gudgeon pin centre *A*. The vibratory acceleration of the rod at any point on *AB* can then be found by measuring off to the same scale as *BC*, the length of a line drawn from that point to the line *AC* at right angles to *AB*.

We must next plot a curve giving the weights of small slices of the rod taken along the whole of its length, so that the sum of these weights will equal the total weight of the rod. To do this, divide the rod, as shown in Fig. II., into a series of vertical sections marked in the figure L M N O P—Y. The more numerous these sections, the more accurate will be the final result. The weight of that part of the rod on each side of any particular section may be taken as acting along the line of that section, thus in Fig. II. the

weight of the shaded portion is taken as acting wholly along the line w. By setting off ordinates to a convenient scale, we may construct a curve of weights as shown in Fig. II. by the line *DEF*. The well-known formula for finding the force necessary to produce acceleration is

$f=\frac{Wa}{g}$ where f=the force required in lbs., W=weight of body in lbs., a=acceleration in feet per second per second, and g= 32.2.

Therefore, supposing we take the weight at each of the points already obtained on the weight curve in Fig. II., and find the corresponding acceleration at these points from the acceleration curve in Fig. I., by the use of above formula a figure may be obtained for each point giving the force necessary to produce that acceleration. These values must be plotted to a suitable scale giving a curve which may be termed the inertia force curve, and it will clearly be seen that the connecting rod can be considered as a beam supported at two points, viz., the gudgeon pin and crank pin centres, and loaded with a distributed load the values of which are given by the curve A in Fig. III.

Of course the vertical component of the weight of the rod should be added to these values, but the amount is so small, especially in vertical engines, as to be negligible without leading to any really serious error.

Having constructed this curve, the resulting bending at the various points along the rod may be calculated by the method of moments, or obtained by the quicker and less tedious graphical construction by means of a polar diagram. Care must be taken to calculate the *bm* along the whole length of the rod, as the negative *bm* on the further side of crank pin centre has a considerable modifying influence on the positive *bm*, occurring between the points of support.

Another source of considerable stress in the connecting rod is that due to the friction of the big end brass on the crank pin. This load acts in the same direction as the load due to inertia force, and therefore increases the total bending

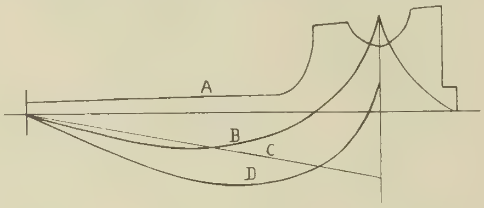


Fig. III.

moment. A good proof of this stress due to friction may be seen on examination of a rod which has failed, due to a big end having seized on the crankshaft. Such a rod will always show signs of having been subjected to severe bending stresses, in many cases having been doubled up even to the point of fracturing.

To obtain this *bm* the rod may be considered as a cantilever with a force applied at the centre of the gudgeon pin and tending to revolve the rod round the crank pin against the frictional resistance. The curve for this *bm* will therefore be a straight line starting from zero at the gudgeon pin centre and reaching its maximum at the crank pin centre. This latter value will be equal to the moment of frictional resistance and may be calculated from the following formula: $P\times$coefficient of friction between brass and pin$\times$radius of crank pin. P=the total pressure on crank pin and is the resultant of the thrust on the connecting rod, when the piston is in the position shown in Fig. I., and the proportion of the inertia force load borne by the crank pin. As these two loads are acting at right angles to each other, the resultant, *P*, will be :—

$$\sqrt{(\text{connecting rod thrust})^2 + (\text{inertia force load})^2}$$

The curve for frictional resistance bending is shown at B, Fig. III., as is also the curve of combined *bm*, which gives the total bending on the rod at D. The stress at any point of the rod can now be found from the modulus of section, note being taken of the fact that the stress due to piston pressure should be added to the stress due to bending on the compression side of the beam, and subtracted from the bending stress on the tension side. This means that in a single acting engine, one side of the connecting rod will always have a higher stress, and this stress should not be more than is safe for a column of a proportion of length to depth similar to the connecting rod under investigation, calculated with a factor of safety suitable for rapidly alternating stresses.

THE 15 H.P. STRAKER-SQUIRE CHASSIS.

A chassis which is unusually accessible throughout and has a neatly designed gearbox.

THE 15 h.p. Straker-Squire is peculiarly interesting because Brazil Straker and Company are one of the few firms that have adopted the American system of concentrating all their energy on the production of one model. There is much to be said for this practice, since patterns and dies are greatly simplified thereby, and the cost of production is likewise lessened. Neglecting the policy of the firm, however, there is no trace of American practice in the actual design of the car, as the following description will show.

The engine, Fig. I., has four 87 x 120 mm. cylinders cast in one piece and provided with fairly large water jackets for natural circulation. Care has been taken to remove superfluous metal on the valve chambers, so that the retention of undue heat may be avoided and the cooling water brought as near the valve seating as grinding allowance will permit. Still further improvement might perhaps be made by reducing the weight of the valve caps. Ten holding-down bolts secure the cylinders to their base chamber, and these are carried through the aluminium in order to form studs for the main crankshaft bearing caps. The cylinders are closed at the top by plugs on which studs are formed to secure the aluminium water outlet casting by which the upper portion of the water jackets are sealed and the water is conducted to the radiator. As the nuts on the tops of the casings of this kind often allow water to escape and spoil the engine appearance, it would perhaps be worth while to evolve a more water-tight joint for holding down this casting.

A single pipe, held to the casting by eight studs, conducts the exhaust gases to the silencer, care being taken to arrange the nuts in a suitably accessible manner. Although a five bearing crank is used the cylinders are not spaced from each other, and the engine is in consequence considerably shorter than is usually the case. The crankshaft is a stamping in Vickers forty-ton carbon steel, with journals and crank-pins of 40 mm. diameter, and we understand that the shafts are turned before being finally ground to size.

Four bolts are used to secure the big end bearings and Fig. I. clearly shows the packing which is introduced to facilitate the adjustment for bearing wear. White metal is used on all the big ends and crankshaft bearings.

The connecting rods are manufactured from stampings in high tensile steel, with a gunmetal bush pressed into the small end for the gudgeon pin, which is retained in the piston by a distinctly unusual method. No set screw or ring fixing is used, but each end of the pin has a semi-circular gunmetal cap, which is allowed contact with the cylinder wall. Although peculiar, the continued use of this method is in itself proof that it is satisfactory, and it should certainly be reasonably inexpensive.

Concerning the pistons there is no call for remark, except perhaps for the fact that reciprocating weight has been carefully attended to, and the pistons are probably as light as is possible with cast iron.

By means of skew gears a silent drive is obtained for the camshaft, which is turned from a solid bar of Ubas steel. Although this is a very general practice and one which has many commendable points, it is conceivable that results quite as good could be obtained from a stamped and ground shaft, while the saving, both in time and money, would be worthy of consideration. The shaft has plain bearings and the flat-sided cams strike the plain ended tappets direct, each tappet being slotted and pinned to prevent rotation in its gunmetal bush, while a light spring holds the tappet head against the valve stem : the gunmetal bushes are a driving fit in a sleeve formed in the cylinder casting. This design of cam gear, although simple and inexpensive, has one very noticeable omission in that there is no allowance for wear. It would be easy to provide an adjustment for the tappets, and such a refinement is undoubtedly a genuine advantage, for without it the removal of the cylinders is necessary when a new tappet may have to be fitted. The valves are 50mm. in diameter and have well radiused heads, while the bearing surface provided in the guides appears to be ample.

A cover plate is fitted having a fixing device which is at once simple and efficient, as is shown clearly in Fig. I., re-

membering that the disc is pressed inwards before it is rotated one quarter of a turn, thus releasing the cover plate. It is perhaps worth suggesting that a device similar in principle might be used with advantage for the crank chamber inspection plate instead of the small nuts and studs at present employed.

Lubrication is by force-fed oil to the main bearings and troughs, combined with splash to the big-ends, piston and gudgeon. A skew gear on the camshaft drives the gear type oil pump which, like most of its class, is situated in the sump and therefore not too accessible. A gauze grid is fitted over the pump inlet, and an indicator is attached to the dashboard, by which the correct working of the pump can be readily ascertained, while, unlike the majority of designs, the copper oil leads supply the underside of the bearings, instead of the point of minimum pressure. It is also noticeable that there is an oil-thrower ring at each end of the crankshaft. Small copper pipes fitted to the big end brasses form the scoops, and these are rather smaller than is generally the case. For replenishing the sump a large aluminium funnel is provided on the near side of the base chamber, but the sole means of emptying the latter seems to be the removal of the plate seen beneath the pump, though a tap for regulating the oil level is provided.

Ignition is by a magneto driven by a skew gear from the camshaft and provided with an adjustable coupling for timing purposes. The magneto driving shaft runs in plain bearings and an oil-thrower ring is again used, a most unusual fitting for this part and one to be highly commended. An easily movable clip holds the magneto to its base, and particular care is taken to render the terminals and contact maker accessible by the means of raising this base. Wiring is led through a neat brass tube clipped to the exhaust pipe and rounded at the corners to protect the insulation.

Keyed to the forward end of the crankshaft there is a pulley driving a small aluminium fan by means of a flat leather belt; it will be noticed in Fig. I. that the bracket on which the fan is mounted has an eccentric adjustment, locked when in position by a set screw and nut seen above the bracket. There is no possibility of shake developing in the long fan bearing, while a neat greaser takes care of the lubrication.

Four crankcase arms support the engine, which is hung on the frame in an interesting manner plainly shown in Fig. I., it being only necessary to explain that the frame is pressed to form an engine carrier integral with the main side member, and that aluminium packing pieces are used, on which the engine is bedded. Another commendable point is the provision of a marked flywheel and a pointer for setting the valves and the ignition, for though this is a feature which ought to be found on all engines, it is lamentably rare.

Careful design is again obvious in the neat manner with which the induction pipe is bolted to the cylinders, for there are lugs on the pipe which bring the holding down nuts to a readily accessible position. A pipe of the ordinary Y type is used, having a main diameter of 40 mm. and branches of 44mm., with a flange at the base for attachment to the carburettor.

This carburettor is one of the most interesting features of the car, and Fig. II. is a section through the float and throttle chambers. Petrol is admitted through the needle valve shown and rises in the central tube, flowing through the lower pair of orifices until it reaches a level somewhat above the upper pair. At the top this tube is open to the air and provided with a nut and locking device for adjustment purposes. The quadruple cone spray communicates with a passage formed outside the tube and consequently engine suction causes a flow of air down the tube where, on meeting the petrol it carries the latter up through the spray and throws it into the mixing chamber, at which point the rush of main air carries it to the engine. An ordinary sliding piston type throttle is used with a vent arranged as shown in the drawing so that, with the throttle closing the main pipe, enough mixture can leak through to keep the engine running at a low speed to an extent governed by the position of the set screw. The needle actuating levers are of the simplest description, being merely pieces of bent spring-steel dropped into position under the regulating needle he[?] This needle valve is itself unusual, as it is provided with a large steady bearing, a most commendable feature. Heating of the vaporiser jacket presents no difficulty owing to the high position of the carburettor.

A 60 mm. diameter steel pipe carries the exhaust to a silencer, which is suspended from a cross member by two bolts and a bridge piece, and thence a smaller pipe passes to the back. Such a silencer suspension is unusual, but seems stiff enough in practice, still a debatable point is the securing of the silencer ends with screws, as these have a knack of rusting into position after some service.

The pressed steel leather-covered cone clutch is shown in Fig. I., the outside rim of the pressing being slotted to allow a certain amount of natural spring to facilitate easy engagement. Wisely, a large and prominent greaser has been provided for spigot lubrication. As the spring collar is on the end of the crankshaft all thrust is neutralized until the clutch is held out. The very large leverage given to the pedal is apparent on the drawing, as also is the spring-actuated clutch stop seen behind the actuating collar, which slides against the coil springs on the two studs shown. Adjustment can be effected by undoing two lock nuts and unscrewing the studs for the necessary number of turns. The fulcrum tube for the clutch pedal is held in a socket bolted to the neutral axis of the side member at the off-side end, and in a strong, but rather heavy bracket secured to the gearbox by two 10mm. bolts at the other end.

A large universal joint is used between gearbox and engine (see Fig. I.), which should be quite capable of dealing with any movement which may take place in any direction. Greasers are fitted to two of the hollow bolts in order to force lubricant to each bearing surface.

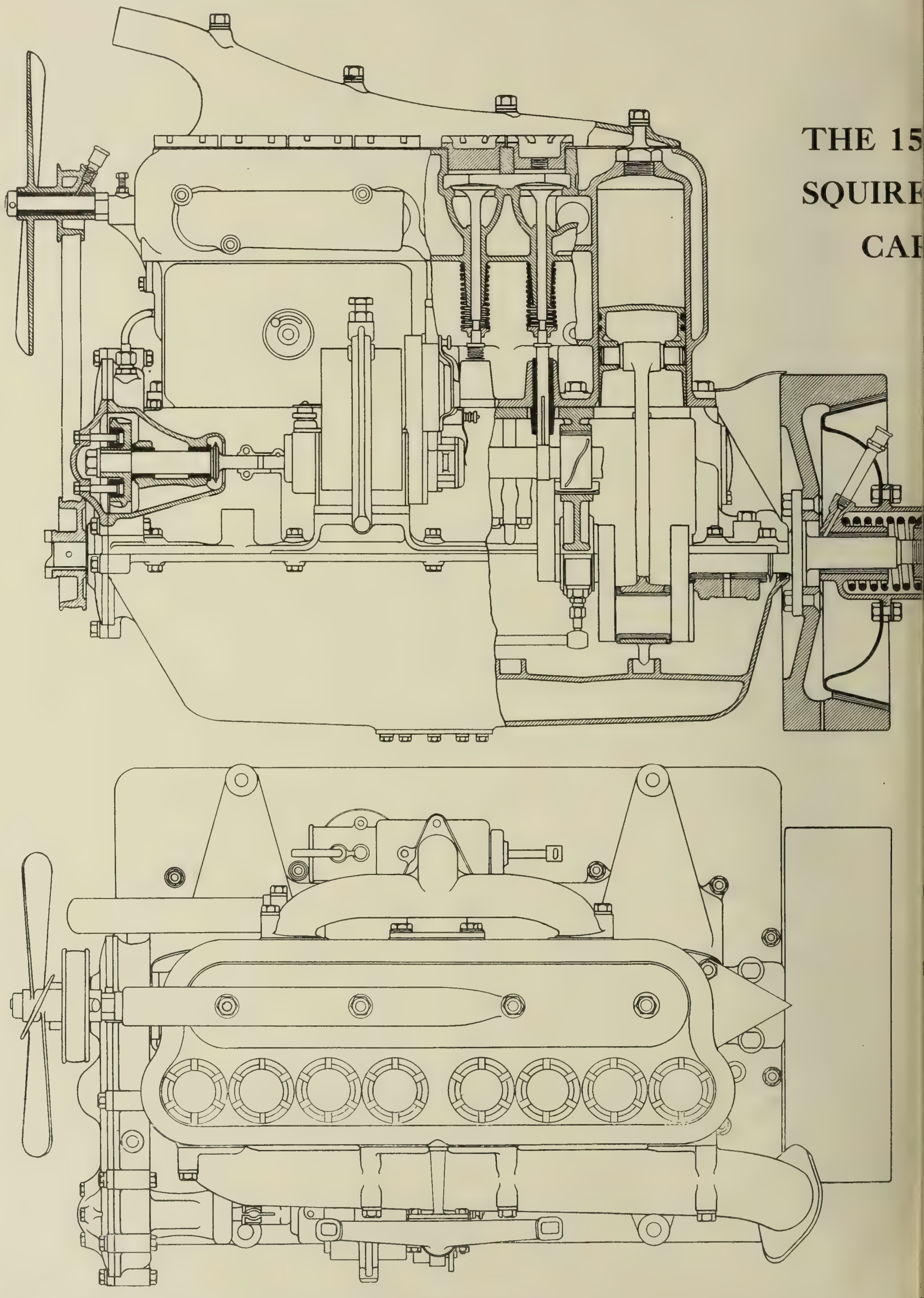
Sliding movement of the clutch is taken up on the splined end of the main gearshaft shown in the gearbox views, Figs. III. and IV. A single aluminium casting is used for the box itself, the shafts being fitted through chambers bored in bosses at the sides. The box is carried on the same portion of the frame as that used to support the engine, while exactly the same type of aluminium packing pieces are used for alignment. It will be observed that the box is set slightly out of the chassis centre, because the first and second motion shafts are on the same horizontal axis.

There are only three speeds with ratios of 12.8, 6.85 and 4.062 progressively, and the dog clutch direct drive is on the third speed, the secondary shaft always remaining in mesh. Ubas steel is used in the manufacture of all the gears which are run in with oil and, if exceptionally noisy, rubbed down by hand with oilstone. A shaft with four splines is used for the sliding gears, while there are small double ball races fitted inside the constant mesh gear, and outside the spigot end of the primary shaft. To meet stress advantageously the two journal bearings on the main shaft are situated some distance apart, and at the tail end of the first motion shaft there is a very large ball bearing housed, in common with all others, in the aluminium. The ball bearings on which the second motion shaft runs are of slightly smaller diameter than those on the other shafts All bearings have small guards as far as possible to prevent the intrusion of metal particles, and this gives proof of careful design.

The reverse has the same ratio as the first forward gears, namely, 12.8, and an interesting feature is the manner in which the reverse shaft is thrown out of gear when the forward gears are in use. It will be seen in the drawing that there is a large spiral spring which normally holds both reverse pinions out of gear. This spring is compressed when the pinions are engaged, by the action of the striking lever, which forces the engaging gear into contact with the flange shown rivetted to one of the reverse pinions, thus moving the two reverse pinions to the right, against the spring action. Plain gunmetal bearings are provided for the hollow reverse shaft.

Taken as a whole, the gearbox bears evidence of careful thought and a great regard to accessibility, as the clearance is enough to permit the easy removal of any article which might be carelessly dropped to the bottom of the box. The escape of grease from the bearing caps is probably hindered greatly by the fragment excluding washers on each ball bearing, as well as the felt pads which are not, by themselves, very effective. Again, the lid has been designed so that full advantage of the aforesaid clearances may be taken and all adjustments to the striking gear or other parts are rendered extremely easy thereby.

Turning now to the gear operating lever, Fig. V. This is of the usual gate pattern, with a trigger for operating the reverse and a notch into which the lever springs to resist the coil spring in the gearbox. Daimler type selector levers are used with a somewhat different form of spring locking gear rivetted to their base, and fixed to the actuating tubes by grub screws. These tubes are wisely provided with a lubrication hole countersunk into the bracket, but would be better if provided with a small spring lubricator, and it may here be well to remark that the same applies to the pedal gear. Each striking fork is keyed and bolted to the actuating tube, and each of the forks has a swivel piece at the inner end so



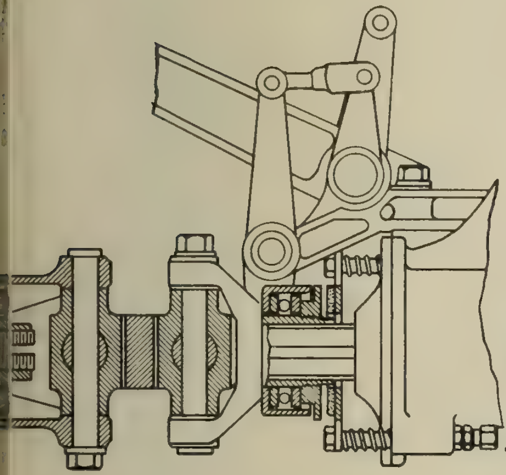
THE 15

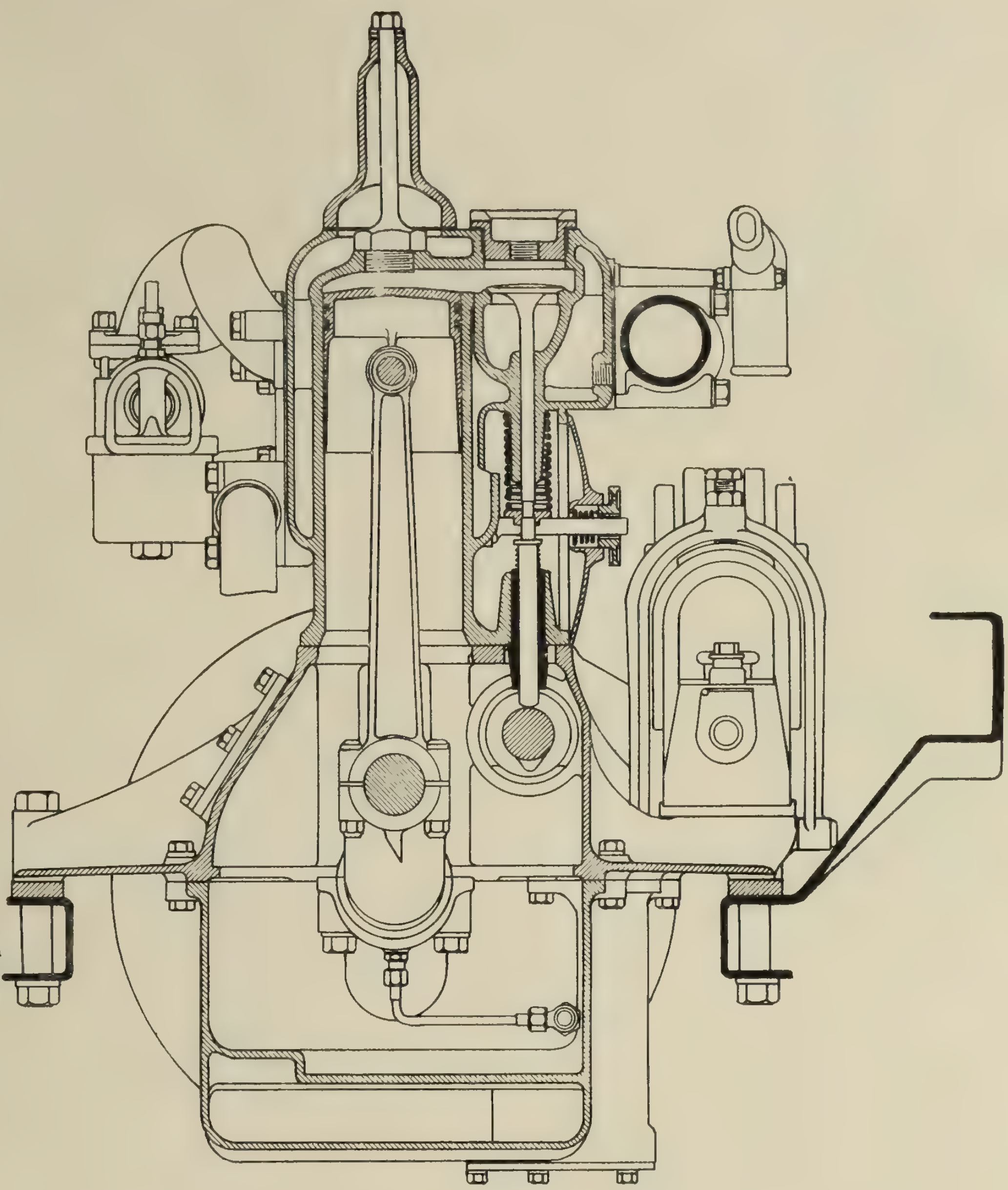
SQUIRE

CA

Fig. I.

H.P. STRAKER-
ENGINE AND
BURETTOR.





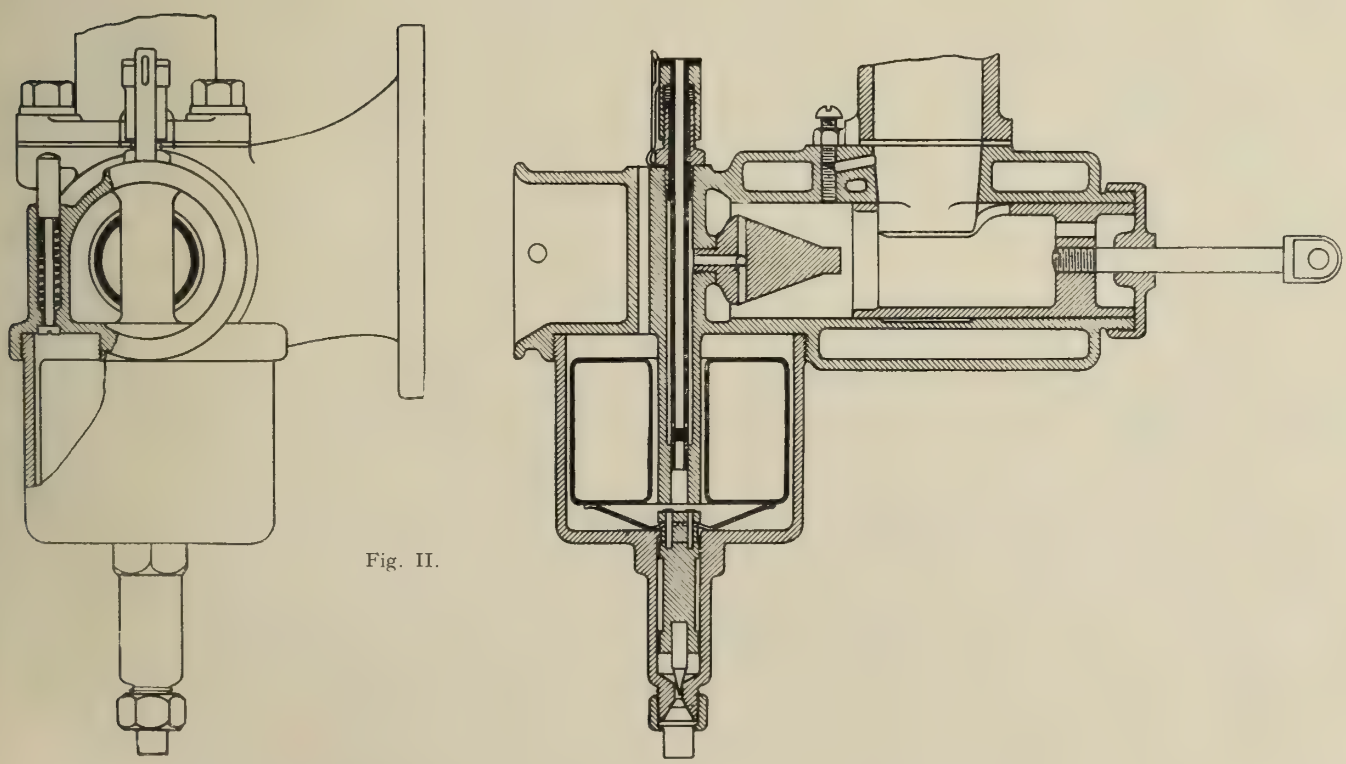


Fig. II.

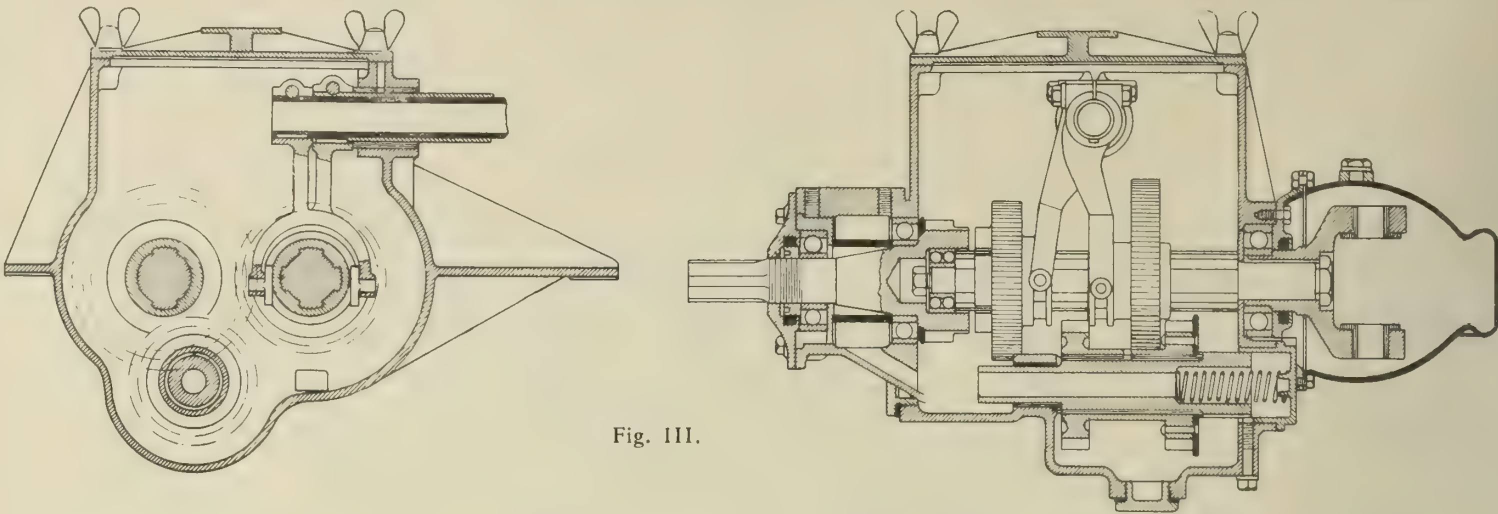


Fig. III.

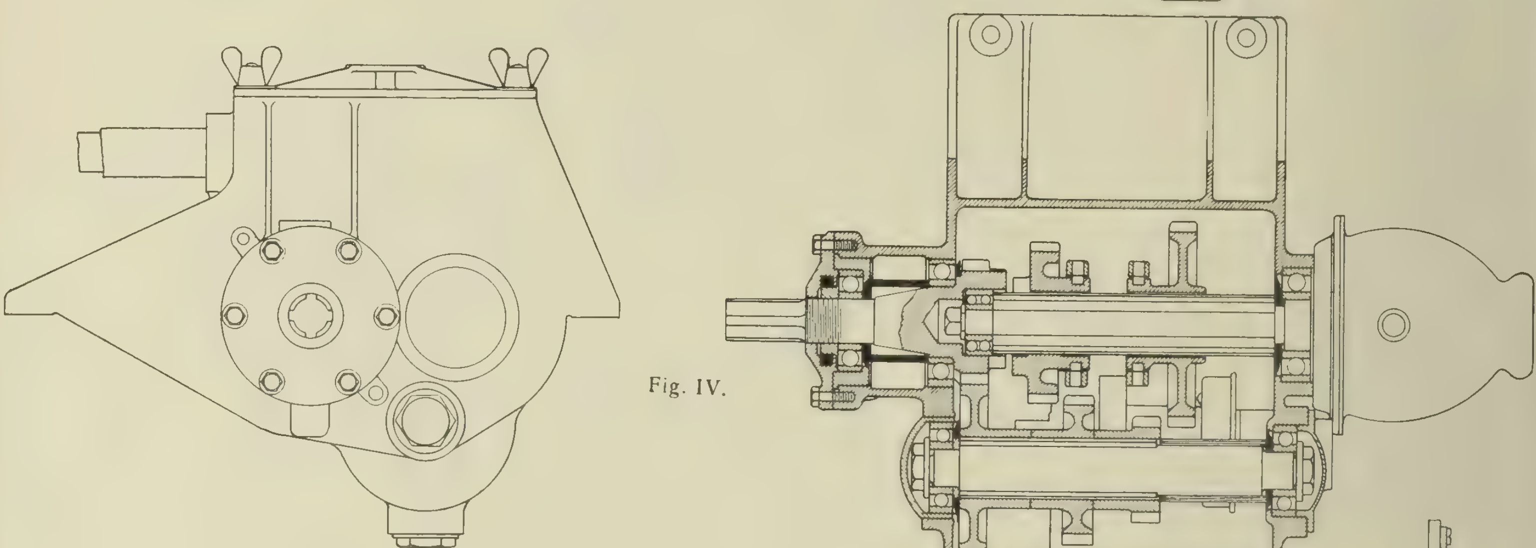


Fig. IV.

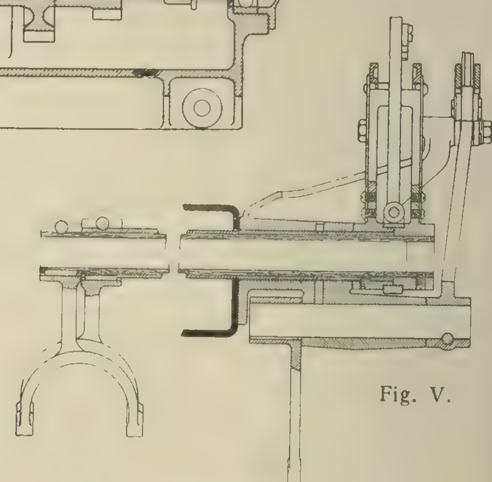
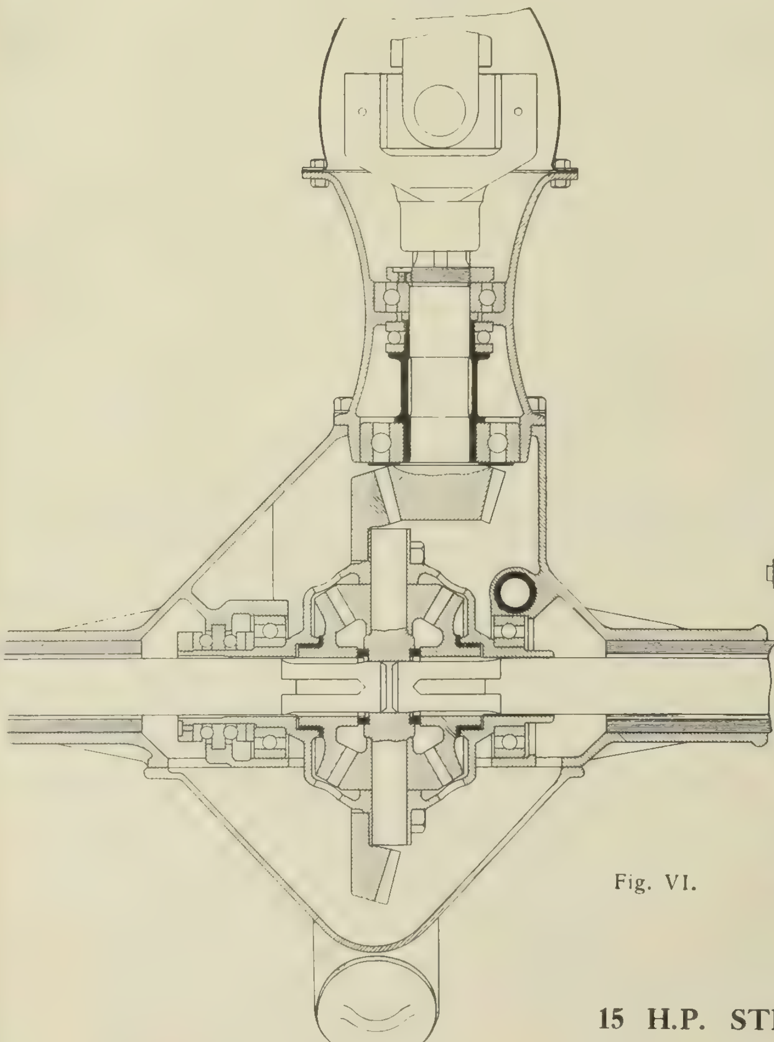


Fig. V.



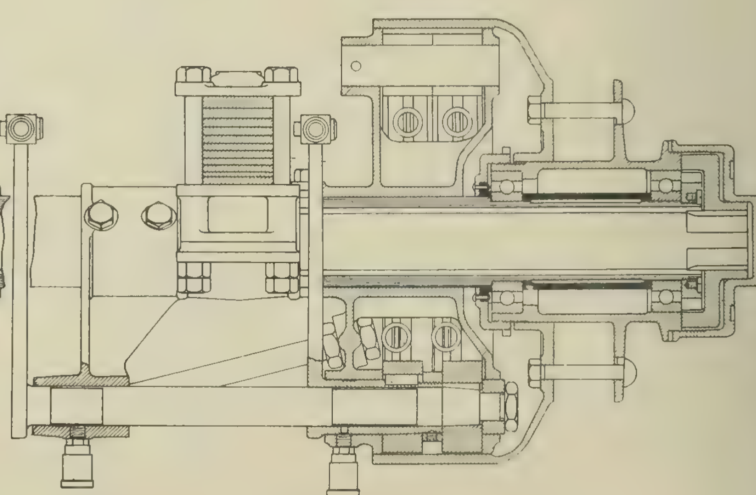


Fig. VI.

15 H.P. STRAKER-SQUIRE TRANSMISSION DETAILS.

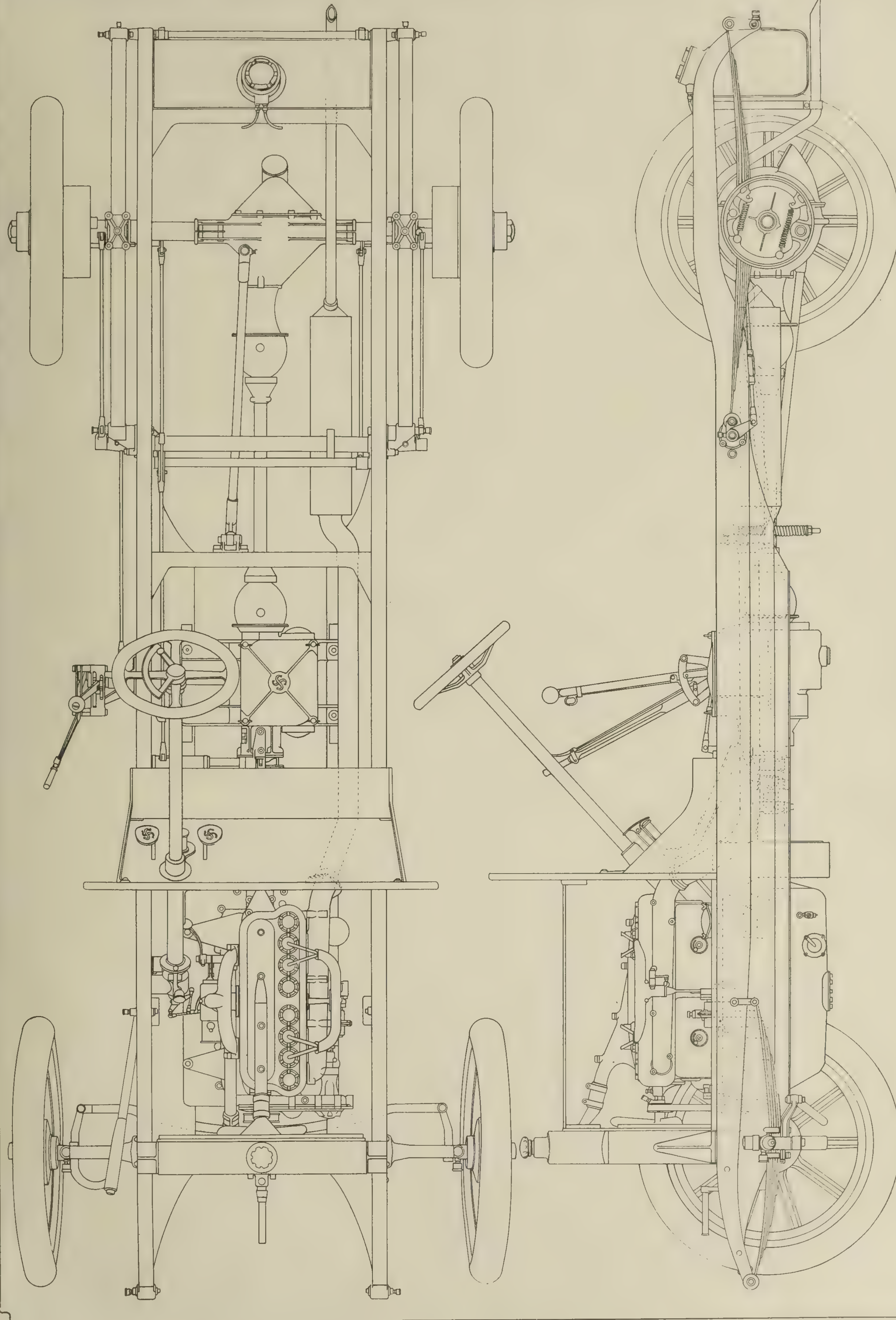


Fig. X. THE 15 H.P. STRAKER-SQUIRE CHASSIS.

that it is always parallel to the collar with which it engages. Below the gear change lever, and in a lug on the same bracket, the hand brake lever spindle will be observed. This is a somewhat unusual but very neat method of assembling which, however, would be better for a separate lubricator instead of reliance being placed upon the surplus oil from the gear actuating tubes, especially as both bearings are of great length.

As the shaft drive is open, the forward universal coupling is encased in a neat spun brass covering bolted to a flange on the gearbox. Lubricant is inserted through an orifice at the top of the case, which is normally covered by a set screw and, unless overfilled, very little oil escapes. At the tail end another flexible joint is provided with a similar brass cover lubricated however, from the bevel case. This joint is free to slide bodily on the splined shaft, which is coupled direct to the bevel pinion, an arrangement that is likely to produce undue wear in course of time, but is rendered necessary by the use of the springs as radius rods.

Fig. V. is a section showing the arrangement of the rear axle that is of the floating type, in which each driving shaft can be removed through a wheel hub, allowing the bevel and differential, together with the bearings, to be removed as soon as the rear cover is detached. The petrol tank has been given sufficient clearance to allow this to be done without first moving the tank. The bevel gear ratio is 4.0625, and both bevels are manufactured from Ubas steel. Large differential pinions of Kirkstall J5 steel are used, meshing with bevels running in bushes fitted to the casing. The thrust bearings, both for the bevel pinion and worm wheel, are of large size, that for the bevel having a double row of balls, and the bushed holes seen drilled in the casing above the right hand journal bearing accommodates the torque rod pin. An interesting and unusual method is employed for gripping the axle tubes to the casing, there being split sockets into which the tubes are fitted, the whole being then secured by two large bolts (see sketch, Fig. VI.). It will be noticed in Fig. V. that the journal bearing immediately behind the bevel pinion has a thin washer to prevent excess of oil find-

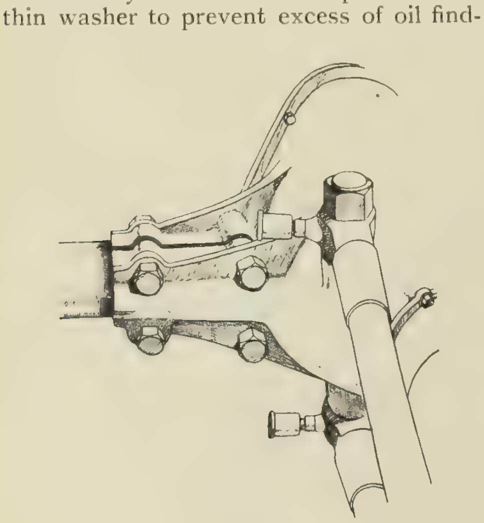


Fig. VI.

ing its way out through the forward universal joint casing.

Both hand and foot brakes are situated in the cast drums on the rear wheels, and both are of the cam actuated expanding type with coil spring returns. The situation of the hand lever operating one pair has already been described; from this a rod is carried to the cross tube on which the compensation gear is situated, and thence two rods, provided with turnbuckles for adjustment, are taken to levers on the cam gear. For the foot brakes the pedal is directly coupled to the balancing device, seen in Fig. VII. by a long rod, and thence to the rear wheels in a similar manner to the hand brake actuation. Both these brakes are capable of

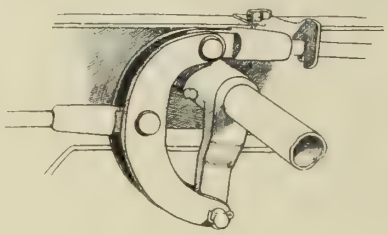


Fig. VII.

locking the wheels while their normal action is smooth and free from chatter or squeak, although a steel on cast iron contact is used.

Front and rear hubs are both provided with large ball races set some considerable distance apart, thus in some way excusing the absence of thrust bearings. Steering is by worm and worm wheel, the wheel being designed to give four different

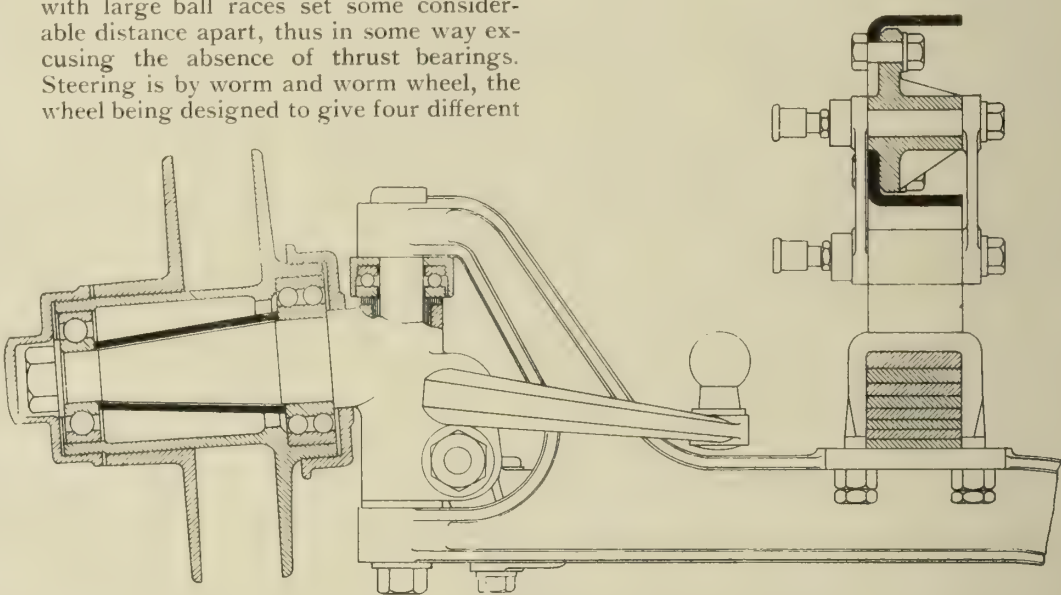


Fig. IX.

positions by meshing one of four keyways with a small pinned feather on the shaft taper, so that a certain amount of adjustment for wear can be effected. As the tie rod is placed behind the front axle it has a set to clear the underscreen, and the shape of the rod is clearly shown on the chassis plan, Fig. X. The tie rod arms are stamped separately and are bolted to the stub axle pieces. Fig. IX. shows the form of forked axle employed, and also the separate ball ended steering arm fitted to the same lug as the tie rod arm. Thrust bearings are, of course, provided for the swivel pins and fitted with a screw-down grease cup not shown in the drawing. A noticeable feature of the front wheel is well brought out on the same drawing, namely the double ball race at the inner side. Each front wheel is inclined, so that the tread shall be reasonably near to the vertical axis of the steering pin.

Semi-elliptic springs are used, which give remarkably good results on the road, in fact, the car is quite unusually comfortable. The approximate lengths are, rear 48 ins. and front 26 ins. In connection with the springing further evidence of careful design occurs, for the rear springs are held in special clips which have plates at the top and bottom held together by four bolts. Thus the rear springs are not drilled, thereby avoiding a fruitful source of trouble. Fibre packing pieces are fitted between all the springs and their bed plates. Two U-clips hold the forward suspension in the ordinary manner, and in this case a bolt holds the spring leaves together. As the rear suspensions act as radius rods they are fixed at the forward ends and hung in the ordinary H shackles at the rear. All shackle or spring holding pins have grease cups fitted.

Control is by foot throttle, which only controls the range beyond that set by the lever and segment on the steering wheel, and the firing point is also hand controlled by means of the magneto contact advance. Fig. XI. is a sketch of the throttle pedal connections. Petrol is delivered to the carburettor by a low exhaust pressure, which seldom exceeds 2 lbs. on the gauge. Although it might be imagined that the particular type of carburettor used would need more than usual, yet in practice there is never any difficulty in maintaining the supply.

The frame is very gradually upswept over the rear axle, while in plan it is quite straight, although a turning circle of 37 ft. is allowed for. Rivets fix the dumb irons in position at either end and there are four cross members, three of angle steel and one tubular, the latter accommodating the hand brake gear. There is

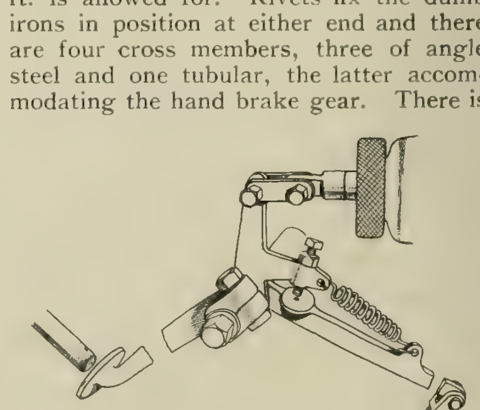


Fig. XI.

also a steel tube connecting the rear spring-hanger brackets. The peculiar "underframe" construction is adhered to until well behind the gear box, and on a

level with the second cross member, rendering the side members particularly stiff.

At the front the angle steel member is dropped to accommodate the vertical gilled tube radiator, which is provided with a water tank of large size, standing some distance above the bonnet level.

On the road the car holds the ground remarkably well, even at fairly high speeds, and the engine is extremely flexible. The only period at which there is any sign of hardness is when the engine is revolving at its highest speed, otherwise it is smooth and very powerful. No objectionable noise is caused by the tappets, that is to say, although they are audible their sound is very far from insistent. Altogether the engine is scarcely noticeable when the car is moving at moderate speeds.

In the open country there is no doubt that a fourth speed would be a great advantage, and its absence is to be regretted. As has already been remarked, the springing is very good, and the absence of rattle or small noises is quite noticeable.

One point in the design which might perhaps be criticised is the number of rods on the rear axle, which move on different centres, thereby entailing unnecessary wear. Although this is quite a minor point, it would be better were it possible to place them all on the universal joint axis. On the whole, every part bears evidence of careful study with a view to evolving a design which shall be both accessible and efficient, and the design may certainly be said obviously to be the work of men who themselves have had long and varied experience, not only of car making, but of car using, too.

AEROPLANE SUPPORTING SURFACES.

A brief review of some of the prevalent methods of construction.

By W. G. Aston.

BEFORE proceeding to a detailed account of the most notable type of plane construction at present in use, it is advisable to consider some of the elementary desiderata connected with their design. After a careful examination of nearly all the existing types of aeroplane the writer has come to the conclusion that some of these desiderata, although so elementary, have nevertheless succeeded in escaping the attention of builders whose previous experiences ought to have taught them to know better, hence he makes no apology for mentioning a few of them here, although they are patent enough.

(1) Lightness. In addition to the obvious advantage obtained from making the supporting surfaces of the least possible weight, *i.e.*, increasing the fuel supply to be carried or enabling passengers to be conveyed, there is the secondary, though scarcely less important one, that it is important to maintain the greatest possible proportion of the total weight of the machine as a compact mass about the centre of gravity. The disposition of weight in this manner has an important bearing upon stability in flight, as can very easily be proved by a few experiments with a simple glider composed of a strip of paper loaded at the middle of one of its longer sides with a shot. The behaviour of this rudimentary aeroplane when the equivalent weight of the shot is attached symmetrically all along the front edge is sufficient to show that concentration of mass is in such a glider of paramount necessity, and hence, at any rate, advisable in the controlled power-driven machine.

(2) Rigidity. On basic theoretical grounds an aeroplane supporting surface should be absolutely rigid in all directions otherwise unequal resistances and loading must occur with a consequent application of unnecessary strain to the construction, but for practical purposes there is a decided limit to the amount of rigidity, beyond which it is inadvisable to go, in so far as this rigidity applies to the *whole* of the supporting surface. Thus, if the machine be controlled by wing warping it is clear that a certain modicum of flexibility must be incorporated to allow for this warping to be carried out without damage to the fabric. Most designers are content to allow this factor to be furnished by the natural experimental error in construction and the natural "give" of the materials employed. At the same time there is another weighty practical disadvantage in rigidity carried beyond a certain point; namely that since the machine in landing and alighting must inevitably be submitted to heavy shocks, it is a matter of absolute necessity to provide a construction which has sufficient elasticity to bend without breaking. At last year's Aeronautical Show at Olympia there was exhibited an extremely well constructed machine which appeared to be almost unnecessarily strong, yet it was braced up to such a pitch of rigidity that its first run on a comparatively smooth piece of ground was sufficient to cause its disintegration. It is almost impossible to lay down any fixed law as to the amount of flexibility desirable in a supporting surface, as it appears to be, if not a matter of instinct, at least a result of experience which must differ with every make of aeroplane.

There are, too, certain parts in the wing construction which should purposely be provided with a certain amount of latitude for spontaneous motion beyond that required for avoiding damage upon meeting with shocks. There seems to be no room for doubt that if the trailing portion of a wing be flexible, it has a better chance of maintaining its stability in the face of a gusty wind than one of perfect rigidity. In the event of a current of air striking one wing of an aeroplane to a greater extent than the other, then, if these wings be partially flexible, that upon which most pressure is exerted will automatically tend to "flatten out" and so equalise the lifting reaction over the whole of the sustaining surface, whereas if both wings were highly rigid, only a dexterous movement on the part of the pilot could avoid a dangerous loss in equilibrium.

(3) Simplicity in construction. Until the aeroplane reaches a stage far beyond its present liability to damage it is most important that the building of surfaces be kept as simple and with as few parts as possible, so as to localise the result of a "smash" to the greatest possible extent. At the present time the development of the aeroplane depends largely upon the patronage which it is capable of commanding, and it is therefore most desirable that the inevitable breakages which must for some time to come characterise the performances of aeroplanes, should be prevented from being far reaching in their effects and consequently expensive.

(4) Of minimum head-resistance and maximum lifting efficiency. These points refer especially to the plotting out of the curves of the surfaces and to the depth of the plane at various points, to its plan form, and to the angle of incidence at which it is designed to be set. It is clear that if the plane is to be capable of imparting a uniform downward acceleration to the air which it meets with its centre line, it must conform to the conditions of a hyperbola, whilst its sectional form must be such as to comply also with the conditions of least resistance. In other words it must be as nearly as possible of streamline form. As to the hyperbola the writer has not as yet been able to find that any designer has made this curve the basis for his wing formation; for the most part the wing curvatures are plotted by eye, and eye alone, and more with regard to its adaptability to easy construction than to anything else, but in some cases formulæ of questionable value and of mysterious antecedents have been evolved. One remarkably successful designer employs scissoid curves as his basis, but in apparently every case the construction is built upon purely empyric foundations, a fact which is unfortunately due to present lack of definite knowledge upon the subject. Whilst admitting the necessity of approximating to streamline form, it is certainly remarkable that in more than one notable instance this form has been mutilated with apparent impunity, and one can therefore only fall back upon the consoling belief that almost anything will fly, however wasteful of power.

Plan form includes both aspect ratio and contour. A high aspect ratio is certainly desirable, and it may be assumed it must be infinite in order to obtain cent. per cent. efficiency. On the other hand practical considerations demand its being maintained at a moderately low figure, which however is higher for biplanes than for monoplanes, and it is decidedly the exception to find anything in excess of six to one. Head-resistance is also largely dependent upon the combination of plan form with the variation in angle between the centre of the plane, *i.e.*, the extremity of one *wing* and the tip. The expediency of so tapering off the wing tips that the air is smoothly and progressively deflected appears to be at present frequently overlooked and accounts to a large extent

for the very decided inefficiency of certain prominent machines. In order to provoke the minimum of disturbance about the tips of the sustaining surface it is important that the distance which separates the undiverted "strata" (*sic*) of air, from the strata subjected to the maximum deflection should be as great as possible, otherwise a series of eddy currents giving rise to waste of energy are produced. The best wing formation is therefore one in which the width, the depth, the "camber," and the angle of incidence decrease towards the tips.

In the matter of wing construction the biplane type of machine offers, with its simple girder formation, greater facility than the monoplane, and at the same time possesses the advantage that the foundation of the sustaining surface can be made extremely lightly as it is capable of being braced by tension wires both against head resistance and against the lifting action. In the monoplane, unless of the Valkyrie or "tail-first" type there is nothing in front of the main plane to which it can be braced, and at the same time there are certain practical difficulties in the way of putting the propeller behind the plane to as to allow some form of horizontal king-post to be erected. In consequence it is necessary to make monoplane wing framework of great strength in the horizontal plane, even more so than in the vertical since in the latter case a mast above and below the centre of the surface can easily be arranged and from this wire supports can be taken. There is however, no reason why the rear spar should not be triangulated by wires, either inside or outside the wing itself to the inner end of the front spar, yet this wire provision is very rarely carried out in spite of the fact that its introduction is an extremely simple matter.

Rigidity and complication have been referred to above, but it is perhaps as well to point out that it is difficult to trace any material advantage resulting from elaboration of means for keeping the fabric covering at all times at the predesigned curvature. So long as the supports of the fabric are not too far apart as related to the fabric's strength, and so long as the fabric be stretched to adequate tension, it appears as though the curve can be left to look after itself. As cases in point the Antoinette wing may be cited in comparison with that of the Bleriot. The former comprises a most careful construction designed to remove any possibility of the plane section becoming deformed in any circumstances, whereas the latter is conceived on much simpler lines, and the fabric having fewer points of support must undoubtedly tend to a slight "bagging," especially after having been in use some time. Yet as a supporting surface purely and simply the Bleriot wing is undoubtedly superior to that of the Antoinette, though this result may be partially ascribed to the choice of an unsuitable curvature in the case of the latter. Again the Farman construction, which is the simplest possible and allows a very great expanse of fabric between supports, is also superior as a lifting affair to the Antoinette. It is an undeniable fact that the best machines are fitted with the simplest forms of wing construction with very few exceptions, whilst, of course, cheapness in production is a quality which is of very great moment, at all events from the commercial aspect.

Much depends upon a careful choice of materials, especially as regards the total weight of the finished supporting surface. In this respect there is rather more room for improvement than there ought to be, as—this is more marked on Continental than on British machines—the fact of the structure being out of sight and out of mind has led builders rather to skimp the internal work. This is frequently of the very roughest description, abounding in cracks and haste-marked all over, whilst much of the wood used is totally unsuitable. For main spars (*i.e.*, longitudinals) Honduras mahogany or seasoned English ash are most suitable, and for long spars require to be very carefully selected. If the ribs, or "formers," are to be solid, spruce is a light, easily worked wood well adapted for this purpose, and may also be used for the narrow longitudinal stringers introduced to preserve the curvature of the fabric. These longitudinal stringers might, however, be replaced with advantage by a larger number of piano wire strands either nickel-plated or coppered to prevent rusting, and it is rather surprising to find that this has not yet been employed on the main wings, though it has in one or two instances been used on the smaller planes, such as elevator and rudder and tail.

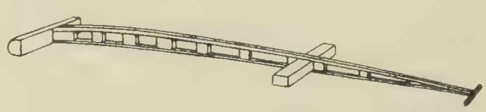


Fig. I. Wright.

The writer is decidedly not in favour of those methods of building up ribs into I sections of thin wood, as adopted on the Bleriot and some other machines. It is quite wrong to treat wood as if it were metal, especially when it is got down to small size, owing to the difficulty of making satisfactory joints without splitting the wood badly. Long spars of considerable initial thickness may, however, be channelled out and are all the better for it, but such a method of construction is not applicable to ribs.

The Wright rib, Fig. I., as used on the original Wright biplane and in subsequent machines by this constructor, has been largely copied, and is generally speaking a good method of building wings. There are two main spars, one of which forms the leading edge, whilst a thin strip of wood is attached to the rear to form the trailing edge. The rib-members are held apart by a number of distance pieces tacked in position, and the resulting sustainer fulfils most of the conditions laid down, especially as regards ease of construction, though it is not so well adapted for use with planes tapered from the centre outwards as this involves the gauging of a very large number of distance pieces. The front block is screwed to the leading edge. No stringers to support the fabric between the ribs are used, and the latter are spaced about a foot apart. The fabric is applied as a double surface.

In the Farman machine, Fig. II., a totally different form of construction is adopted, although here again the disposition of the main spars remains the same. In this case the rib is solid, is attached to the top of the front spar and to the bottom of the rear one. Two surfaces of fabric are used, but the effect of them is to form a single-surfaced plane, as they are stitched together close on each side of the ribs which are thus provided with pockets. These pockets are indicated in dotted lines in the figure. Viewed from in front the planes exhibit a number of parallel ridges formed by the ribs and for these ridges it has sometimes been claimed that they have the advantage of compelling the air current to take a direct line across the plane instead of having a tendency to escape out sideways and

Fig. II. Farman.

promote eddy currents at the extreme tips. In the greater part of the Farman supporting surface the plane ends at the rear spar, and is continued by a controllable flap or aileron corresponding in size to the extension shown, both on these ailerons and on the trailing edge of the fixed plane the trailing edge itself is formed by a steel wire stretched along the ribs The wire is put in tension by the fabric itself, and therefore forms a series of "bows" between each rib. It will be readily understood that between the ribs the section of the Farman planes is by no means of streamline form, as there is a distinct hump both before and behind, but against this it has the advantages of great lightness, ease and cheapness of construction and a very desirable strength.

The Cody rib, it will be perceived from Fig. III., is similar in type to that of the Wright brothers, from which, however it differs in a most important point, namely, instead of increasing in depth between the two spars and having its greatest depth at or about the normal centre of pressure, it does just the reverse, and tapers to considerable shallowness at this point. A line drawn through this point divides the rib into two approximately streamline forms arranged in tandem, and the claim is made that by this simple means head resistance is considerably reduced, and the lifting effect, if anything, increased. These claims, it

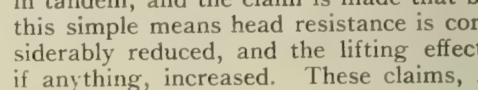


Fig. III. Cody.

appears to the writer, are borne out in practice for the Cody biplane, whilst being of rather heavier surface-loading than the Farman certainly has a perceptibly flatter gliding angle, and in this respect appears to be superior to any other machine yet built. At the same time it is more than 30% faster in flight than the Farman, whilst equipped with a motor of scarcely 20% more power. The members of the rib are united in front by a solid web behind which are a number of distance pieces, and a narrow strip of wood is used for the trailing edge.

The Paulhan plane construction is in a class entirely by itself. There is only

one spar, which consists of a couple of flat ash strips united by inclined ash distance-pieces forming a cellular construction which may of itself have a certain amount of lifting effect. The solid spruce rib is supported, in front only, by a couple of wire stirrups being therefore readily detachable and replaceable in case of damage. The fabric is a single surface provided with pockets for the reception of the ribs, and is laced in front to the bottom member of the spar guide and at the rear to the extremities of the ribs. A good deal of controversy was excited by this construction when exhibited at the 1910 Paris Aero Show,

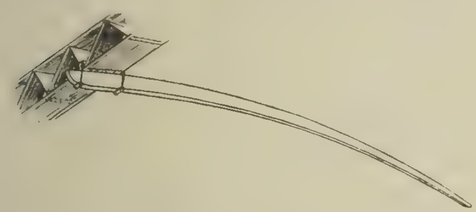


Fig. IV. Paulhan.

but the machine has since more than vindicated its designer. Two very notable features are the fact that with the rigid girder no intermediate vertical struts between the planes are required, and that when it is desired to decrease the supporting surface to enable the highest possible speed to be obtained, this can be easily done by substituting a narrower piece of fabric for the original one, which advantage is possessed by no other machine at present capable of flying. The ribs are cut (to shape) out of silver spruce, and are quite springy towards the rear, so that the plane has a decidedly flexible trailing edge.

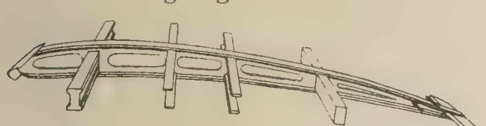


Fig. V. Bleriot.

A far larger variety of methods of construction is to be found in the monoplanes. Fig. V. illustrates the Bleriot design. The spars are of ash, the front being channelled throughout its entire length, and the rear one plain. The rib itself if of **I** section and consists of white pine, the central web being about 3-16 inch in thickness, and considerably weakened by the wholesale "piercing" to which it is subjected. The distance pieces are thus very small and are in addition taken across the grain so that they have very little strength to withstand any sudden shock. The leading edge is of aluminium sheet, and the trail-

Fig. VI. Radley.

ing edge a thin strip held between the termination of the upper and lower flat members of the rib. Stringers are also employed as shown. The tacking on of the thin flat members results in unavoidable splitting of the vertical web and in process of manufacture it is more than likely that the distance pieces get split across. Generally speaking, however, the Bleriot wing seems to be of adequate strength. Mr. Radley built a pair of wings for his No. XI. Bleriot in which he sought to overcome some of the disadvantages mentioned above. The spars were only channelled between the ribs, not for their entire length, and the rib was consequently provided with a stronger attachment. The piercing of the vertical web was more carfully carried out, and this member was also slightly increased in thickness. An extra pair of stringers between the rear spar and the trailing edge were added, and the total increase of weight was about 20lb. per wing, but the increased strength was quite out of proportion to the extra weight involved. Even so however, Mr. Radley's method is open to objection, although this is perhaps as light a form as can be devised for machines of moderate spread.

In Fig. VII the Howard Wright construction is shown, which is characterised by its general robustness of construction, and by the fact that the spars are reinforced with 20 gauge hard rolled steel strip passing their entire length both top and bottom. The ribs are solid

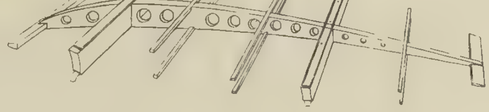


Fig. VII. Howard Wright.

and are cross-jointed into the ash spars. This involves practically no sacrifice of strength, as the joint between spar and rib is in compression, whilst the opposite effect on the spar is resisted by the steel strip in addition to the ordinary staying wires fitted to receive the larger proportions of the lifting strain. The supporting surface made in this manner is certainly heavier than that of the Bleriot, but is, on the other hand, a good deal stronger and undoubtedly easier of construction and repair.

The Poynter construction is illustrated in Fig. VIII., and exhibits several individual characteristics. The spars are composed of built-up girders, and the front one, which is much the stronger of the two, is arranged so as to be as nearly as possible coincident with the normal centre of pressure. The rib is cut out of a plank of American white wood (this

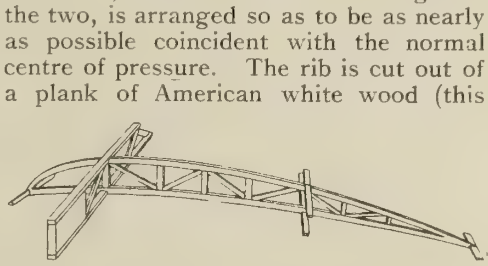


Fig. VIII. Poynter.

material could easily be improved upon) and is reinforced by a number of vertical and diagonal struts. The front and trailing edges are formed by wooden strips.

The small Poynter ribs (those nearer the wing tips) are made as shown in Fig. IX., being cut out of a plank in one piece. The disadvantage of this idea is similar to that cited in the case of the Bleriot, and lies in the employment of weak cross-grained distance pieces which are extremely liable to fracture. In the Poynter this disadvantage could be practically overcome by adding vertical strips of straight grained wood each side of these distance pieces, so as to prevent them having to sustain even a tensional pressure.

The Martin-Handasyde (Fig. X.) method is somewhat similar to the Poynter, except that the sustaining effect is carried by two main spars of considerable depth consisting of channelled beams of ash. These are arranged as nearly as possibly equidistant from the centre of pressure. The large indirectly-supported area of plane behind the main spars allows a certain degree of desirable flexibility.

A view is given of the Petre rib in Fig. XI. (this machine is now extinct), for the purpose of illustrating the original method of construction employed in

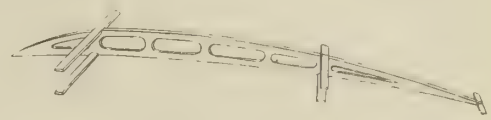


Fig. IX. Poynter.

building up the wings. The main spar consists of a couple of horizontal ash planks stayed by diagonal cross-pieces. There is also a rear span which has however very little load to sustain, and acts principally as a stringer. The rib is entirely constructed of ash, the strips being screwed together and the labour involved is therefore somewhat heavy, though the resulting wing is both light and strong.

The Antoinette wing, a sketch showing the construction of which is given in Fig. XII., is far and away the most elaborate affair of its kind which has yet appeared. It also seems likely to remain so since the complication is so unnecessary. The sketch shows only how it is constructed, and makes no attempt to portray its real appearance—which is practically that of a cobweb of tiny girders. The main spars consist each of four rails, two above and two below

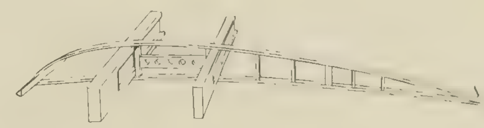


Fig. X. Martin-Handasyde.

united by diagonal ties fixed in the space between the upper and lower pairs. The principal ribs are composed of separate square-sectioned strips, strutted to one another with a number of vertical and diagonal braces. In the sketch two sets of stringers only are shown, but in point of fact there are actually six, there being two sets between the spars and two more between the front spar and the leading edge. These stringers are braced rigidly to one another, and have their upper and lower edges serrated as shown. On the top of the teeth so formed are fixed a number of false ribs, occurring about every inch and a half. There are also at intervals horizontal diagonals connecting all these parts up to the main spars, but these are not shown in the sketch. The result of such a meticulous and alto-

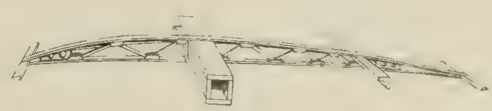


Fig. XI. Petre.

gether unnecessary construction is, as might be supposed, a prohibitive cost which more than anything else has prevented the Antoinette form becoming a commercial proposition. Repairs to such a wing are almost impossible, and

damage is rendered all the more likely by its extreme rigidity. It is in fact neces-

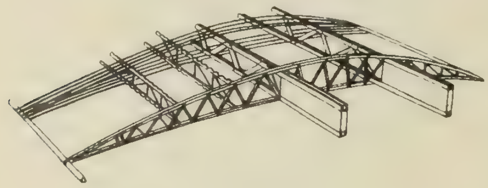


Fig. XII. Antoinette.

sary to gear down the wing warping control very considerably in order to get any movement of the wings within reasonable limits in the force required. In two fatal accidents the rigidity of the wing construction seems to have been to blame, so that in some respects the complication is not only futile but actually a source of danger. Exclusive of screws and nails there are slightly more than 2,000 parts in each Antoinette wing (*i.e.*, 4,000 for the whole surface as compared with the 38 parts for one Farman deck) including wooden parts and aluminium flitch plates, etc. It is not surprising, therefore, to find that a wing costs well over £100.

It should be noted that both the upper and lower curves of the Antoinette plane are arcs of circles. This results in an extremely sharp leading edge, which is possibly provocative of wasteful eddy currents.

Fig. XIII. is a view of the Santos-Dumont arrangement, as used in the "La Demoiselle" machines. This is a very excellent type of wing construction, but is only suitable for machines with a comparatively small spread. There are two main spars of ash, the double ribs being of bamboo and the leading and trailing

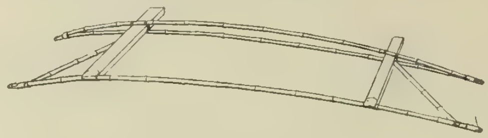


Fig. XIII. Santos-Dumont.

edges of steel wire. For this purpose the extremities of the wings are triangulated up to the spars as shown. The fabric is folded on the front edge, and the two surfaces are then laced up to the wire at the rear. The result is that both edges are strained into a series of "bows" which is neither expedient nor of good appearance, but otherwise

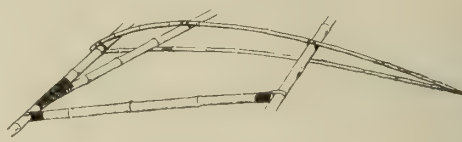


Fig. XIV. Weiss.

the Santos construction leaves very little to be desired, especially in point of lightness.

The Weiss supporting surface, Fig. XIV., is in its general conception entirely different from that of any other machine, and as it will be described in a later article it is unnecessary to go further into it at present than merely showing how it is made. There are two main bamboo spars which are united by a diagonal spar. The upper members of the bamboo ribs are attached to these three spars and the fabric is sewn to the ribs, both above and below. All joints are made by lashing with tarred string

PROBLEMS RELATING TO AIRCRAFT.

Being extracts from a paper given before the Institution of Automobile Engineers by Mervyn O' Gorman.

With respect to aeroplanes there is a certain similarity in the devices used at present by most makers to get some degree of lateral stability when flying, turning and gliding, but there is apparently greater diversity in their ideas as to longitudinal stability, and the best position of propellers. These two features therefore are used as a basis for a suggestion for a classification—

1. *Class S.*—Those of which the main wings are preceded by a small plane which is more intensely loaded and succeeded by the propeller (Fig. I.).
2. *Class B.*—Those of which the main wings are preceded by the propeller and followed by a smaller plane (or planes), which is more lightly loaded, if at all (Fig. II.).
3. *Class F.*—Those which have the main wings followed by a smaller plane more lightly loaded, if at all, as in 2, but the propellers of which are placed between the main wings and the tail plane (Fig. III.).

Those familiar with the machines of to-day will include in *Class S.* the latest Voisin, Valkyrie, Cody, Clarke and Santos Dumont very early machines, &c; in *Class B.* the Antoinette, R.E.P., Bleriot and Avro, &c.; in *Class F.* the Voison, Farman, British and Colonial, &c. No fundamental importance need be attached to the classification, but all attention must be given to the method adopted in each case to secure fore and aft stability, which is throughout by some attempt to get a Vee between the surfaces back and front.

Since a plane carries a weight by throwing air downwards, its general slope must be with the trailing edge downwards, and the more heavily it is loaded per square foot, the more steeply must it slope downwards to get the adequate reaction for any one speed of travel. Accordingly, as the forward and back planes of one and the same machine must move at the same speed, it is not possible to get the desirable Vee or inclination between them without loading one more intensely than the other. Not only so, but the front plane of the two, whether it be larger or smaller, must always be the more heavily loaded if the Vee is to have its opening to face skywards. Any more lightly loaded auxiliary plane, such as is placed in front of the carrier planes or wings, diminishes the longitudinal stability of the combination by giving rise to an inverted Vee. This is instanced by the front elevator planes of the Farman—the effect of which has to be counteracted by increasing the size of the tail planes. (See Fig. XI.).

Machines Veed fore and aft, that is, those having the front plane set at a greater angle of incidence than the back plane, are, to some extent at least, stable in the fore and aft direction. This is shewn graphically on the accompanying diagram Fig. IV. together with Figs XI. and XII.

In the upper case Fig. IV. it is assumed that the front plane is half the size of the back plane, but that it is set at an angle of incidence of eight degrees against an angle of four degrees for the back plane. For small angles the lift is proportional to the angle, and consequently the front plane, being at double the angle of the back plane, will lift twice as much per square foot. The centre of pressure when the aeroplane is horizontal will consequently lie half-way between the centres of pressure of the two planes, and the centre of gravity must therefore be adjusted to fall at the same point.

In the centre diagram the aeroplane is imagined to be tilted up at an angle of two degrees, and consequently the front plane will make an angle of incidence of ten degrees, and the back plane one of six degrees, that is to say, the front plane will have increased its lift by a proportion of ten to eight, that is of five to four, while the back plane will have increased its lift in the proportion of six to four. The centre of pressure has consequently travelled back, and will now be behind the centre of gravity, and thus a righting couple will be applied, which will tend to move the aeroplane back to its horizontal position. In the third case the aeroplane is imagined to be inclined downwards at an angle of two degrees. In this case the lifts of the two planes will be respectively three quarters and two quarters of their original lift, consequently the centre of pressure will have moved forward, still giving a righting couple. It must be noted that the angles chosen here are arbitrarily selected because they are easy to handle. In practice the angles are more like 9 and 13 degrees. It can readily be seen that if the front plane is at a lesser angle of incidence than the back plane, the reverse effect will take place.

If experience up to the present had shewn definitely any superiority in one of the classes over the others, it is not improbable that this division would not have been made owing to the extinction of the inferior classes. A few salient facts resulting from each combination with the propeller position may be noted, remembering that wherever a plane is on the suction side of the propeller, i.e., in front of it, the operation of the engine does not make much difference to the amount of air thrown down by that plane, that its lift is but little affected by the engine stopping. For example, it is found in *Class S* that when the engine stops the machine automatically adopts its proper angle for a glide, and indeed cannot easily—with the engine stopped—be made to glide too steeply so long as the Vee is maintained. (See Fig. XI.) If, however, any plane which is in the full propeller blast be a loaded plane, that is, one given a slope for the purpose of lifting part of the weight (and usually of curved profile), such a plane loses part of its normal lifting power in proportion to the falling off of air velocity past it, due to the stopping of the engines. As an instance, every Farman machine drops its tail the moment the engine stops—a movement which requires that the airman shall *at once* give a diving movement to his elevator planes lest he begin to glide backwards.

In the case of *Class B.*, the Bleriot machine, which usually has a lifting tail, both the main wings and tail ride over the propeller blast, therefore the stopping of the engine diminishes the lift of both—with the result that the speed of travel downwards with the engine stopped, tends to increase a little over the normal speed—a matter which may involve alighting somewhat more rapidly from a glide, but the tail does not drop. In the case of *Class S.*, the gliding conditions appear to be very favourable with a good design of machine. The head resistance can be made low, and I rather think the small number in use is chiefly due to its not resembling a bird, a fact which need have no particular relation to its merits. For the moment it is fashionable to have a tail.

The History of Stability.

I am not in these notes concerned with the claims of inventors to priority, but wish to consider for the moment the doings of builders merely as experiments on stability, and not as attaching prime importance to the date when the experiments were made, and accordingly I have not attempted to verify exact dates. I neglect the early trials of Lilienthal (1886), Pilcher (1892—93), Ader (1897), and others, and note that the first successful Wright aeroplane was tailless. It was evolved from a biplane glider with a rudder (1901) and elevator. I do not know whether a tail, properly so called, had been seriously considered since the Wrights found so much trouble with the breaking of their rudder by slipping back on to it that they hinged it vertically. It was on the results of their gliding tests that they decided not to use a tail, and hence their first power machine was a biplane with no tail, but having a plane in front which acted as an elevator, and which did no weight bearing. (Fig. V.). Such a machine has no automatic stability. It was solely the skill and activity of the man which prevented tipping forwards or backwards, and stability in these direc-

tions had to be attended to ceaselessly. Lateral stability was secured by a system of warping one wing and simultaneously counter-warping the other; this was only slightly automatic, due to the intentional loosenesss of the girder work. This was the state of progress when Wilbur Wright arrived in France (August, 1908), where the reports of his doings which had come from America had incited others to produce notable results, though the study in France up to that time had moved on somewhat different lines.

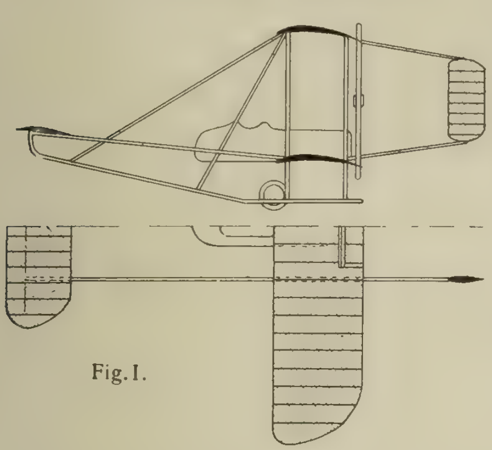


Fig. I.

The first to make any mark in actual flying there was Santos Dumont (1906). His biplane was in some respects the same as the Wrights', but the wings were set at a Vee to one another, that is, a dihedral angle as a help to lateral stability. More important than this was the fact that the front elevator was fairly large, and, unlike the Wright elevator, was set to bear a part of the weight (Fig. VI.). This introduced an advance in that it provided a fore and aft Vee for longitudinal stability, an important departure from that of the Wright Bros., whose front plane was merely directional, and was at times set at a negative angle of incidence. Dumont paid no heed to wing warping, and he apparently recovered his balance by a process the reverse of what one does on a bicycle, namely, by steering away from the side he was falling to. This will be explained later when turning is dealt with.

The first machine that flew for any considerable distance in France was that made by Voisin and flown by Farman in 1907, and this was a biplane which was not boxed in, with a tail which was set to take a part of the lift (Fig. VII.). This formed a shallow Vee with the main wings, and therefore helped towards fore and aft stability. It was also fitted with a front

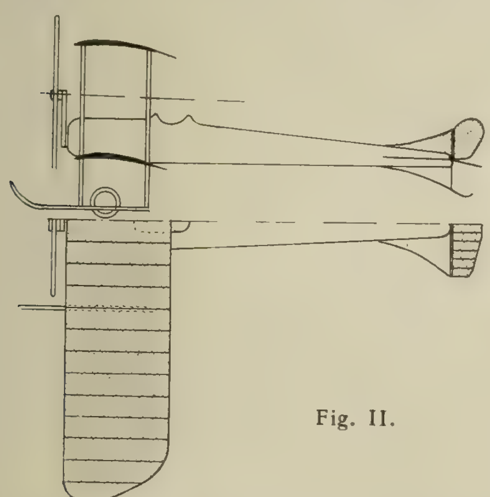


Fig. II.

elevator placed so very close to the main plane as not to exert an important effect on stability. Voisin then very seriously tried the boxed-in ends; he left them in 1910, learning the lesson from his own disciple Farman, but he returned to them in 1911. The Voisin plane was also, like a bicycle, kept from tilting sideways by the use of steering rudders until 1910. The steering method has advantages, and is freely used by all aviators on all machines when they have acquired a little skill.

The effect on France of Wilbur Wright's visit was the adoption there of either his system of wing warping or else the use of the flaps or ailerons, which are inferior in efficiency and speed of response, but which answer the purpose of raising the wing of which the aileron is lowered. The drawbacks of ailerons are touched upon later. Take the Voisin biplane, omit the side panels, give it ailerons, and we have the Farman machine (Fig. VIII.). This was done by Henry Farman in 1909. The elevator is farther in front than that of the Voisin, and not being set to form a Vee with the main planes, it tends, so far, to reduce longitudinal stability, an effect which is countered by increasing the size of the tail, which does form a Vee with the wings. Later, in addition to the front elevator, a flap was put on the tail (1910) to serve as an additional elevator (Fig. III.).

It is a question whether the Wrights learnt in Europe the practicability and advantages of employing the Vee, but it was after their visit here that they used a Vee relationship between the wings and a tail which has been added. It is particularly interesting to note that their front elevator has been abandoned, thereby avoiding the necessity for increasing the size of the tail plane, while we find the French influence in that the rear of the tail is hinged to act as an elevator. It will be noticed that Mr. George, Mr. Curtis, Mr. de Havilland, and the Valkyrie Co. seem to have learnt from Wright to use their small vertical planes (or blinkers) in front of the machine as a point to thrust against when steering round a corner. Pischoff also uses a vertical plane, but it is so near the centre that it is evidently only used to damp oscillation due to a very low centre of gravity. It is to be observed that the latest Wright model retains

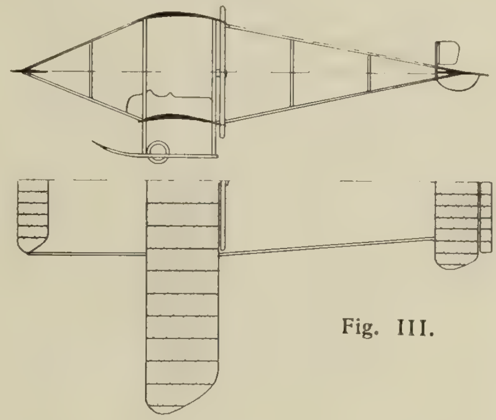


Fig. III.

the blinkers. It would seem to be a feature weighing so little that it is worth retaining on the *F Class* at least, even if the advantage gained be but small, which is by no means clear.

Though studied in France more or less concurrently with the others, machines of *Class B.* were later in obtaining success. The Bleriot and Antoinette were the most notable. These followed the Wright in warping their wings, and the reason that they were somewhat later in succeeding is possibly due to the fact that from their lesser sail area (for they are monoplanes mostly) they are higher speed machines, a fact which was evidently not known by their makers, and later, when engines of the necessary large power were found, it may have taken longer to find pilots for the high speeds of travel and higher speeds of landing. The Bleriots as usually known (except in one type) have the weight-bearing or lifting tail, and these machines have all been practically of a single type, with the propeller in front of the main planes and the elevator aft of the tail, and hence of *Classs B.* Bleriot has, however, recently (1911) brought out a type of passenger-carrying monoplane, which appears to be essentially a Farman biplane with the bottom plane missed out. This is *Class F.*

The 1910 Farman machine has a slight Vee on the lower of the two main planes, while the pilot and the passenger are enclosed in a streamline body. With the exception of the experimental machine previously mentioned, the Voisin people were devoting their attention (1910) to a biplane on somewhat similar lines to the Wrights, that is, with no front elevator, but, unlike the Wrights, they used a tail that bears some proportion of the weight. Santos Dumont entirely changed his designs and produced a small monoplane of importance only because of its small dimensions (1909), and this, as well as practically all the other successful machines can, by a slight effort of the imagination, be regarded as modifications of one or other of the above-mentioned types.

Automatic Stability.

In addition to the above broad lines of practice for stability, which of course is automatic, but of limited range, there have been a great many attempts to obtain what is called automatic stability. The problem is, briefly, to keep a machine in a generally upright position in spite of the peculiarities of gusts. The first

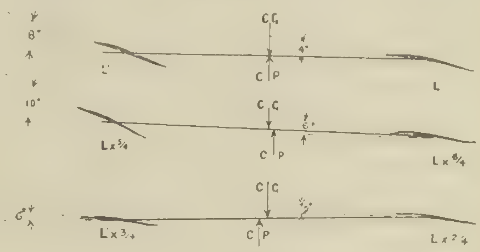


Fig. IV.

thing to realize is, that the behaviour of an aeroplane is entirely unaffected by any steady wind—a steady wind is identical with still air once the machine is launched. A puff into the face of an aeroplane causes it to rise without rolling, and since the machine has inertia, it momentarily relieves the engine of its load and tends to make it race; if the puff desists when the engine has attained an increased speed, the effect of the flywheelage of the engine is to keep up the speed in relation to the air by taking work out of the stored momentum. It is clearly desirable that when a puff strikes one wing only, this wing shall not rise unduly and upset the airman. I have stated that at points 8 yards distant winds may differ in velocity in a 2 to 1 ratio. As things are now, when a wing rises unduly, that is, in a heavy roll, the man has two resources: (a) *by the aileron*—he pulls down the flap attached to the lower wing; this causes it to throw down more air and to be accelerated upwards by the reaction; (b) *by the rudder*—he steers towards the raised wing; this has the effect of diminishing the effective velocity of the raised wing, which therefore drops, and of increasing the effective velocity of the wing on the outside of the circle, which therefore rises. It is to be observed that in either case the force used to oppose the roll is derived from the momentum of the aeroplane, which is accordingly slowed, unless there be a margin of engine power from which to recover speed.

When both devices are used we may see that the use of the aileron or flap, which is similar to but aerodynamically inferior to wing warping, has the effect more particularly of slowing the forward movement of the wing which is lowest, while the steering method has the opposite effect of accelerating it, and the two effects upon the direction can be made to cancel out. In addition there is a momentary tendency for the local

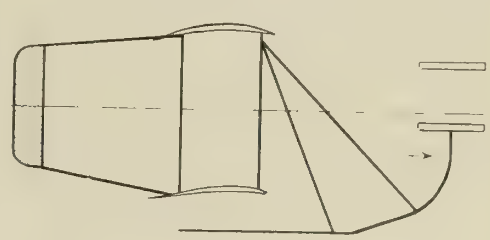


Fig. V.

gust itself to push back the wing which it strikes and raises, but this is probably very small. Practice shews that this effect on the airman's course is not much more than that which steering for equilibrium has on the course of a bicyclist.

In order to turn, either or both of the movements above named for securing stability are employed; but to turn in a small radius, that is to say, to turn without side slipping, it is necessary to slope the machine in precisely the same way that a car running on the Brooklands track is caused to slope by the banking. Now, as an aeroplane when once made is, within narrow limits, a constant speed machine, the banking is always the same for a curve of given radius, and therefore the amount of banking the airman gives to his plane is simple, and varies inversely as the radius of the circle negotiated.

Having regard to the fact that until full banking is effected the aeroplane is pivoting on an axis normal to the plane of the wings, the wing on the outside of the circle has a longer path to follow than the inner wing, and accordingly, as a general rule, the outer wing should be accelerated with regard to the inner. It is unfortunate that the lowering of the wing flap has

precisely the opposite effect, and therefore the relative advance of the outer wing is obtained partly by slowing the rest of the machine and of the energy so obtained. The general slowing of the machine causes it to droop, as a whole, on turning corners. This general slowing must be put down to the increased head resistance introduced by pulling the flap and the rudder plane into the live air.

It is desirable that the aeroplane should not lose momentum, and therefore the wing should be raised for banking by some means (precisely opposite to the wing flap) which shall accelerate it. Moreover, for very important practical reasons, it would be preferable if the inner wing were *not* lowered, the whole of the banking being done above the level of the previous flight. It is advisable, therefore, on turning to raise the centre

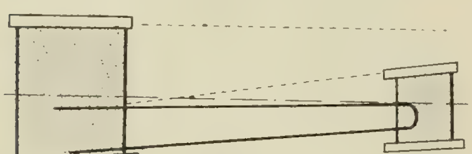


Fig. VI.

of gravity of the whole machine, and the energy for doing this must be derived from the engine, and not from the momentum of the aeroplane, if we are to keep the full available stability.

My suggestion for raising and accelerating the outer wing without retarding or lowering the inner one is to provide for an increased blast of air to be directed mechanically past the wing to be raised; the best way of carrying this out would be largely a matter of experiment on the full-sized machine. Given a single central propeller in front of the planes, and an engine with a good margin of power, a deflector could be employed to throw an increased blast to one side or the other. If, however, there be two propellers acting as tractors, one in front of each wing, the draught from each should be controlled individually by the use of differential gear or crypto type of change speed gear. As an alternative the propellers might be twisted slightly round, though this offers mechanical difficulties. Whatever method is adopted I think that advantage will surely be gained in stability if we do not retard the machine as a whole for the sake of turning a corner, and above all do not retard the wing which requires acceleration. At present it is found that the amount of aileron work done on a gusty day sensibly retards the machine and gives it a tortuous course. It increases its average head resistance, and on large military type machines with one or two passengers the physical fatigue of working the aileron is very burdensome. The superior quality of wing warping over flap pulling is probably an additional reason for the superior average speed of Bleriots over Farmans, but as neither device is "balanced" in the sense in which a rudder is balanced, there should be a distinct

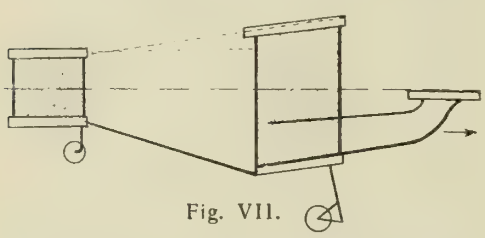


Fig. VII.

advantage to the pivoted and balanced aileron in the matter of physical fatigue. When the air blast method of steering shall have seen the light I think it may show advantages over both, but it wants experiment.

Side gusts introduce difficulties of their own. A machine which is Veed between the wings, and which has its wings mostly above the centre of gravity, is clearly liable to be tilted over by a side wind, the remedy for which is a fin or keel. It is clear that the area of the vertical fin should approximately equal the projected area—(on elevation) of the wings, etc.—and that the centres of pressure should come opposite one another, and the same distance from the axis.

Any Veed aeroplane flying on an even keel will by its inertia oppose resistance to one or other of its wings being rapidly lifted up; as a rule, the presence of an upsetting couple implies the existence of an excess of wind pressure on one wing, so that either :—

(a) The intensity of the loading per square foot of surface (generally about 3½lb.) is increased sufficiently to accelerate the wing upwards, or diminished sufficiently to drop it; or :—

(b) The effective area of the wing is increased—as, for example, when the flap used for banking is pulled into the air stream by the pilot.

If the fabric of the wing were so made that it became a sieve as soon as a pressure of 3½lb. per square foot was exceeded, any important increase of loading might be precluded, and stability thereby improved. It is not wise to say that such an arrangement is easy within the limits of weight, but it is not impossible. Thus, a series of silk slats, like a venetian blind, could be spring-controlled, so as to be opaque to the wind until the spring tension is exceeded, when they would open and let the air through.

Looking into it further, we find that there is no merit in keeping the loading to a fixed limit such as 3½lb. per square foot, provided that any increased pressure is the same in both wings. That means that instead of a spring of definite strength, a pneumatic or other connection must be made between the two controlling spring tensions, so that the slats open when the wing pressures are unequal. This does not preclude the use of ailerons for banking. Another suggestion is that hinged wing extensions should be held by springs which are *set up*, but which yield a large movement when the pre-arranged tension has been exceeded.

This gives us an insight into the reason for the preference of some flying men, such as Wright and Farman, for a *sloppy* rigging of their machines, as against the rigid tightening-up of all wires. This will also, I surmise, be some day part of the justification for the use of tension wires artificially provided with a long elongation, either by inserting springs under the compression struts, or compression springs with a limited travel in the tension wires themselves, or by the use of extensible wires such as stranded cables. Equally the use of a flexible trailing edge to the main planes serves the same pur-

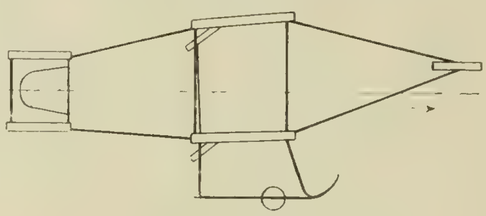


Fig. VIII.

pose, namely, that of relieving whichever wing may be temporarily loaded by a gust, or by a local up-draught beyond what the other wing is bearing. This diminishes the magnitude of the roll due to the local current, and delays its effect so that the airman can deal with it.

It is as well to say that the inertia of the machine which is being used in these devices, that is to say, the more sudden the movement the more efficient the scheme.

There are reasons for keeping the moment of inertia of the aeroplane round its centre of gravity as small as it possibly can be kept, and we must not attempt to improve on the effects above indicated, by spreading the weight to add to the inertia of the wings for example, otherwise we shall soon find trouble which will far outweigh these small advances towards stability.

One more suggestion is to free the two main wings so that they can bodily flex or hinge upwards to a limited extent; and to connect each wing by a pull wire to an aileron at the other side. The object of this arrangement is that, if a local gust lifts the right wing, its movement upward shall, by pulling the left aileron down, thus throwing down more air on the left. at once lift the left wing to a corresponding degree. The movements should be nearly simultaneous so as to avoid a delay during which equilibrium may be lost sufficiently for the man to become aware of it; the mere flexing upwards of the whole of the right wing in itself, apart from the control of the opposite aileron, conduces to stability, since this movement reduces the effective supporting area, and alters the direction of its reaction.

Head Resistance.

The one essential improvement which we require in aeroplanes, other than in stability, is the diminution of their head resistance. To make the idea graphic at once it is only necessary to state that during certain periods the average wind over large portions of the globe, including parts of the British Isles, is reputed to move slightly (*i.e.*, at an angle of 4°) upwards; if therefore we could reduce the head resistance of our aeroplanes till their gliding angle instead of being 8° were 4°, we could often remain in the air like birds supported by the wind, fluttering, perhaps, from point to point to keep in the upstream, but still acquiring a totally new advance in the art of flight.

Progress in connection with the diminution of head resistance can only be made by an extremely slow process, namely, the detailed study of every individual organ and part. It is a double investigation—we require to sacrifice nothing of strength—yet we must gain all that can be taken away of head resistance.

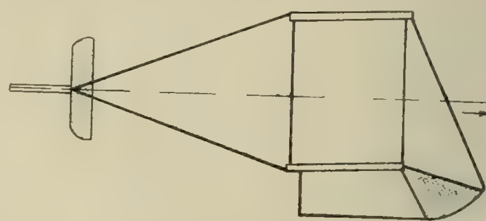


Fig. IX.

All spars at present in use have a section which is more or less pear shaped, but if we compare them to the shape found to give the minimum head resistance for dirigibles we get the contrast shewn by Fig. X., in which the spar section appears much too fat for its length, and has the maximum diameter too near the middle part. We want to know as a certainty (for we already suspect it strongly) that less head resistance and weight would be involved by making larger spars of better form and fewer in number. When, however, we begin to give to the spar an appreciable depth C D, the fin effect of this will react a little upon the steering quality and stability of the airplane in gusts, and this must be tested before any statement of the utility of such an alteration can be made.

It does not necessarily follow that the airship form, having the maximum diameter at F E, is the best for struts, but at any rate one machine which has a notably good gliding angle adopts these thin-shaped spars—I mean Mr. Cody's—which is often careering about Laffan's Plain and Farnborough Common. I have pleasure in thanking him for the several occasions on which I have been his passenger.

The total skin frictional resistance of an aeroplane is about 10 per cent. of the whole resistance, and in investigating the subject it seemed advisable to study the form resistance carefully, and above all the lift, drift, and movement of the centre of pressure for various inclinations of each. Thus a series of aeroplane curves linked together by some common mathematical quality was necessary before any intelligent sequence could be given to a series of experiments. Supposing one has got a set of equations which include all the curves of all known aeroplanes, [and I will show how this may be done,] we may proceed as follows :—Make a model according to any selected equation, conduct a

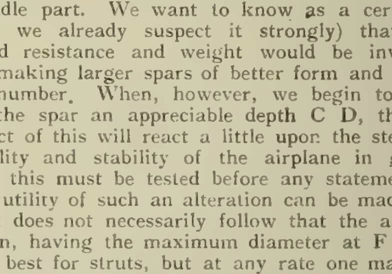


Fig. X.

Fig. X.—Shewing the contrast between the sections of a British made Farman aeroplane and a good pear shape for dirigable work. The eddy caused by the spar will be considerable.

quantitiative experiment upon it, and repeat this experiment on other models whose equation differs from the first by one variable only. Plot the curves and find the minima and maxima desired.

I do not propose to enter into any details of the experimental investigations, but the equation method is as follows :—

After many hours of assiduous "trial and error" and plotting work at various times by Mr. W. C. Claypole, Mr. John Damon and Mr. Watts, we found that the head equation of any such shape can be given by a curve of the form

$$ay^m = x^n (x - b)^2 \qquad \text{(Equation A.)}$$

while the trailing end is given by another curve of very similar equation, viz. :—

$$cy^{m'} = x^{n'} (x - b)^2 \quad \text{(Equation B.)}$$

with very little trouble these two curves are made to meet and oscillate at a point of maximum rise.

An investigation of these equations shows that—

the constant a determines the rise,
the constant b determines the chord,
while m determines the quickness of curvature of the curve,
and n the position of the maximum camber.

If m increases the leading edge becomes bluffer, *i.e.*, bluntly curved down; if m increases the trailing edge becomes bluffer, *i.e.*, bluntly curved down; but the effect of m is always more marked because of the forward position of the maximum camber.

For the value of $m = 1$ the trailing edge becomes cup-like, and lies parallel with the chord, but a smaller value of m can usefully be employed for the leading edge curve.

In practice the position of the maximum ordinate of the curve—i.e., the maximum height of the camber is distant from the front of the plane from 25 per cent. to 35 per cent. of the length of its chord. Now it can be shewn that if this percentage is expressed by p,

$$\text{then } p = 33{\cdot}33 n^{\cdot 7186} \quad \text{(Equation C.)}$$

when n is the exponent of x in Equations A and B above.

For instance, if n = 0.68, then the maximum camber occurs at a point one-fourth of the distance along the chord. If the maximum camber is required one-third of the way along, then n = 1, and the Equation A becomes expression.

$$a.^{m} = x(x - b)^2.$$

To find the Equation when we have the Curve in practice.—The under curves of the De Havilland (1910), Farman (1910), and Bleriot (1908), aerofoils were obtained by setting a straight edge below them—that is, on the concave side—and measuring the distance between the plane and the straight edge at various distances from the front. The figures so obtained were plotted, and curves drawn out to represent the respective aerofoils.

Assuming that the equation to the heads and tails could be represented by the equation of form A and form B respectively, the position of the maximum camber was found, and the exponent n found, from the law of Equation C, namely—

$$p = 33{\cdot}33 n^{\cdot 7186}$$

where p = percentage of length of aerofoil of maximum camber from front. In the De Havilland n = 0.68, while in the Farman and Bleriot n = 1.07.

Having the equations with two unknowns, a and m, two points were taken on each head and tail curve as measured on the machine and values of a and m worked out for these two points. A number of points were then worked out from these equations, to see if the equation did accurately represent the curve. The equations eventually found were as follows :—

De Havilland.

Head. $ay^{0\cdot 8} = x^{0\cdot 68} (x - b)^2$.
Tail. $cy^{1\cdot 7} = x^{0\cdot 68} (x - b)^2$.

Farman.

Head. $ay^{1\cdot 2} = x^{1\cdot 07} (x - b)^2$.
Tail. $cy^{1\cdot 7} = x^{1\cdot 07} (x - b)^2$.

Bleriot.

Head. $ay^{1\cdot 1} = x^{1\cdot 07} (x - b)^2$.
Tail. $cy^{1\cdot 9} = x^{1\cdot 07} (x - b)^2$.

For the above the numerical values of the various constants in inches were—

	De Haviland	Farman.	Bleriot
a	69 5	183 2	97·05
b	6 inches	6 inches	6 inches
c	214·3	355·6	193 6

Diminishing power or increasing speed.

If we can seriously reduce all resistances this would have two effects—(a) on the one hand such a diminution would mean a saving of power; then the economy in weight would seriously react upon sail area, and so a redoubled economy of resistance and power would result. This would tend towards the evolution of the fine weather pleasure flier—the canoe of the air. (b) On the other hand the serious machine for war and business would gain in speed, and hence march along what many consider to be the straight and serious road to stability.

The future of aeroplane flight is in some ways wrapped up with the question of fast flying. Many of us believe that the difficulty of landing at high speeds on prepared landing places can with certainty be overcome, and if so this gives great promise of a solution of all difficulties in regard to stability. That speed is the boldest path and also the straightest short cut past the many difficulties in regard to stability, is proved by the behaviour of models. A well-made model may be dropped in any direction, backwards, forwards, or sideways, after being supported from any part of its being, yet it will within a reasonable height recover its proper position and finish at the ground in a gliding flight, glancing along the floor without shock. Why is this not repeated every day on the full-sized machine? The answer is that it would be repeated if the two essential conditions were the same, viz. :—

(a) If the travelling speed were proportioned to the size, i.e., the proportional speed of a mdoel is proportional to the square root of the scale to which it is made.
(b) If the alighting ground were analogous to the parquet floor on which the model alights, and where it meets with no obstructions to bump, deform, or suddenly stop it.

An example of (a) is the Ding model of 2 ft. span; if we take it as gliding at 16 miles per hour—the full size analogue of 40 ft. span must glide at $16 \times \sqrt{20}$ m.p.h. = nearly 72 m.p.h.

It is a question whether the "70 miles-an-hour aeroplane" can be arranged so as to approach the ground at much less than 70 m.p.h. Knowing as we do that by increasing the speed of the wings, we can cause an aeroplane to be self-supporting at lower speeds, and desiring that all landings shall be effected slowly, we are tempted to ask for a mechanical means of increasing wing area at will. It is well known that one (if not two) of the largest constructors of aeroplanes is in fact evolving something of the kind. The doubt, however, arises whether the attempt to alight slowly will not re-introduce the very dangers we wish to avoid; for not only are the winds more likely to be gusty near the ground, but their fluctuations are greater. In fine, the very occasion on which high speed is required of an aeroplane—if high speed be the defence against upsets due to irregularities in the air currents—is precisely when it is either alighting or rising within the lower 500 feet.

The Righting Couple.

A simple *exposé* of the effects which conduce to the fore and aft stability of an aeroplane can be given if for the sake of clearness, the subject is not burdened with considerations of minor importance at first. We can thereafter revert to these to see whether they will negative, modify or strengthen the approximate conclusion first arrived at. To do this, let us consider an aeroplane to be travelling horizontally, and to continue travelling horizontally as a whole under the influence of its inertia and of the thrust of the propellers during the occurrence of the incidents to be investigated, namely :—

1. The air is approached by a machine which rides at its normal angle of flight.
2. The machine is *cabré*, i.e., tilted or pitched up.
3. The machine is tilted or pitched downwards.

These three positions are clearly shown in the three diagrams, Figs. IV., XI. and XII.

NOTE.—It would be valuable if we could earmark the words *pitch* and *roll* for longitudinal and lateral movements of a dendulous kind respectively.

The assumptions are :—

(a) Neglect the fact that the centre of pressure on each of the planes of an aeroplane moves slightly back when the uptilt of the plane is slightly increased and vice versa.
(b) Assume (which is correct for all small angles) that the effect of tilting a plane or a wing is to increase the lift of that member in simple proportion to the increase of its angle.
(c) Assume that at the outset, the aeroplane is in normal flight with each plane at its normal angle.
(d) Assume the centre of gravity little, if at all, below the centre line of wind resistance.
(e) The angles taken are purely arbitrary—to get simple figures.

Clearly the weight of the aeroplane is a constant, and therefore the positive or negative righting couple, if there be one, is proportional to the travel of the centre of pressure away from the centre of gravity. In other words, the further the centre of pressure is at any moment away from the centre of gravity, the greater is the righting couple. This distance in feet is plotted for each angle of upward and downward tilt in Fig. XI. for an aeroplane of *Class F*.

Commence the investigation by disregarding the existence of the elevator plane in front. A not unusual loading of the planes in this type is about 1,000 lb. on the forward plane and 150 lb. on the back plane, see Fig. XI. Assume with planes of the usual area an angle of 6 degrees for the front plane and 3.5 degrees for the back plane, and on calculating in accordance with the method of Fig. XI. for 2, 4, and 6 degrees of pitch of the machine upward, we find that the righting couple increases as the angle increases in accordance with the full line of Fig. XI. marked "without elevator."

This machine is accordingly stable under these conditions, and will recover from upward tilt of 6 degrees or a little more, provided it be not subjected to any disturbing or capsizing couple greater than, say, its weight acting at a distance of rather less than 1 ft. (or about one-twentieth of its weight acting at the tail). Moreover, it is still more stable in relation to tilt downward (called "below normal" in the diagram), for the righting couple after 4 degrees of deviation is already 1.5 ft. multiplied by the weight, and is many times more at 6 degrees, which is beyond the range of the diagram. The diagram shews that although the front elevator has its uses, it leads to a very much less satisfactory state of affairs from the point of view of fore and aft stability. The elevator in question is supposed to be flat (as is usual), and arranged so that its normal position places it on edge to the horizontal direction of flight.

The effect is shewn by the dotted line (Fig. XI.), where on the right-hand side of the diagram at angles of 4 degrees and 6 degrees there is positively a capsizing couple, i.e., a negative

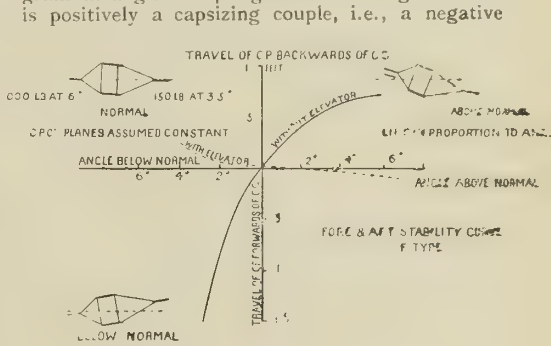


Fig. XI.

righting couple, while at the small angle of 1 or 2 degrees there is just, and only just, equilibrium. When, however, we consider the case where the planes are tilted by a gust so as to be below the normal, we get the state of affairs shewn by the dotted line on the left upper quarter of Fig XI., namely, an appreciable increase of instability at angles of 2 degrees and 4 degrees, calling for the active intervention of the airman, who has to correct this by raising his elevator. Though stability therefore exists at the zero deflection, its maintenance is in fact very soon referred to the dexterity of the operator, say after an angle of 3 degrees or 1 degree respectively for the two sides, has been reached.

Needless to say, an aeroplane responds much more positively and with less loss of momentum, and sometimes more rapidly to any control, when it is nearly unstable in the direction in which that control operates. The existence of the elevator in this position to-day may be due to the strong demand for visible, positive and rapid response in the fore and aft direction, for that very vital operation of correctly alighting. This desire is generally expressed by airmen by stating that they feel more clear about the response of the aeroplane to the elevator if they can see it moving before them. They are quite right in wanting the quick response, and the aeroplane does doubtless respond more thoroughly, but for all that, I think that experiments should be made to modify, if not totally abandon, any member which diminishes the stability of normal flight, even though such member may, as is the case with this elevator plane, slightly increase stability under certain conditions, for example, when it is tipped up and heavily loaded on the occasion of checking a gliding flight.

In view of the above results one is naturally tempted to examine another type of machine, especially one where a plane in the position of the elevator plane is an essential feature. In

the left-hand top corner of Fig. XII. an "S" type aeroplane is shewn diagrammatically. The distribution of loading is such that the small plane in front is set at an angle of 5.5 degrees, i.e., it is more heavily loaded per square foot than the main planes, which, being much larger, carry more total weight at a less angle, namely, 4 degrees. The result is a Vee fore and aft.

First assume, that owing to a puff of wind, or some other accidental circumstance, the aeroplane as a whole becomes *cabré*, i.e., it pitches upwards 2 degrees. As before, the increased angle of incidence of the air on the front plane increases its lift in the same ratio as the angle is increased, namely, as 5.5 is to 7.5, while the lift of the back of plane is increased by a larger percentage, namely, from 4 degrees to 6 degrees. Accordingly, the centre of pressure travels towards the back plane, thereby tending to counteract the upward pitch, that is, it throws the machine back to the normal position.

To draw the curves it was necessary to determine the position of the centre of gravity (from the weight distribution of an ordinary machine), while the centre of pressure which must come in the vertical over the centre of gravity during horizontal flight in homogeneous air, must also be correctly given by the distances from the two planes and their total load.

It is worth while noticing that when the condition of the aeroplane indicated in the left-hand bottom corner exists, namely, the commencement of a downward pitching movement, a much more rapid increase of the righting couple with angular deviation exists than at the corresponding commencement of an upward pitching movement. So greatly does this couple increase, that it is reasonable to say that an unintentional dive or header with this type of machine is difficult. Stability similar to this in all directions is what we want.

In a country where 40 m.p.h. winds are the usual maxima, a man should have a chance of being safe if he can quickly accelerate the aeroplane to 60 m.p.h. Such considerations as these point to the increased speed of the practical machine, while the natural demands of learners will develop, for perfectly still days, a slow, low-powered machine on which air experience can be easily got by waiting patiently for the opportunity. I therefore own myself converted by Lanchester to his declared view that in the air safety lies in speed, and that the elimination of danger from flying is dependent on two factors—(1) The evolution of flying grounds as flat and

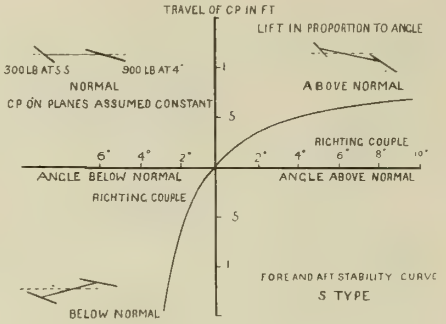


Fig. XII.

perfect as a moderate cricket field, so that alighting at speed is certain to be safe. (2) The development of a speed habit as an instinct of self-preservation in the air, though the departure from our present disposition in favour of slowness for safety, will doubtless require a long period of growth and development.

As the future aeroplane would probably have a gliding angle of one in eight or one in seven, the one lesson which it behoves the airman to learn is the proper height from the ground at which he must restart his propeller at the end of a glide down, so as to diminish his angle of incidence. The learning of this corresponds to acquiring the art of bringing a big ship truly alongside the dock. To be able to start and land is perhaps as important as to be able to balance, since one can never balance the machine without starting, nor start the machine a second time without having safely landed. In the early days, French machines were fitted with wheels for landing, while the Wright machines saved both weight and wind resistance by using skids only. They had, however, to be launched off a special trolley which was not available away from home; the inconvenience of this was soon learnt and they were altered.

The Wrights on their return to America obviously profited by their European experience, for the latest Wright aeroplane has wheels on the skids very much after the European style. The wheels run and keep the skids clear of the ground when the ground is level, and diminish the friction at starting, but the skids are present to bridge over any groove or ditch, as well as to take any bump severe enough to carry the wheel springs to the limit of their travel. One of the duties of the landing frame which may be overlooked is to take side pressures in the event of a flyer alighting across the wind. In some measure the indiarubber device and the triangular guide on the Farman type allows for this, but the contingency is more completely provided for in the "castor" mounting of his wheels arranged by Bleriot. The latter, however, suffers on bad ground by having no skid. One difficulty due to the heavy strains on the landing chassis is its weight, amounting to well over 100 lb. in a Farman, for example. The members are in compression, which generally means weight unless some pneumatic system can be introduced by which the energy of the blow can be taken up by allowing a piston to do work while travelling through some considerable distance, as, for example, by forcing air or a fluid through a fine hole.

At present the combined landing frames and wheels seem to give rise to much regrettable wind resistance. The only means of reducing this appears to be by raising the propellers sufficiently to get a ground clearance and by making water-trough studies of wheel skid arrangements for minimum head resistance, bringing the whole as far as possible within a "stream line form" such as may constitute the body.

CORRESPONDENCE.

No responsibility is accepted by the Editor for the opinions of correspondents, and the right is reserved to omit any portion of a letter if it is considered desirable to do so. Letters to be inserted with a nom de plume must be accompanied by the name and address of the writer. If publication in the next issue is desired letters should reach the Editor in no case later than the 20th of the preceding month.

HORSE POWER RATING.

Sir,—In a letter on this subject in your February issue, Mr. C. A. Brantsen objects to some of the conclusions in my letters in December and January issues. I stated that the R.A.C. formula gives a very close approximation to the horse-power at a piston speed of 1,000 feet per minute, and Mr. Brantsen cites the cases of the 12-14 h.p. Crossley and the 28 h.p. Lanchester against me, the former engine giving 20% in excess of the rating and the latter 20% in deficit. In my letter I stated that I would be inclined to increase the constant in the R.A.C. formula from 0.4 to 0.5 for *modern high compression* engines, of which the Crossley is one, which would bring the rating in close agreement with facts in that particular case. In the case of the Lanchester no bench test was made, but the maximum horse-power attained at 2,800 revs. per min. is taken and the horse-power at 1,000 feet per minute piston speed is deduced from it by simple proportion. The horse-power characteristic of the Lanchester must deviate very considerably from a straight line at these high frequencies of revolution because of the shortness of the stroke, hence the actual horse-power at 1,000 feet per minute must be greater than that calculated by Mr. Brantsen. I shall be surprised if the actual horse-power at 1,000 feet per minute falls below the R.A.C. rating.

For purposes of taxation the formula must be simple and the R.A.C. is the simplest of all. The only logical objection to it, is the one which I mentioned connected with the speed limit, and which Mr. Brantsen has evidently misunderstood. The Government makes it illegal to exceed a speed of 20 miles per hour, yet it taxes large cars on horse-power which they cannot possibly use, except by exceeding the speed limit. For example, the gear-box-less Sheffield-Simplex must travel at 42 miles per hour before the piston speed attains 1,000 feet per minute, which is necessary for the horse-power to equal its R.A.C. rating. In motor cars the greatest comfort and silence are only to be obtained by keeping the piston speed down, but the tax encourages high piston speed and is therefore a tax on comfort. If it were desired to prevent these large engines from ever using the high horse-powers on the road it would be quite easy to do so mechanically without damaging their flexibility and silence at low speeds. A diaphragm in the induction pipe would do it most effectively. Of course some of these large cars do occasionally use their full power on the road, but the average speed of the majority is very little greater than that of the modern 16 h.p. car, although the comfort of the two is, I understand, very different. The proposed tax on "ungotten minerals" was killed by ridicule, but no protest was raised against the tax on "ungotten horse-power."

Mr. Brantsen draws curves of horse-power on a stroke base, as calculated by the different formulæ which have been proposed, but entirely neglects to point out that the differences between the majority of the different curves are purely those of definition. I have been emphasising this point from time to time for some years now, and at last I see it is officially recognised by the powers who dictate in these matters. In the report of the horse-power formula committee published last month the following paragraph occurs, commenting on the $0.4D^2$ formula:—"This simple formula, notwithstanding frequent statements to the contrary, involves both specified mean pressure and piston speed. It assumed, however, that piston speeds did not vary materially from 1,000 feet per minute."—Yours faithfully,

I. B. HENDERSON.

BALL AND ROLLER BEARINGS.

Sir,—The Auto Machinery Company's letter in your February issue still labours the point of difference between us; they endeavour to maintain that my original statement to the effect that the radius forming the ball race, or bearing surface of an annular ball bearing being struck from 7/10 of the ball diameter, is wrong, and that it is more like their formula:—

"The most efficient form of curve section for the outer race is:—

r—.56d.

and for the inner race:—

r—.52d."

which they say is more like common practice, and this radius being so much smaller approximating more to the actual radius of the ball itself, does present a larger bearing surface than my figures disclose.

Following this argument to its logical conclusion, of decreasing the radius to obtain more bearing and wearing surface, why not make it the same radius as the ball itself and have done with it?

Fig. I illustrates a ball between two flat surfaces, and represents the most efficient use to which a ball can be put as an annular bearing, and if the balls could be kept from touching each other without a cage this bearing would be frictionless, and would run indefinitely without lubrication. For obvious reasons this ideal condition has to be modified, as shown in Fig. II., which is a similar type of bearing with the radius struck at 7/10 of the ball diameter, which I maintain is common practice and is in present use.

No doubt, as the Auto Machinery Company have shown, different makers may slightly vary this, but any variation towards Fig. I. will be towards higher efficiency, and any variation towards Fig. III. will be towards greater inefficiency. (It is fully admitted that this type of bearing will not stand any side thrust load.) Fig. III. is an illustration of a section through the edge of two cast iron horizontal discs, the top one (A) being the revolving disc, and the bottom one (B) being the stationary disc. The balls are placed into this semi-circular groove (while the top disc is lifted up), with as many balls as the groove will contain, together with some oil and emery powder of suitable grade. The top disc is then brought down, considerable pressure being applied, and the top disc is then set revolving for some hours according to how much the balls require to be reduced in spherical diameter. This grinding process makes the balls spherical, removes irregularities, and reduces diameters equally.

From this it is obvious why Fig. I. is frictionless, Fig. II. more frictional, and any less radius towards Fig. III. still more frictional, and so on, and this fact of the ball-spinning or twisting in all directions at conflicting circumferential

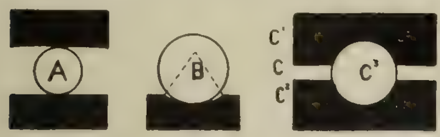


Fig. I. Fig. II. Fig. III.

speeds in relation to the ball races in which they revolve, is taken advantage of as a manufacturing process to reduce the ball's diameter, both in the soft and hardened conditions.

This is the sole reason why all makers of ball bearings must keep so near to a point to point contact, for otherwise the balls set up a destructive wearing process, and it is no credit for any bearing to claim a smaller radius than another, because it can only do so at the corresponding sacrifice of efficiency.

R. F. HALL.

[This correspondence must now cease.—Ed.]

CASTELLATED SHAFTS.

Sir,—The letter from Mr. A. R. Hendon in last month's *Automobile Engineer*, clearly brings up the point, that Mr. Garrard's excellent paper on Castellated Shafts would have been still more valuable, had his careful tests embraced more than one form of splined shaft: the section chosen being by no means representative of the various forms used by modern manufacturers.

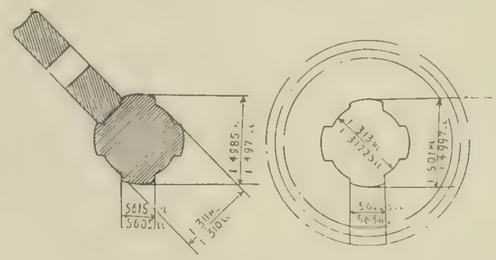
I give sketches of one section which we have used for some years, both for gear boxes and rear axles, with excellent results.

The section differs considerably from those shown by Mr. Garrard and Mr. Hendon, insomuch that both the top of the slots and the spaces between are radial, while the faces of the slots are parallel.

This form is particularly good for gear box shafts on which the sliding gears are to be mounted, where, of course, it is necessary for the sliding wheels to be a good fit, and also essential that they run true in all positions.

The shafts themselves are cut in the milling machine with a special cutter of the section shown in the sketch, and as these shafts are made from some steel which does not require case-hardening, it is evident that the bottom of the slots will form part of a true circle which depends only upon the accuracy of the cutter.

The wheels are bored roughly with allowance for grinding. They are then slotted out and drifted to the required section, care being taken to leave the width of the keyways correct to very fine limits, the depth however, being quite



unimportant except to give good clearance for the shaft.

After the wheel is hardened, the inside is ground out dead true with the outside to the limits shown on the sketch, which gives a sliding fit on the bottom diameter of the shaft.

The result is a true running wheel, with plenty of bearing area and a strong form of spline entirely free from unpleasant re-entrant angles.

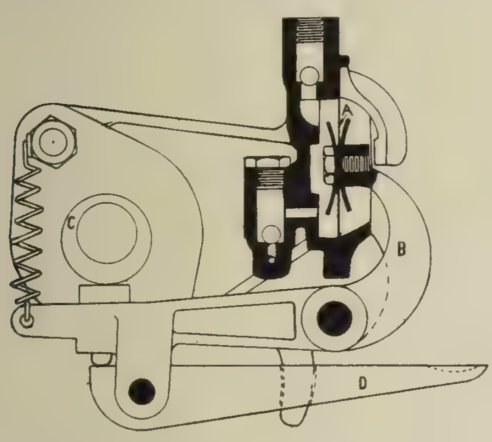
MARCUS C. HUNTER.

RECENT AUTOMOBILE PATENTS.

By Eric W. Walford, F.C.I.P.A.

A Petrol Pump.

To obviate the trouble usually associated with petrol pumps of the plunger type, this pump is constructed with a diaphragm A supported between two cupped plates, and actuated by a lever B

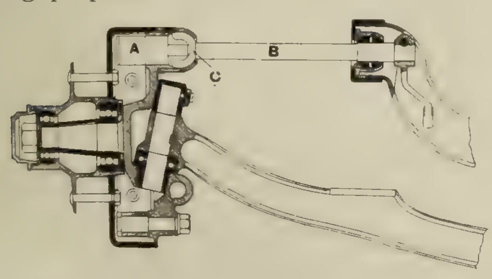


rocked about a pivot by means of an eccentric C driven by the engine. The pump is provided with non-return ball valves as usual, and to enable it to be operated by hand an extra lever D is provided, which, when actuated, forces some petrol into the engine for starting purposes.

E. E. Bentall and G. C. Bingham. No. 5,236/10.

Front Wheel Brakes.

The brake expander, or cam A, is mounted in a bearing on a fixed member which swivels with the wheel for steering purposes. This is connected to an

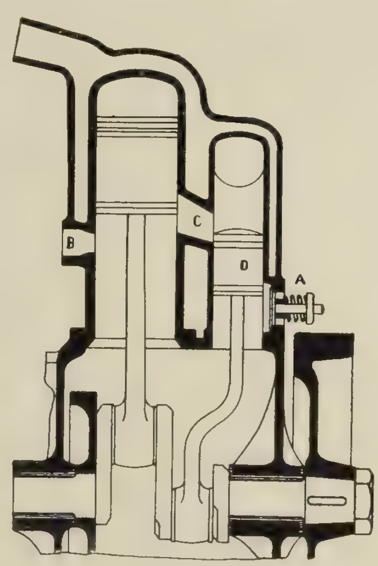


operating rod B by means of a universal joint at C, which is arranged on the axis of the steering pivot. The rod B carries the operating crank connected to the brake pedal or the like, and it has a universal and sliding bearing on the frame so as to be unaffected by the relative movement between the axle and the frame owing to the spring action.

Argylls, Limited, H. Perrot, and J. M. Rubery. No. 6,807/10.

A Two-Stroke Engine.

Air is drawn into the crank chamber through an automatic valve A, and passed under pressure through any suitable carburettor into the working cylinder through an inlet port B. The exhaust aperture C is controlled by a piston valve D, and the exhaust is opened and closed before the inlet port is opened up. The exhaust piston D works in a cylinder, the upper part of

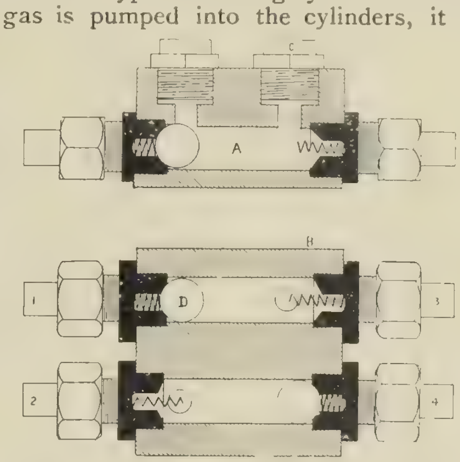


which is in communication with an exhausting device or blower, which, directly the exhaust port C is uncovered, draws the burnt gas out of the cylinder, the port being then closed before the inlet port is uncovered. Thus none of the fresh gas is used for scavenging, and all the exhaust residues are removed before the fresh charge enters the cylinder.

No. 9064/10. S. C. Newson.

A Distributing Valve for Pump Starters.

In the type of starting system in which gas is pumped into the cylinders, it is



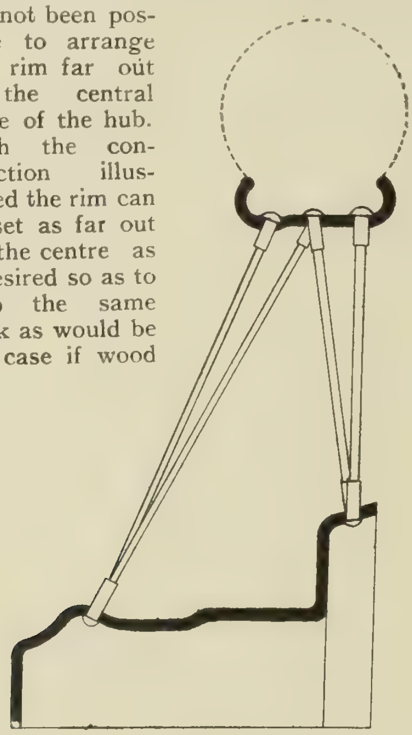
necessary to either pump gas into all the cylinders to ensure filling the one on the firing stroke, or to provide a distributing valve driven by the engine to put the pump into communication with the right cylinder. Both these constructions mean a certain amount of complication, and the present invention obviates this, enabling starting systems of this kind to be applied to existing engines. It comprises a valve body with two passages as A, each of which communicates with two cylinders by means of pipes. Thus the chamber A communicates with cylinders 1 and 4, and another chamber with 2 and 3, this being arranged so that no chamber is in communication with two cylinders any valve of which is open at the same time. Each of the chambers is supplied with gas under pressure by means of supply pipes C, and each also contains a rolling ball D, which constitutes a ball valve adapted to close the outlet at one end of each chamber. When the pump is operated, gas under pressure is forced to both chambers, and tends to flow out to those cylinders the valves of which happen to be open. The flow of gas in this direction causes the balls D to roll to the corresponding end of the chamber and seal the outlet to those cylinders. The result is that the pump can only supply those

two cylinders the valves of which happen to be closed, that is to say, the cylinder on the firing stroke and that on the comto be fired immediately the current is switched on in the first cylinder. Also on the second stroke there is already a compressed charge waiting.

J. D. Bell. No. 22,484/10.

Wire Wheel Construction.

This patent specification states that in previous constructions of wire wheels it has not been possible to arrange the rim far out of the central plane of the hub. With the construction illustrated the rim can be set as far out of the centre as is desired so as to keep the same track as would be the case if wood

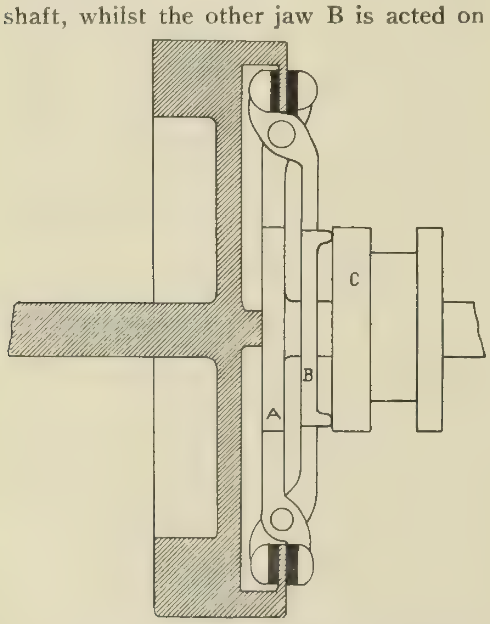


wheels were used. The usual spokes are supplemented by inclined spokes, which support the rim laterally.

Dunlop Rim and Wheel Company, Limited, and A. J. Bowron. No. 4,423/10.

A Peculiar Friction Clutch.

The clutch is of the plate type, a flange on the driving disc being clipped between jaws on toggle levers, one of which is attached to the disc A on the driven shaft, whilst the other jaw B is acted on

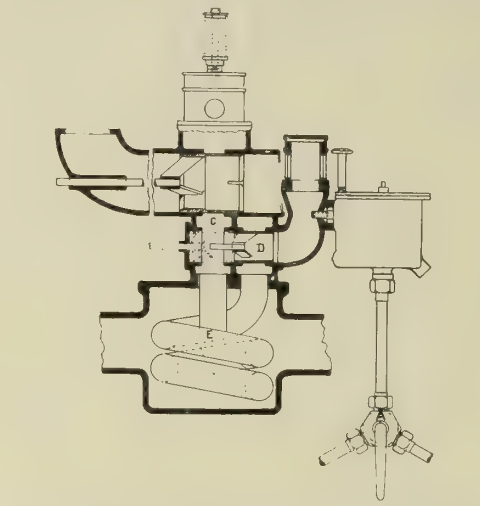


by a collar C sliding on the driven shaft and pushed forward under the action of a spring to cause the jaws forcibly to grip the flange and transmit motion from the driving to the driven shafts.

J. E. I. Baudoux. No. 11,158/10.

A Paraffin Carburettor.

The float chamber is supplied through a three-way cock A, adapted to admit

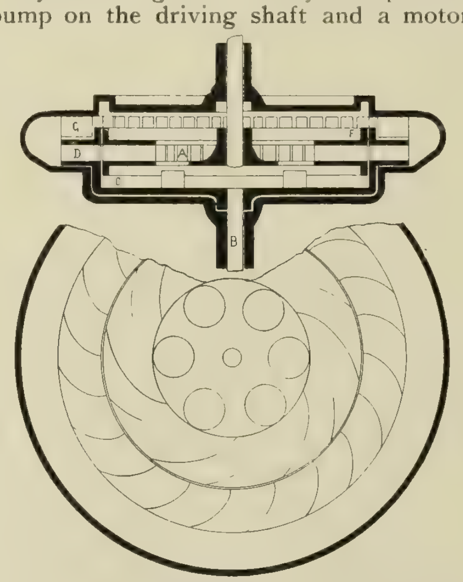


either petrol or paraffin to the float chamber. The jet nozzle projects into the carburetting chamber, which communicates with the passage C either directly through the hollow valve D, which is the case when the valve is in the position illustrated, or indirectly when the valve is drawn to its extreme left hand position, as shown in dotted lines, in which case the gas has to descend the coil E and rise again before it gets into the passage C and the ultimate induction pipe. The coil E is arranged in an enlargement of the exhaust branch, and the gas is passed through this so as to be heated when paraffin is being used.

Wolseley Tool and Motor Car Co., Ltd., A. A. Remington and J. D. Pitt. No. 5431/10.

A Hydraulic Change Speed and Reverse Gear.

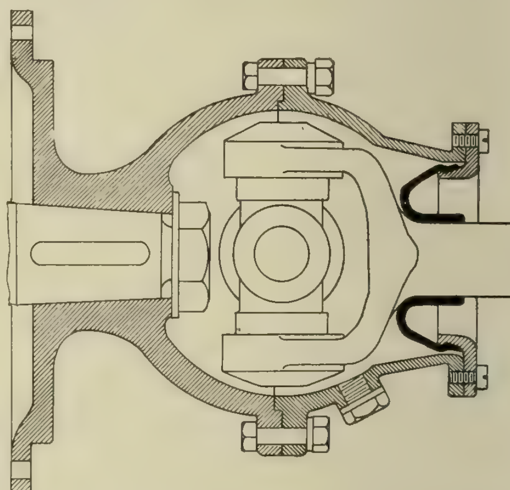
Hydraulic gears usually comprise a pump on the driving shaft and a motor



on the driven shaft with some means of varying the capacity either of the pump or the motor. This construction is somewhat different, the pump and motor being of the paddle wheel or turbine type, and axial displacement of one in relation to the other being employed to vary the speed. The driving shaft carries a paddle wheel A, and the driven shaft B has attached to it a ring C carrying blades at D, the whole being mounted within a liquid-tight casing. With the parts in the position illustrated rotation of the driving shaft and paddle wheel A circulates the liquid through the blades D imparting rotation to the ring C and driven shaft. If the driving shaft and paddle wheel are raised or lowered the delivery of the paddle wheel pump to the blades D decreases, the flanges at F cutting off the passages. When the wheel is raised to its highest limit the liquid delivered by it impinges on some fixed vanes G and acts upon the driven vanes so causing reversal of the driven shaft.

A Universal Joint.

The feature of this invention consists in constructing the joint so as to retain its lubricant without impairing its flexibility. The driven shaft carries a sleeve portion, to which is attached one of the cross pin members, the other cross pin being mounted in jaws carried by the driven shaft. Between the end of the first shaft and the driven shaft is arranged a leather ring, which is cupped as

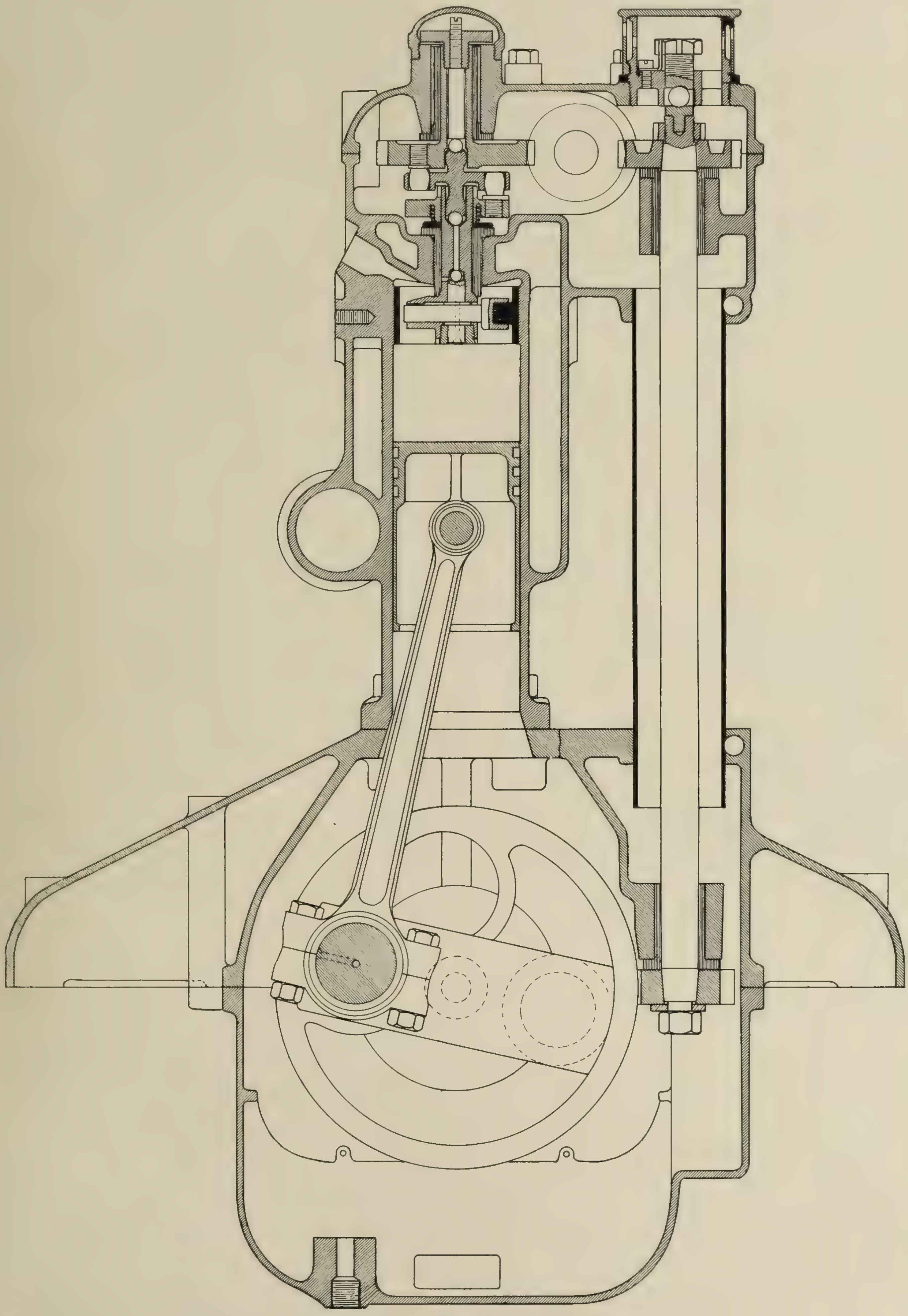


shown so that should any pressure arise inside the universal joint casing the tendency will be to compress the leather, instead of expand it, and so force its fibres close together preventing any oil leaking therethrough. The action of rotation of the joint tends to throw the lubricant away from the leather to the points which require most lubrication, so that the chief objection to leather oil retaining pieces—their liability to leak under pressure—is obviated.

Société Anonyme des Automobiles Delaunay-Bellville. No. 25,561/10.

THE COTTEREAU ROTARY VALVE ENGINE.

This engine, which is illustrated on the opposite page, was one of the most interesting designs at the last Paris show, both because of its intrinsic peculiarities and on account of the fact that it is known to have been in practical, satisfactory use for several months at least. The system of the valve gear was dealt with briefly in the January issue of *The Automobile Engineer*, and consists of a rotating cylindrical ring (shown in black section in the accompanying illustration) which is split like a piston ring and has sufficient natural spring to maintain it in close contact with the main cylinder wall. The ring is also provided with a port which registers with corresponding openings in the cylinder walls as the valve revolves. These ports are, of course, set close together and are not shown, but their position can be judged by that of the two flanges, the edges of which can be seen on each side of the cylinder a little behind the plane of section. It will be observed that the driving arrangements are elaborate, the idea being to give opportunity for expansion to take place without distorting any part. In the lower portion of the spindle there is an oil passage closed at each end by steel balls which also appear to act as thrust bearings, and thus only quite a small quantity of lubricant reaches the combustion chamber from the top. Mechanically the design is certainly not attractive, both because the valve appears to work under most disad-



THE COTTEREAU ROTARY VALVE ENGINE.

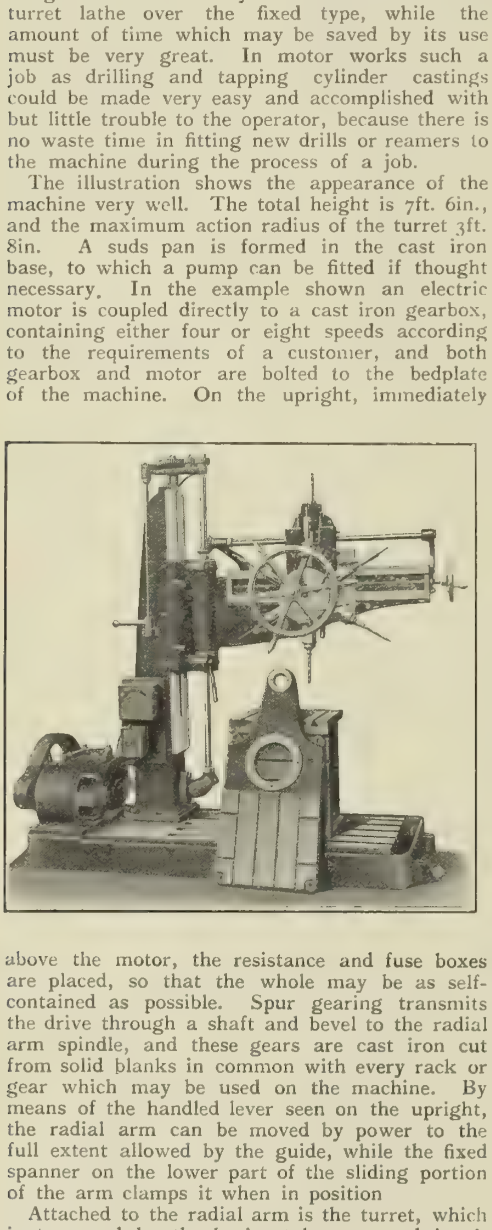
See description on page 334.

vantageous conditions and because the driving gear is so elaborate. Obviously the cost of manufacture would be high, but against this must be set the facts that an excellent valve diagram is obtainable and there is no part likely to produce sound in running. Also the manufacturers are one of the old French firms with many years of experience behind them, and it is unlikely that they would put a new design on the market until they were entirely satisfied with its reliability.

AN INGENIOUS AND USEFUL RADIAL DRILL.

One of the most interesting machine tools which have been placed on the market recently is the sensitive radial turret drilling machine, manufactured by John H. Storey and Company. This machine is quite as much an advance in design over the ordinary radial drill as was the turret lathe over the fixed type, while the amount of time which may be saved by its use must be very great. In motor works such a job as drilling and tapping cylinder castings could be made very easy and accomplished with but little trouble to the operator, because there is no waste time in fitting new drills or reamers to the machine during the process of a job.

The illustration shows the appearance of the machine very well. The total height is 7ft. 6in., and the maximum action radius of the turret 3ft. 8in. A suds pan is formed in the cast iron base, to which a pump can be fitted if thought necessary. In the example shown an electric motor is coupled directly to a cast iron gearbox, containing either four or eight speeds according to the requirements of a customer, and both gearbox and motor are bolted to the bedplate of the machine. On the upright, immediately



above the motor, the resistance and fuse boxes are placed, so that the whole may be as self-contained as possible. Spur gearing transmits the drive through a shaft and bevel to the radial arm spindle, and these gears are cast iron cut from solid blanks in common with every rack or gear which may be used on the machine. By means of the handled lever seen on the upright, the radial arm can be moved by power to the full extent allowed by the guide, while the fixed spanner on the lower part of the sliding portion of the arm clamps it when in position

Attached to the radial arm is the turret, which is traversed by the horizontal screw and handwheel gearing with the saddle; the total traverse is 2ft. 6in. Feed is by the large diameter handwheel shown, and is extremely sensitive. After the completion of an operation with any one separate spindle, a hand nut on the turret face is slacked, and the lever seen between the handwheel spokes is lifted: this declutches the operating spindle, and allows the turret to rotate until the next tool is vertical, at which point it drops thereby connecting up the drive. The turret is then locked by the hand nut.

Normally there are six spindles, but twelve can be supplied if necessary. A balance weight is connected by chains and pulleys to the turret to enable the latter to move with perfect ease. Each spindle is allowed a nine-inch feed, so as to be capable of dealing with most work without further movement of the arm. Of course, an automatic reverse can be provided if thought necessary for tapping. In order to facilitate handling, care has been taken to place all control levers as near to the operator as can be allowed by the design. A loose table, to which the job can be attached, is bolted to the slots in the base, but can be removed easily should occasion require its absence

For shops which are not fitted with electric power, or where the motor arrangement entails the use of line shafting and a belt drive, a speed cone with fast and loose pulleys, and the necessary belt shifting apparatus can be fitted in place of the motor and accessories usually provided.

Care has been taken to make the radial arm as stiff as possible, in order that the extra turret weight may not eventually cause enough play to interfere seriously with correct drilling.

THE MANUFACTURE OF STEEL BALLS.

We have received from the Auto Machinery Company, Limited, a full report of the finding of the judges in a case which was recently before the Courts, and in which they were the successful defendants. The dispute concerned the validity of some patents concerning apparatus for the accurate grinding of hardened steel balls, and the document we have received gives a full account of two somewhat similar processes.

From this it appears that the plaintiffs' method of grinding was as follows:—In the original machine there were two disc of metal having a number of concentric circular grooves, in which the balls to be ground could run as the discs were revolved under pressure and, in order to submit every ball to an equal amount of grinding, the plaintiff devised a machine in which the discs were vertical, and in which there was a segmental pocket, so that whenever the balls came round to that place they escaped from confinement in the double half grooves in which they had been grinding, and fell into an open space which probably sloped up a little so that they had a tendency always to fall back on the other rapidly revolving disc out of which the pass had not been cut. The effect of that was that every time they came round they escaped from their grooves, they tumbled into a heap, from that heap they were picked up by the rapidly revolving disc, and then went into some other groove. It is evident that there was no law by which they should go into one groove more than another, so when that operation had gone on perhaps for very many thousands of times, it would produce a very high degree of uniformity. The defendants' machine, on the other hand, did not rely upon accident to send any particular ball through each groove in succession. In it each groove is provided with a pocket in connection with one other groove only so that the effect is to pass the whole sequence of the balls through grinding grooves, so that every one of the balls has precisely the same history of grinding as its neighbour. The balls in this machine, without any interference from the outside, pass from the outermost groove to the next, and so on, moving inwards. Finally to the innermost one, and then pass out again and back again at the outside, so that however long or however short the operation is, all these balls are treated precisely alike and average has no part nor law in the matter.

REVIEW.

Our contemporary, *The Aero*, believing that the interests of the new science and sport can at present better be served by a monthly publication more on the lines of *The Automobile Engineer*, than by a weekly, has, during the past week, so changed its state, the first issue of the new order appearing during the Aero Show at Olympia.

Naturally a large section of the paper is devoted to the exhibits, and the remainder contains articles interesting both socially and scientifically. *The Aero. Iliffe and Sons, Ltd. Price 6d. monthly.*

CARBURETTOR ACTION.

With reference to the recent correspondence in these pages between Professor Morgan and Mr. R. W. A. Brewer, which was closed last week, we have received a letter from Professor Morgan expressing his keen desire to reply to Mr. Brewer's last communication.

We thought fit to close the correspondence, because it had ceased to be of an informative nature, but would like to take this opportunity of expressing our extreme regret that Messrs. Morgan and Brewer cannot discover the agreement which undoubtedly exists between their separate researches. We cannot help feeling that both parties are right in their facts, and that it is in the expression only that they disagree. That personal disagreement should have occurred is therefore all the more to be regretted, and, for the sake of the advancement of knowledge, it is to be hoped that further investigations by both will show them the real concurrence of their ideas.

MISCELLANEOUS.

MR. H. E. LANCASTER, consulting automobile engineer, has removed from his old address and now occupies offices at Craven House, Kingsway, London, W.C.

VAN RADEN AND CO., LTD., inform us that a rumour has reached their ear to the effect that their well-known accumulators are not made in this country, and they ask us to give publicity to their denial of this statement, the articles in question being made throughout at their Coventry works.

S. WOLF AND CO., who are the suppliers of the Solex carburettor, the N.F. magneto, and the S.R.O. ball bearings, have removed to larger premises in order to cope with the recent increase in the demand for these several components.

GEORGE MOORES AND COMPANY have removed from 46a, Market Street to more central premises at 14, Cross Street, Manchester.

CATALOGUES RECEIVED.

CASE HARDENING COMPOUND.—W. H. Palfreyman and Company have sent us a pamphlet descriptive of the method of using their special case-hardening compound, "Hydro-Carbonated Bone Black." This material is prepared from ground bone freed as much as possible from grease, etc., before grinding. Carbonising takes place in retorts kept at an even temperature, and as soon as the process is completed the hot product is charged with carbon obtained from a hydro-carbon oil. This is claimed to fill all the pores with a substance which is entirely unaffected by moisture, so ensuring that the finished compound shall be of a regular quality notwithstanding changes in atmospheric conditions. This substance is recommended by the makers for use where exceptional hardness or a very deep case is desired, and is in regular use by several of the principal automobile manufacturers of this country.

VENTILATING FANS. — The Electric Ordnance and Accessories Company, Limited, have published a pamphlet list of their fans for ventilating purposes. These fans are made in various sizes for belt driving, or combined with electric motors for all varieties of current.

AIR COMPRESSOR.—A convenient portable electric air compressor for use in conjunction with pneumatic mechanism, or more especially for cleaning work, has been put on the market by Lacy Hulbert and Co., and is described fully in an illustrated leaflet obtainable from that firm.

TYRES.—New lists have been issued by Spencer Moulton and Co., Ltd., as well as supplementary lists dealing with rubber matting and various other rubber products.

TYRES.—A new catalogue has been issued by the Polack Tyre and Rubber Co., Ltd.

BLOW LAMP.—A combined soldering and blow lamp of a new pattern is dealt with in a leaflet issued by Harris and Samuels.

MACHINE TOOLS.—A complete catalogue, well illustrated, but in convenient pocket size, has reached us from Brown and Ward, Ltd.

MACHINE TOOLS.—"The Milling Machine Analysed" is the title of an interesting booklet issued by Alfred Herbert, Ltd. It deals in a most exhaustive manner with the construction and handling of certain types of machines, and should prove useful.

POWER TRANSMISSION AND MECHANICAL CONVEYING, is the title of a booklet issued as an advertisement of Scandinavia belting. It consists of photographic illustrations of belting as applied for the different commercial purposes indicated by the title.

ANTIFRICTION METAL.—Some interesting tests of Glacier bearing metals are described in the latest pamphlet issued by the Glacier Anti-Friction Metal Company, Limited. The same booklet also gives particulars of the Company's various compositions.

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CONTENTS.

A SUGGESTION FOR THE INSTITUTION OF AUTOMOBILE ENGINEERS.

RATHER less than a year ago in referring to the Institution of Automobile Engineers, we remarked that the immediate prospects seemed to be bright. At that time there is no doubt that the most interesting session which had so far been held was just over, and we had hoped to see still greater progress this year. It is not our present purpose to comment upon the papers which have been read since last November, but, before meetings have ended for this session, there is one point which certainly is worth discussing, and that is whether any means could be found whereby the attendance at meetings could be improved.

Of course the total membership of the Institution is small, but this is probably principally because the I.A.E. has not been long in existence, and we believe that the number of new members enrolled this year is regarded as entirely satisfactory. Still the fact remains that the average attendance at meetings is very poor, even by comparison with the small total membership roll. Of course this can be said with equal truth of almost all the engineering societies, and this question of poor attendance has been debated many times by divers governing bodies, while the subject has also been discussed in the engineering press generally. The case of the I.A.E. however, is not quite the same as the older engineering societies. These have duties, and are useful to their members, in ways which are not shared by the I.A.E. Some of them hold examinations which are in themselves qualifications all the world over. Others use the power of election in much the same way—to bestow similar qualification, and so advancement, upon the successful ones of the profession. Therefore the older societies would still be valuable and necessary without the reading and discussion of papers, and also, when the scope of an institution is very wide, it is only to be expected that a moderately small percentage of members will be interested in any one paper.

With the Institution of Automobile Engineers, much more depends upon the value of the body as a debating society. The field is comparatively narrow, that is to say the interests of the members and associates are sufficiently similar to make it easy to find a subject for a paper which will be interesting to practically each one. Possibly in the past the selection of papers has not been ideal, but there have probably not been too many put forward for choice. Still there may very likely be a certain amount of truth in the suggestion (which has sometimes been made) that too much attention has been given to the academic and not enough to the practical. We believe however, that no good practical paper has yet been refused, and it is therefore upon the rank and file of membership and not upon the governing body that any blame must rest. Doubtless as the number of members grows, so will the number of papers offered and the value and interest of the papers, but such growth is necessarily very slow and needs as much artificial stimulation as possible.

It is noticeable to regular attendants that, as might be expected, the men whose views would be valued, but who most frequently fail to put in an appearance, are those whose work ties them to parts of the country other than London. Men who are actually engaged in the industries—men whose opinions are worth most to automobile engineers as a whole—have seldom much time on their hands. A journey to London from the Midlands occupies practically twenty-four hours, for although it can, of course, be accomplished in much less, still to travel during the afternoon, attend a paper in the evening and travel back during the night does not leave the average man feeling that he is as ready as usual to start the work of the day on the following morning. An analysis of the membership would shew that there were far more men actively engaged in the manufacture of automobiles within a twenty mile radius of Birmingham than within a similar circle struck from Westminster, but it would certainly be most unwise to transfer the head-quarters, for the reason that in almost every way London still remains the centre of the industrial world, and it is right that an important engineering body should have its headquarters in the capital city.

During the average session some six or seven papers are read, occupying a total time of about twelve to fifteen hours. To attend every one, for a Londoner, costs an extremely small sum of money, perhaps even nothing at all, and absorbs no noticeable portion of his time For men in the Midlands, let alone anyone living still further afield, attending these meetings is equivalent to losing practically ten days of work whilst necessitating an expenditure of, at the very least, as many pounds as there are meetings. Looked at in this way it becomes obvious that the papers must indeed be valuable to gain the constant attendance of those at a distance It also shows that the amount of time which has to be given by a provincial member is out of all proportion to that occupied in actually hearing the paper or discussing it. It would therefore appear to be difficult greatly to improve the attendance without making some more or less drastic alteration in the arrangement of meetings, altogether apart from their nature.

A number of learned societies have from time to time over

come the trouble of maintaining interest (and thereby maintaining the usefulness of the body) by means of organised mass meetings lasting for several days at a time, and one may instance the British Association or the Iron and Steel Institute, as being examples of large bodies who accomplish their ends most successfully by these meetings. It would seem worth while considering whether the advancement of the Institution of Automobile Engineers could not be hastened by a similar procedure. Supposing for instance, that a summer meeting was held in London, and that all the papers of the session were read during this one meeting, we believe that a large proportion of the membership would attend, that the discussions would be extremely vigorous, and that no one would lose unreasonably in other ways by the time occupied. In the summer, say in July for example, the majority of automobile factories are slack, comparatively to what they are in winter.

None the less valuable side too, of an Institution should be its power to make men of common interests acquainted with each other personally, as this undoubtedly assists the interchange and dissemination of ideas to an enormous extent. A summer meeting during which excursions, and even amusements, might be organised, in addition to the solid work of the papers, would supply what is needed. It might even be possible to arrange not only an annual but a bi-annual meeting, holding one in June or July and the other in November, December or January, but we think most probably if the scheme were adopted it would be best to begin with a single meeting in the summer and to then see what were the feelings of the membership as regards the institution of a second annual affair. Still more useful would it be to hold an annual meeting at a changing centre, say London one year, Birmingham the next, Glasgow the next, and so on. It might even be possible to get a sufficient party together to support a meeting held in one of the manufacturing centres of France and Germany. Manufacturers all the world over are generally willing to assist anything of this nature in their own vicinity in every way, and we believe that a week in, say Birmingham, would be filled in a manner which would be of really greater value to members than the present meetings.

Sometimes long meetings have been spoilt by too many papers or by papers of too great a length. If a meeting lasted for five days there ought not to be read more than two short papers each day, or two digests of longer papers. There is much to be said in favour of the system of publishing a paper in advance and the authoı reading but a *précis* of it, the discussion following immediately, as no time is wasted thereby and, to a man who has read a paper carefully beforehand, the whole time spent in hearing it read is practically thrown away so far as he is concerned. However, these are mere matters of detail and there is no need to consider them unless there is some prospect of the broad idea being acceptable.

We should be extremely glad to hear the opinions of our readers, both members and would-be members, on this subject, for if it were once settled that the members would like to try the experiment of holding an annual meeting in place of the whole or of a portion of the present monthly papers, we do not think that the necessary arrangements would be at all hard to make or difficult to carry through to a successful conclusion.

THE SPEED OF RECIPROCATING ENGINES.

A consideration of the limitations with additional reference to horse power formulae.

By James Langmuir Napier.

THE subject of this article is speed and the factors which influence it. It has no direct concern with horse-power formulæ, nor have I any desire to add to their number, being satisfied that the R.A.C. formula at present in common use embodies what may be described as the maximum practicable amount of convenience. It is impossible however, in dealing with the subject of engine speed at this date to avoid consideration of the recently issued report of the Horse-power Formula Committee, and the conclusions as to engine speed arrived at by the Committee. These conclusions include a general statement that piston speed varies with the ratio of stroke to bore, and a particular determination that the variation can be expressed by the equation $\sigma = 600\ (r + 1)$, σ being piston speed in feet per minute and r the ratio of stroke to bore.

In order to ascertain what support for these conclusions was to be found in the table of engine data on which the report is based, I extracted from the table all results derived from two classes of engines, the first including those having a stroke-bore ratio of 1.08 or under, and the second those having a stroke-bore ratio of 1.5 or more. These numbered twenty-four and twelve engines respectively, and yielded certain numerical averages which I set down here for what they are worth, with the warning that the great majority of the trials were incomplete, and that averages should be used with caution in any case.

Stroke-bore Ratio.	σ at Max. H.P.	σ at Max. $\eta\rho$	Max. $\eta\rho$.	$\eta\rho$ at Max. H.P.
1.005	1264	675	88.8	70.57
1.59	1512	828	82.0	66.81

As in the Committee's report, $\eta\rho$ means effective mean pressure, including the mechanical efficiency of the engine. It will be noted that the average speed at maximum horse-power supports the general conclusion of the Committee. The tendency of the other averages may be obscure at this stage.

From these twenty-four and twelve examples I have selected those in which the trials were sufficiently extended to reach, or nearly reach, both the speed at maximum h.p. and the speed at maximum $\eta\rho$, and in which the results were not vitiated by obvious errors of observation. I have plotted the results of these trials in Figs. I. and II., of which Fig. I. represents the results of the engines of high stroke-bore ratio, and Fig. II. those of low stroke-bore ratio. I have taken a liberty with the result of Trial No. 45 by altering 63.9 h.p., which may be a misprint, to 53.9 h.p., which is in accordance with the general trend of the curves. In both figures the upper series of curves represent the mean effective pressures and the lower the horse-powers. Even where these cross there should be no difficulty in distinguishing between them. For ease of reference I append here a table of the principal dimensions of the engines of which the trial results are plotted in the figures:

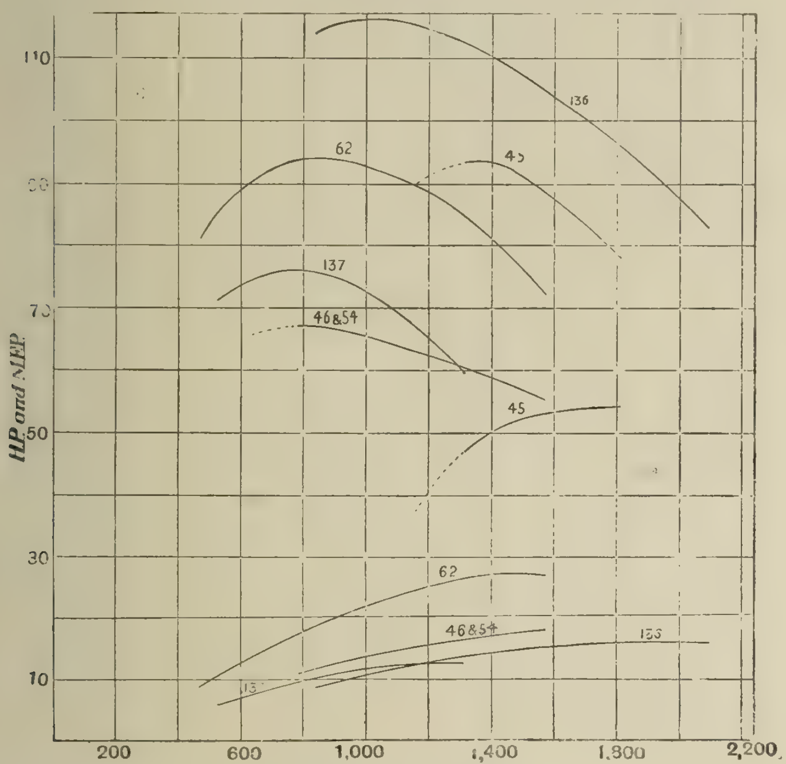
Fig.	No.	Bore.	Stroke.	Stroke-bore Ratio.	No. of Cyls.
1	45	4.00	7.00	1.75	4
1	46 and 54	2.95	4.73	1.60	4
1	62	3.15	4.73	1.50	4
1	136	3.94	6.30	1.60	1
1	137	2.60	3.94	1.52	4
2	4	4.73	5.00	1.06	4
2	14	4.73	5.00	1.06	6
2	78	4.63	5.00	1.08	4
2	79	5.00	5.13	1.03	4
2	100	4.50	4.50	1.00	6
2	110	4.73	5.12	1.08	4
2	131	4.50	4.75	1.06	6

It is not necessary to examine in detail the curves plotted in Figs. I. and II. A glance is sufficient to indicate that the simplicity of the relation $\sigma = 600\ (r + 1)$ must be considerably modified to account for the variations exhibited in each figure and as between the two. There are, in fact, only two points of consistency observable, and both relate, not to speed at maximum h.p., but to speed at maximum $\eta\rho$. It will be observed that between the two diagrams there is no overlapping. The speed at maximum $\eta\rho$ is always higher in Fig. I. than in Fig. II., and taking Fig. I. by itself, the speed at maximum $\eta\rho$ is always higher when the stroke is greater.

The "maximum practicable piston speed" is not defined, but since it is said to be 600 $(r + 1)$ feet per minute, and that quantity is used in calculating the maximum b.h.p. rating, it seems reasonable to assume that "maximum practicable piston speed" is intended to be taken as piston speed at maximum horse-power, and this speed is said to be the same for an engine with cylinders 2½in. × 2½in. as for an engine with cylinders 5in. × 5in., with the somewhat paradoxical result that doubling the stroke has no effect on the relative power of the two engines.

The explanation is, of course, that the increased piston diameter has a detrimental effect (other than the heat effect which is separately accounted for) precisely equal to the benefit derived from the increased stroke. From Mr. Burls' explanatory paper, it appears that this detrimental effect arises from the increased mass of reciprocating parts corresponding to increased cylinder diameter.

As I shall show later, the influence of reciprocating mass on speed in any reasonably designed engine is practically negligible. It is true that under certain circumstances heavy reciprocating masses may make high speeds unpleasant, but they do not make high speeds impossible, or even (within the limits of the strength of materials) appreciably more difficult of attainment. If any limitation of this nature be admitted, the Committee's formula



Piston speed in feet per minute.

Fig. I.

becomes, not a measure of possible horse-power, but a measure of pleasant horse-power. Mr. Burls is logical in this matter, and has endeavoured to set up a standard of pleasantness, but his standard is too low. It is infringed by every four-cylinder engine.

My view of the matter is that in attempting to determine the variation of piston speed with stroke-bore ratio, the Committee have begun at the wrong end. I propose to begin by considering ηp, which is, after all, the source of piston speed, and probably more effective than a mathematical abstraction.

A cylinder may be supposed to be filled with gas of a definite heating value at the pressure of the atmosphere, and that gas may be supposed to be exploded and expanded behind a piston moving with infinite velocity, so that no external cooling of the gas takes place. Under such circumstances the mean pressure in the cylinder during the working stroke would have a maximum value which I shall call A.

If, however, the gas be required to follow the piston into the cylinder in a short period of time the pressure in the cylinder must be less than the atmospheric pressure. The total heat in the charge of gas, and approximately also the mean pressure, will be diminished in the proportion $\frac{p}{\text{P}}$, corresponding to the pressures inside and outside the cylinder, and if the stroke were sufficiently long the mean pressure during the working stroke would be approximately $\text{A}\frac{p}{\text{P}}$. Supposing the inlet valve to be equal in area to the piston, and the piston to move with a constant velocity v, then the equation connecting v and p is

$$v^2 = \text{C}^2\left(1 - \frac{p}{\text{P}}\right)$$

where C is the velocity with which the gas would flow into a vacuum under atmospheric pressure.

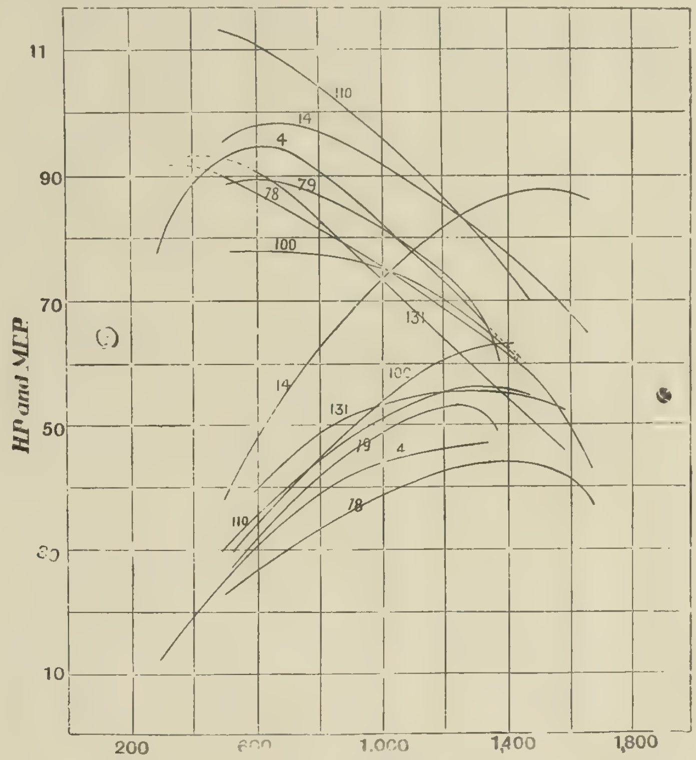
This equation assumes the gas to have acquired velocity by falling through a height, and to be flowing with a constant velocity. We have to consider however, that the gas is at rest at the beginning of the stroke and acquires the velocity of the piston under the influence of gravity and its own elasticity while moving through the distance s, the length of the stroke. Under these circumstances we should have

$$v^2 = \text{B}s\left(1 - \frac{p}{\text{P}}\right)$$

B being a constant. From this we get $\frac{p}{\text{P}} = 1 - \frac{v^2}{\text{B}s}$ and

$$\text{A}\frac{p}{\text{P}} = \text{A}\left(1 - \frac{v^2}{\text{B}s}\right)$$

which would be the mean pressure if there were no cooling and if the motion of the piston were uniform. Neglecting true piston motion exaggerates the effect of long stroke.



Piston speed in feet per minute.

Fig. II.

The diameter of the cylinder will certainly influence the rate of cooling of the exploded charge. I have not had the advantage of reading Prof. Callendar's paper on the subject, and I have made no independent investigation of it. It seems reasonable however, to suppose that the rate of cooling will be, more or less, directly proportional to the cooling surface and inversely proportional to the volume cooled.

Taking only the water-cooled surface of the cylinder, the rate of cooling would thus be proportional to

$$\frac{\pi d s + \frac{\pi}{4} d^2}{\frac{\pi}{4} d^2 s} = \frac{4}{d} + \frac{1}{s} = 4\left(\frac{1}{d} + \frac{1}{4s}\right)$$

The rate of cooling is also directly proportional to the difference of temperature, so that we have, if T be temperature of gas, t time, and C temperature of cooling water,

$$\frac{d\text{T}}{dt} = -m\left(\frac{1}{d} + \frac{1}{4s}\right)(\text{T} - \text{C})$$

$$\text{and T} = \text{R}e^{-mt\left(\frac{1}{d}+\frac{1}{4s}\right)} + \text{C}$$

When $t = o$, that is when the velocity of the piston is infinite, T = R + C, which is the temperature corresponding to the ideal mean pressure A; and, since pressure is proportional to temperature, when the stroke is performed in time t, the mean pressure will be proportional to

$$\text{A}.\frac{\text{R}e^{-mt\left(\frac{1}{d}+\frac{1}{4s}\right)} + \text{C}}{\text{R} + \text{C}}$$

And if we assume (introducing a further error) that C is the temperature of the bottom line of an indicator diagram, this expression becomes

$$\text{A}.e^{-mt\left(\frac{1}{d}+\frac{1}{4s}\right)}$$

which, since t is equal to $\frac{s}{v}$, may be written

$$\frac{\text{A}}{e^{\frac{m}{v}\left(\frac{s}{d}+\frac{1}{4}\right)}}$$

Combining this with the previously assumed effect on mean pressure of increased stroke, we arrive at the equation:

$$\eta\rho = \frac{A\left(1 - \frac{v^2}{3s}\right)}{e^{\frac{m}{v}\left(\frac{1}{d} + \frac{1}{4}\right)}}$$

which is the equation used in plotting the curves of mean effective pressure and horse-power shown in Fig. IV.

The curves plotted in Fig. III. are deduced from the probably less accurate approximation:

$$\eta\rho = \frac{A - \frac{Bv^2}{s}}{e^{\frac{ms}{vd}}}$$

in which, of course, the constants, although indicated by the same letters, have different values. The method by which these curves are arrived at is as follows:

If u and y are functions of x, $\frac{u}{y}$ is a maximum when

$$\frac{u}{y} = \frac{\frac{du}{dx}}{\frac{dy}{dx}}, \text{ therefore } \eta\rho \text{ is a maximum when}$$

$$\frac{A - \frac{Bv^2}{s}}{e^{\frac{ms}{vd}}} = \frac{-\frac{2Bv}{s}}{\left(-\frac{ms}{v^2d}\right)e^{\frac{ms}{vd}}}$$

$$A - \frac{Bv^2}{s} = \frac{\frac{2Bv}{s}}{\frac{ms}{v^2d}}$$

$$As - Bv^2 = \frac{2Bv^3d}{ms}$$

$$m\,(As^2 - Bvs^2) = 2Bv^3d$$

$$\therefore \eta\rho \text{ is a maximum when } m = \frac{2Bv^3d}{As^2 - Bv^2s}$$

From the experimental results plotted in Fig. II. it appears that when s is equal to d, $\eta\rho$ is a maximum at about 600 feet per minute, or 10 feet per second, and by trial it is found that $\frac{1}{20}$ is a probable value of B. Taking A in the first instance at 100 and s and d each equal to ⅓ foot

$$m = \frac{\frac{1}{20} \times 1000 \times \frac{1}{3}}{\frac{100}{9} - \frac{100}{120}} = \frac{\frac{1000}{60}}{\frac{3700}{360}} = \frac{60}{37} = 1{\cdot}62162$$

And this value of m is used throughout in plotting the curves shown in Fig. III.

The constants A, B, and m are affected by alteration of conditions. A depends, among other things, on the quality of the explosive mixture, compression, and ignition; B on such considerations as area of inlet and exhaust valves and passages and on valve timing; m depends principally on the temperature of the cooling water. All such conditions are assumed to be constant unless otherwise stated.

In Fig. III. the curves of mean effective pressure and horse-power numbered 1 are supposed to be those of an engine having four cylinders, 4in. bore by 4in. stroke. A is taken as 100 and B as $\frac{1}{20}$. In the curves numbered 2 the conditions are altered by doubling the stroke; in those numbered 3, by increasing the stroke to 6in.; in those numbered 4, by increasing A to 125; and in those numbered 5, by reducing B to $\frac{1}{30}$. In each case the curves Nos. 2, 3, 4, and 5 represent curve No. 1 with one condition altered.

From curves No. 2 it will be seen that, on the assumed equation, the effect of increase of stroke is to increase the maximum horse-power, to increase the speed at which the maximum horse-power is exerted, and to increase the speed at which the maximum $\eta\rho$ is reached. Also the maximum $\eta\rho$ and $\eta\rho$ at maximum h.p. are diminished.

Curves No. 3 exhibit the increased $\eta\rho$ due to increase of cylinder diameter, and the decreased speed at which the maximum $\eta\rho$ is reached, due to diminished stroke-bore ratio. It is not quite clear on the diagram (Fig. IV. shows it better), but the speed at which the maximum h.p. is reached is slightly less in curves No. 3 than in No. 1.

Curves No. 4 show the effect of increasing A. The horse-power is, of course, increased, as also are the speeds at maximum h.p. and at maximum $\eta\rho$.

The reduction of B, as shown by curves No. 5, has, as might be expected, an effect somewhat similar to that of the increase of A.

In Fig. IV., curves No. 1, we have the same original engine as in curves No. 1, Fig. III. Curves No. 2 and No. 3 show the effect of an increased stroke-bore ratio in both cases. In curves No. 2 the increased ratio is arrived at by doubling the stroke, and in curves No. 3 by halving the diameter. Curves No. 4 are part of curves No. 3 increased with regard to curve No. 1 inversely as the squares of the cylinder diameters. The difference between curves of horse-power Nos. 1 and 3 is, therefore, the difference due to more rapid cooling of the smaller cylinder. In Fig. IV. I have marked with arrows the calculated maximum points.

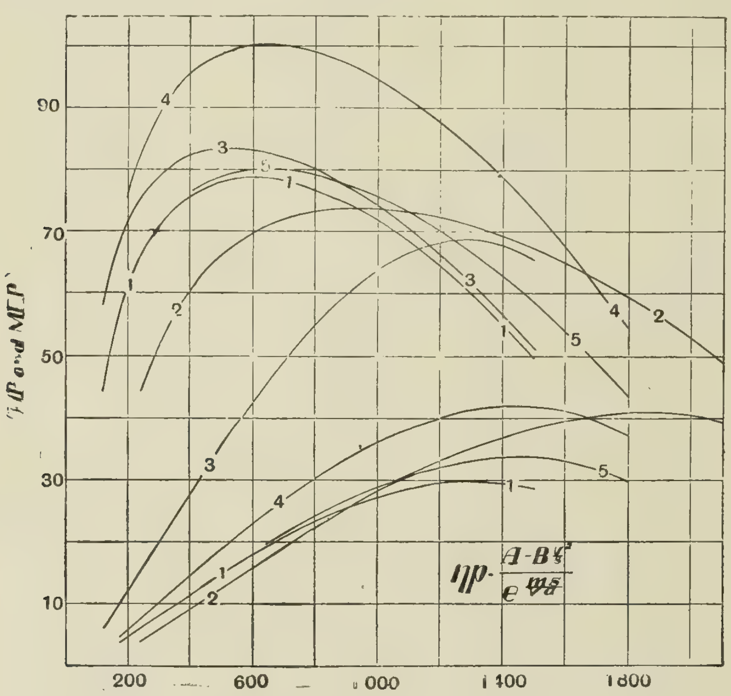


Fig. III.

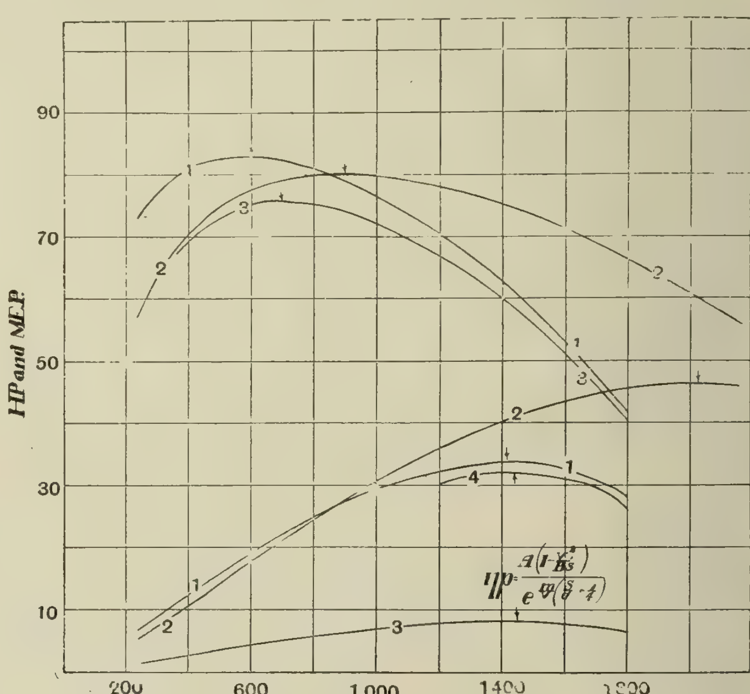


Fig. IV.

It is hardly to be expected that the admittedly erroneous formulæ which I have used in plotting the curves in Figs. III. and IV. can give accurate numerical results. They can only represent tendencies. It is apparent, however, that the curves of Figs. III. and IV. could be mixed with those of Figs. I. and II. without any certainty that, if the numbers were obliterated, the difference between the natural and the artificial could be detected. Applying the method of averages, dividing the curves of Figs. III. and IV. into those which have a stroke-bore ratio of unity or less and those which have a stroke-bore ratio greater than unity, we get the following comparison:

Stroke-bore Ratio.	σ at Max. H.P.	σ at Max. $\eta\rho$.	Max. $\eta\rho$.	$\eta\rho$ at Max. H.P.
0.93	1375	604	84.8	64.1
2.00	1760	850	76.3	57.16

This is fairly comparable with the averages obtained from the thirty-six engines from which the results shown in Figs. I. and II. are selected.

It would be futile without much fuller information to attempt to account for all the diverse results tabulated in Figs. I. and II. It becomes possible, however, by means of a single example to illustrate the apparent divergence from type that may exist when engines are classified solely by stroke-bore ratio. Engine No. 137, Fig. I., has a fairly high stroke-bore ratio, but when it is measured by its concrete dimensions it is a small-bore engine with a short stroke, and it embodies most completely all the appropriate vices. The mean effective pressure is low, owing to the small bore; the speed at maximum h.p. is low, owing to the short stroke; and the speed at maximum $\eta\rho$ is high, owing to the high stroke-bore ratio accentuated by the small bore. Such an engine is peculiarly inflexible, and, as a matter of fact, its record puts its useful limit of piston speed between 800 and 1,200 feet per minute.

Unlike the reciprocating steam engine, which is not designedly subject to external cooling, the internal combustion engine has two critical speeds. The first of these is reached at the point of maximum $\eta\rho$, at which point, or at any lower speed, the engine is, so to speak, in unstable equilibrium, and the least increase of load will slow the engine. The second critical speed is the speed at maximum h.p., and between these speeds the engine is truly flexible. It is important for motor car work that this range of speed should be as great as possible, and in this respect the long-stroke engine has an advantage. A fictitious flexibility may, however, exist at both ends of the speed scale. In Fig. V. A B represents an imaginary curve of resistance horse-power, that is, load multiplied by velocity. The engine horse-power and mean effective pressure are shown by curves similar to those in previous figures. Although the maximum $\eta\rho$ of the engine occurs at D, the engine will still possess relative flexibility down to the speed corresponding to C. The flexibility at this end of the scale may be described as accidental.

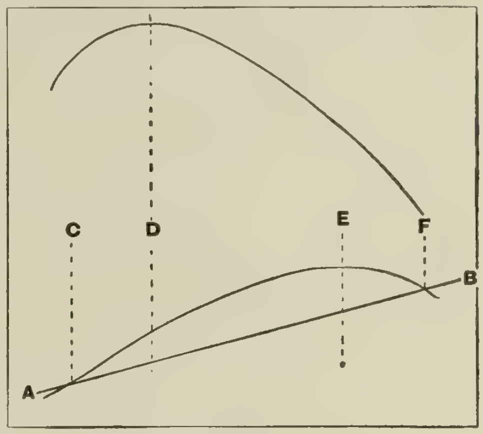


Fig. V.

At the other end of the scale the added flexibility between E and F is gained by employing a more powerful engine than is necessary. The engine cannot exert its full power, and at any speed between C and F must run throttled. If the engine is subject to rating disabilities and it is desired to take full advantage of the rated horse-power, the gearing must be of such proportions as will make the line A B cut the horse-power curve at E. It is this consideration which fixes the diameter of locomotive driving wheels, and which accounts for a good many differences commonly attributed to the efficiency or want of efficiency of marine propellers.

(To be continued.)

THE THEORY OF THE NORTON RUNNING BALANCE MACHINE.

By George S. Bower, B.Sc., (Eng.)

THE following analytical investigation of the action of the above machine was carried out in order to explain the results of experiments performed at the Crossley Works, with the object of finding out the effects of various conditions of "out of balance," so as to be able to apply the results to the case of any rotating part which it might be desired to balance. A curious feature of the machine is that unbalanced bodies rotated in it tend, after a certain speed is passed, to rotate about their centres of gravity, this "critical" speed being different for different rotating bodies. This phenomenon is also observed in the case of the De Laval turbine wheel which is mounted on a flexible shaft supported in rigid bearings, whilst in the case of the Norton machine almost rigid shafts are supported by elastic bearings.

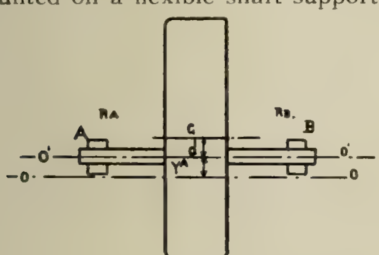


Fig. 1.

In the latter machine the supports for the rotating shaft can only move horizontally against the elastic resistance of rubber discs and are free to swivel about vertical axes, so that, when finding equations giving the maximum displacements of points along the centre line of the shaft from their initial positions when the shaft is at rest, the centre of gravity of the rotating mass must be taken to be in a horizontal plane through the centre line of the shaft, in which position it will, of course, have the greatest disturbing effect on the elastic supports.

The initial axis of the shaft, i.e., its axis when it is at rest, will be its instantaneous axis of rotation.

The application of the ordinary conditions of equilibrium of a rigid body, together with the equations resulting from the assumptions that the shaft is rigid and that the reactions at the supports are proportional to their displacements, will suffice to determine the latter displacements at any speed of rotation.

Flywheels.

It will be convenient to consider first the simple case of a flywheel placed centrally between the supporting rollers A and B, with its centre of gravity G not coinciding with the centre of its spindle. This is shown in Fig. I, which represents a plan view of the machine, A and B being the supporting rollers, of which the reactions R_A and R_B on the shaft are as indicated. O O represents the axis of the shaft when not rotating and is the instantaneous axis of rotation of the wheel, whilst O' O' is the actual position of the shaft when its displacement from O O is a maximum.

Let
y_A = maximum displacement of bearing A.
y_B = " " " B.
e = stiffness of bearings A and B.
R_A = bearing reaction at A.
R_B = " " B.
n = revolutions per second of wheel.
w = angular velocity of wheel.
M = mass of wheel.
d = distance of the centre of gravity of the wheel from the centre of the shaft.

Then, since the wheel is midway between the supports, $y_A = y$ by symmetry.
Also $ey_A = R_A$, and $ey_B = R_B$.
The distance of G from OO $= y_A + d$.
∴ the centrifugal force on the wheel will be $= Mw^2 (y_A + d)$.
Equating this to the sum of the bearing reactions R_A and R_B:—

$$Mw^2 (y_A + d) = R_A + R_B = 2ey_A.$$

or

$$y_A = y_B = \frac{-Mw^2 d}{Mw^2 - 2e} \qquad \ldots \qquad \ldots \qquad (1).$$

value $\sqrt{\frac{2e}{M}}$ the movement of the bearings increases to the value infinity: when this value of w is exceeded, y_A becomes more and more negative, until, when $w = \infty$, y_A becomes $-d$, that is to say the wheel rotates about its centre of gravity.

Of course, when $w = \sqrt{\frac{2e}{M}}$ the vibration of the bearings

does not become infinite, but whilst this speed is being run through the vibration of the pointers becomes very great, decreasing when the speed is increased. It is interesting to note that the natural frequency of horizontal oscillations of the wheel on its supports is given by $n = \frac{1}{2\pi}\sqrt{\frac{2e}{M}}$, thus being equal to the number of revolutions per second at which the vibration of the pointers is a maximum.

Crankshafts.

The case of a four-throw crankshaft supported at A and B, as shown diagrammatically in Fig. II., will now be dealt with by first obtaining the general equations for the case of four unequal throws, and afterwards applying them to various unbalanced conditions of the shaft.

Fig. II. is a plan view showing the directions of the bearing reactions for the displacements of the shaft shown in Fig. III.

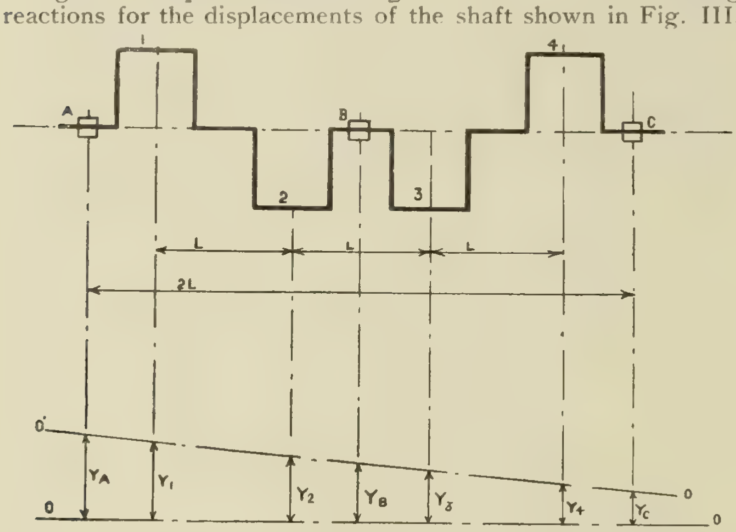


Fig. II.

(also a plan), in which O O is the initial or non-rotating position of the axis of the shaft, and O' O' is the position of the axis when its horizontal displacements are a maximum, y_1 y_2 y_3 and y_4, being the displacements of the respective crankpins, y_A and y_B the displacements of the supports A and B. All displacements are reckoned positive when in the direction of throw (1), as shown.

Let r = the radius, or throw, of each crank.

M_1, M_2, etc. = the masses of the respective cranks, referred to the crank pins.

w = the angular velocity of rotation.

n = revolutions per second.

e = the stiffness of supports A and B.

L = the cylinder centres.

Then, for equilibrium of the forces acting in the horizontal plane, we must have :—

$$M_1 w^2 (r+y_1) - M_2 w^2 (r-y_2) - M_3 w^2 (r-y_3) + M_4 w^2 (r+y_4) = R_A + R_B \quad \ldots \quad \ldots \quad \ldots \quad (2)$$

Also, taking moments about B, and equating to zero, since the supports are free to swivel horizontally :—

$$M_1 w^2 (r+y_1)\frac{5L}{2} - M_2 w^2 (r-y_2)\frac{3L}{2} - M_3 w^2 (r-y_3)\frac{L}{2}$$
$$- M_4 w^2 (r+y_4)\frac{L}{2} - 2L\,R_A = 0 \quad \ldots \quad \ldots \quad (3).$$

Since the bearings are elastic,

$$e\,y_A = R_A \quad \ldots \quad \ldots \quad \ldots \quad \ldots \quad \ldots \quad (4).$$
$$\text{and} \quad e\,y_B = R_B \quad \ldots \quad \ldots \quad \ldots \quad \ldots \quad \ldots \quad (5).$$

whilst, if the shaft remains straight, we shall have (see Fig. III.),

$$y_1 = \tfrac{5}{4}\,y_A - \tfrac{1}{4}\,y_B \quad \ldots \quad \ldots \quad \ldots \quad \ldots \quad (6).$$
$$y_2 = \tfrac{3}{4}\,y_A + \tfrac{1}{4}\,y_B \quad \ldots \quad \ldots \quad \ldots \quad \ldots \quad (7).$$
$$y_3 = \tfrac{1}{4}\,y_A + \tfrac{3}{4}\,y_B \quad \ldots \quad \ldots \quad \ldots \quad \ldots \quad (8).$$
$$y_4 = -\tfrac{1}{4}\,y_A + \tfrac{5}{4}\,y_B \ldots \quad \ldots \quad \ldots \quad \ldots \quad (9).$$

Substituting for R_A, R_B, y_1, y_2, y_3, and y_4 in terms of y_A and y_B in equations (2) and (3), the latter reduce to a couple of simultaneous equations in y_A and y_B. The solutions of these equations are given for the following cases where there is a definite relationship between the masses :—

First Condition.

Suppose throws (1) and (4) are equal in weight, but heavier than throws (2) and (3), which are also equal to one another. Let the equivalent masses of throws (1) and (4) be each = $M + m$; those of throws (2) and (3) being each = M. Then using equations (2) and (9) it is found that :—

$$y_A = y_B = \frac{-\,2m\,w^2 r}{4\,Mw^2 + 2\,mw^2 - 2\,e} \quad \ldots \quad \ldots \quad (10).$$

which shows that as w increases from zero, y_A and y_B are each positive until

$$4\,Mw^2 + 2\,mw^2 - 2\,e = O \quad \text{or } w = \sqrt{\frac{2\,e}{4\,m + 2\,m}} = w_C.$$

after which they are both negative until $w = \infty$, when

$$y_A = y_B = \frac{-\,2\,mr}{4\,M + 2m}$$

At the instant when they change sign, the values of y_A and y^B are each equal to $-\infty$.

From these results it may be inferred that, as the speed of the machine increases from zero, the scribers at A and B will first of all mark the shaft on the same side as throw (1), i.e., the heavy side (since y_A and y_B are positive), until the angular velocity reaches the critical value w_C, as shown by the increased oscillation of the pointers, after which they will mark it on the light side, i.e., the side opposite throw (1), (since y and y_B are negative).

These inferences were proved to be correct by actual experiment with the machine. It may be noted that the critical speed, w_C, mentioned above, corresponds to the natural frequency of oscillation of the shaft on its supports.

Second Condition.

Let M_2 and M_3 each = $M + m$
and M_1 and M_4 each = M.

In this case, which is very similar to that preceding, we find :—

$$y_A = y_B = \frac{2\,m\,w^2 r}{4\,Mw^2 + 2\,mw^2 - 2\,e} \quad \ldots \quad \ldots \quad (11).$$

from which we deduce that when w is small the scribers will mark on the heavy side of the shaft, i.e., opposite throw (1), whilst when w is fairly large they will mark on the light side of the shaft, the change being indicated by excessive oscillation of the pointers, and this has been found to be true experimentally.

Third Condition.

Let $M_1 = M_3 = M + m$
and $M_2 = M_4 = M$

Then it is found that :—

$$y_A = \frac{-\,2\,m\,w^2\,r\,(2\,M\,w^2 + \tfrac{1}{2}\,m\,w^2 - e)}{10M\,w^4 + \tfrac{45}{8}\,Mmw^4 + 2m^2w^4 - 9Mw^2e - \tfrac{9}{2}mw^2e + 2e^2} \quad \ldots \quad (12).$$

and

$$y_B = -\,y_A\,\frac{(2\,M\,w^2 + \tfrac{3}{2}\,mw^2 - e)}{(2\,M\,w^2 + \tfrac{1}{2}\,mw^2 - e)} \quad \ldots \quad \ldots \quad (13)$$

whence it follows that when w is very small y_A will be positive and y_B negative, whilst when w is fairly large, y_A will be negative and y_B positive. So that (as has also been proved

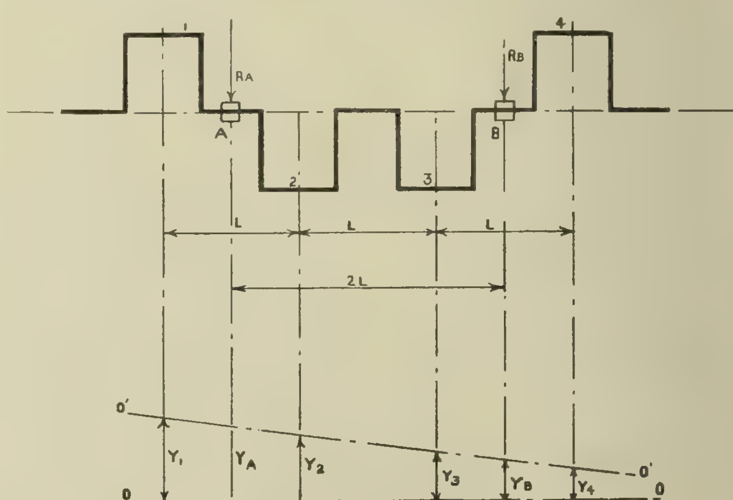


Fig. III.

by experiment), when w is large the scriber at A will mark on the side of the shaft opposite throw (1), and that at B on the same side as throw (1), this type of loading of the crankshaft causing it to wobble.

Fourth Condition.

Let $M_1 = M_2 = M + m$
and $M_3 = M_4 = M$.

In this case the shaft behaves similarly to the third condi-

tion, except that it does not cause so much vibration of the pointers.

Fifth Condition.

Let $M_2 = M + m$
and $M_1 = M_3 = M_4 = M$

In this case,

$$y_A = \frac{m\,w\ r\,(7\,M\,w^2 - 3\,e)}{20\,M^2 w^4 + 6\,M\,m\,w^4 - 18\,M\,e\,w^2 - \frac{5}{2}\,m\,e\,w^2 + 4\,e^2} \quad \ldots \ (14).$$

$$y_B = y_A\,\frac{(3\,M\,w\ - e)}{7\,M\,w^2 - 3\,e} \quad \ldots \quad \ldots \quad \ldots \ (15).$$

from which it will be seen that, when w is small, y_A and y_B are negative, and when w is large, y_A and y_B are positive, y_A being greater than y_B. Hence, when the shaft is run at high speed, the scribers at A and B will mark on the side of the shaft opposite throw (2), i.e., on the light side of the shaft, whilst the pointer at A will oscillate more than that at B. These results have been verified experimentally so far as the positions of the markings are concerned.

Sixth Condition.

Let $M_1 = M + m$
and $M_2 = M_3 = M_4 = M.$

Here, we have :—

$$y_A = \frac{-\,m\,w^2\,r\,(11\,M\,w^2 - 5\,e)}{20\,M^2 w^4 + 14\,M\,m\,w^4 - 18\,M\,w^2 e - \frac{13}{2}\,m\,w^2 e + 4\,e^2} \quad \ldots \ (16).$$

$$y_B = -\,y_A\,\frac{(M\,w^2 - e)}{(11\,M\,w^2 - 5\,e)} \quad \ldots \quad \ldots \quad \ldots \ (17).$$

When w is large, y_A is negative, and y_B positive, so that the scriber at A will mark on the side of the shaft opposite throw (1), whilst that at B will mark on the same side of the shaft as throw (1), the vibration of the pointer at A being much greater than that at B.

It has been found experimentally that the scriber at A marks on the side of the shaft opposite throw (1), and the pointer at A vibrates more than that at B, but that the scriber at B marks on the same side of the shaft as that at A. This discrepancy between the theoretical and the experimental results appears to be due to the bending of the overhanging end near B, since such bending will cause throw (4) to have a greater equivalent mass than assumed in the theory.

The above results are summarized in Table I. hereunder :—

TABLE I.

Displacements of supports.	Type of Loading. Conditions :—					
	1	2	3	4	5	6
y_A	−	+	−	−	+	
y_B	−	+	+	+	+	+

Crankshaft supported at three places.

When a crankshaft is supported at two points only, such as A and B in Fig. II., trouble is experienced, when it is desired to run it at a high speed, owing to the bending of the overhanging ends under the action of the centrifugal forces on the throws (1) and (4), the central portion also tending to bend because of the centrifugal forces on cranks (2) and (3).

The result of this bending is that the shaft, even though in perfect balance at moderate speeds, may be out of balance at high speeds, and, to avoid this taking place, it is desirable that the shaft should be supported at three places, such as A, B, and C, Fig. IV.

The signs of y_A, y_B and y_C (see Fig. V.), corresponding to the six cases of loading treated above—and assuming w throughout to be large—will be as shown in Table II. below, it being understood that when y_A, y_B or y_C is positive the scriber at A, B or C respectively will mark on the same side of the shaft as throw (1).

TABLE II.

Displacements of supports.	Type of Loading. Conditions :—					
	1	2	3	4	5	6
y_A	−	+	−	−	+	−
y_B	−	+	+	+	+	−
y_C	−	+	+	+	+	+

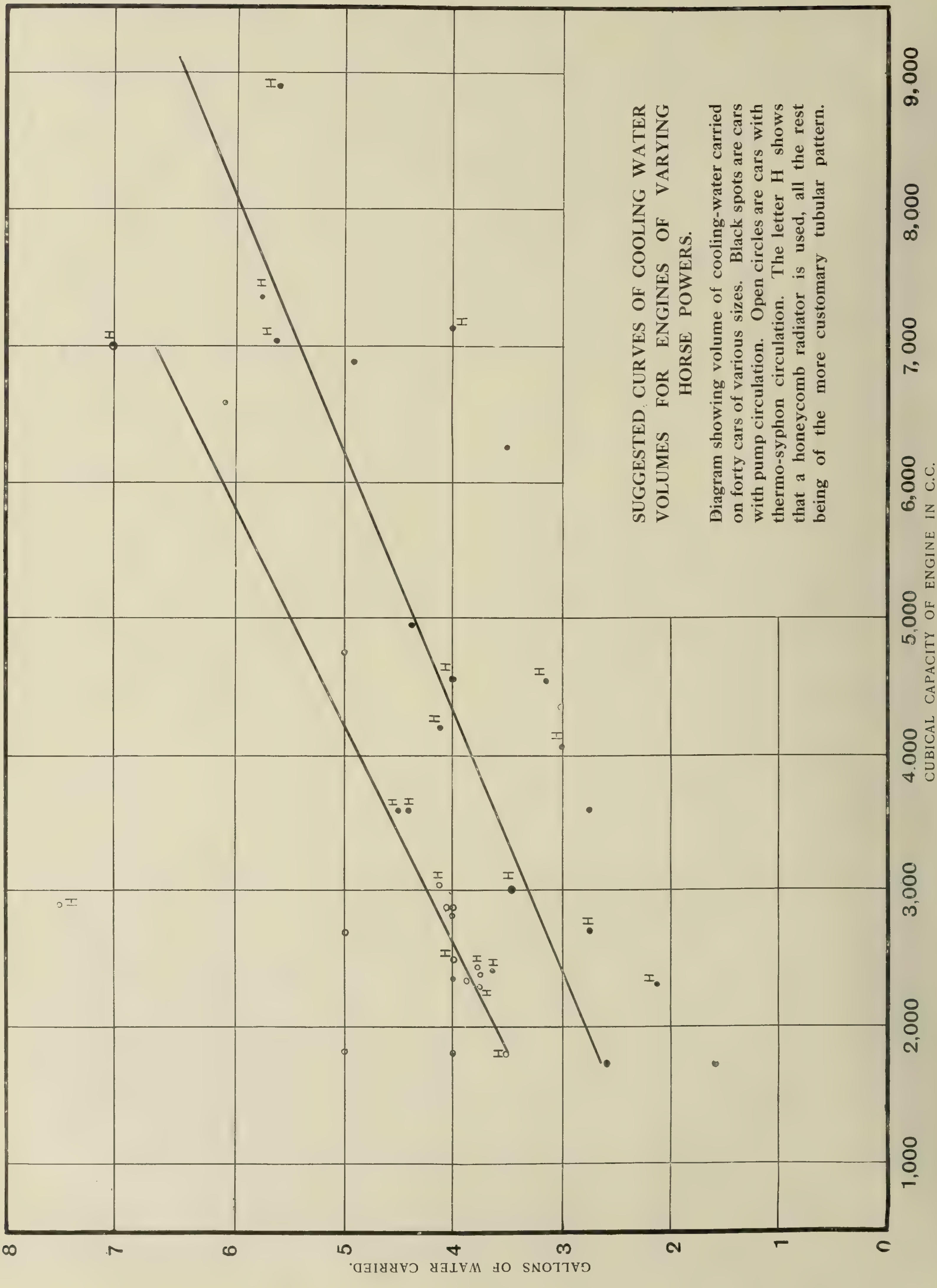
SOME POINTS RELATING TO WATER COOLING.

With a diagram showing the amount of water carried for each of forty typical engines.

An examination of present practice as regards the cooling arrangements of motor-car engines shows that there is very little agreement betweeen one designer and another. Not only are the relative advantages and disadvantages of forced and natural circulation still a matter of debate, but the nature of radiators, and still more the volume of water carried for each brake horse power the engine is capable of developing, exhibit no sort of agreement when they are examined side by side. The principal advantage of thermo-syphon circulation is that by its use the cost of manufacture of the pump is saved, and (from the owner's point of view) a thermo syphon system has no glands to leak, no moving parts to wear and never requires adjustment. Against natural circulation however, must be placed the fact that all pipes and water ways require to be larger than is the case when the water is delivered to the cylinders under pressure. This means increased cost and increased weight, while it would be reasonable to assume that the radiator in a natural flow system ought to be larger, and the volume of water carried larger likewise. So it appears that so far as the manufacturing cost is concerned, a thermo syphon cooled engine would be practically as costly to make as the pump-cooled type.

Again, from the user's point of view, circulation which depends upon convection has a bad fault in that it is essential always to maintain the water level at such a height that the outlet from the cylinders to the radiator is always submerged. In order as far as possible to ensure this, it would be advisable if radiator makers would allow for a considerable body of water normally to be carried above the outlet. In very many cars at present using thermo syphon circulation the loss of considerably less than one pint of water is sufficient greatly to impair, if not altogether to stop, circulation; in even the most efficient and best made systems it does not take very long to evaporate this amount, particularly in warm weather.

However, to return to the main subject, it would seem that there ought to be some rule for determining approximately the volume of water necessary to cool an engine of given b.h.p. With the object of discovering some such rule based on actual practice we have obtained the volume of water carried on a number of well-known cars (to the makers of which we are indebted for the information given below) and from this data the diagram has been plotted. In this there are forty different engines, and they are distinguished as follows :—The black dots represent engines cooled by pump circulation and the open circles engines cooled by thermo syphon, while the symbol H placed beside either mark indicates that the radiator is of honeycomb type. It will at once be seen that there is no striking contrast between the amount of water used by one type as compared with the other. Certainly the average volume of water per b.h.p. for engines with convection circulation is slightly in excess of that for the others, but the difference is surprisingly small. It might be remarked that the lowest values of all,



SUGGESTED CURVES OF COOLING WATER VOLUMES FOR ENGINES OF VARYING HORSE POWERS.
Diagram showing volume of cooling-water carried on forty cars of various sizes. Black spots are cars with pump circulation. Open circles are cars with thermo-syphon circulation. The letter H shows that a honeycomb radiator is used, all the rest being of the more customary tubular pattern.
GALLONS OF WATER CARRIED.
CUBICAL CAPACITY OF ENGINE IN C.C.
1,000
2,000
3,000
4,000
5,000
6,000
7,000
8,000
9,000
0
1
2
3
4
5
6
7
8

which are in each case for pump circulation, refer to cars with Knight engines in which the piping is extremely simple, while there are, of course, no valve pockets to cool, this accounting doubtless for the very low values.

This point leads one naturally to a digression which is possibly well worth making because it is a point that has not previously been taken very much notice of, although its importance is possibly quite great. This is the fact that the Knight engine, at all events as made and fitted by the Daimler Company, appears to need an extremely small amount of cooling. It must not be forgotten that the head castings of the cylinders are approached by somewhat narrow waterways, are in themselves awkwardly placed, and the cup shape quite prevents any possibility of convection circulation. Notwithstanding this though, in the ordinary English climate the Daimler Knight engine runs needlessly cool without a fan behind the radiator, and the pump in this case is by no means powerful. Of course the tubular type of radiator used in connection with this particular car carries a fair amount of water, but not an exceptional amount for the size of engine. If it be granted that it is an advantage to cut down the dead weight of a car even by the small amount which it is possible to save in water, and if we follow up the argument from the consideration of the Knight engine, it seems that simple jacket forms and simple passages have a very much greater effect on the efficiency of the water as a cooling agent than would be expected.

The two highest values in the diagram may probably be neglected, the highest of all being obviously very unnecessarily large. The two lines drawn across the figures probably represent the average of standard practice for pump circulated and syphon circulated engines, certainly if the large number clustered together on the 2,000 cc. to 3,000 cc. lines are correct (and these all refer to modern cars) then the syphon line is not too low. Very possibly the pump line could be dropped still further, this depending upon the power of the pump.

This leads to a point which was raised by very many of the manufacturers who supplied the data of the diagram, which was that the arrangements for draught through the radiator and the efficiency of radiators vary very greatly, wherefore a low value does not in any way necessarily mean that a car will be liable to overheating. As a matter of fact at the present day cars are more often over-cooled than under-cooled, and speaking quite broadly the smaller the amount of water which has to be carried the better; provided cooling is quite satisfactory. Obviously therefore, it behoves makers to ensure ample air draught through their radiators, and then to cut down the amount of water as much as possible consistent with efficient cooling.

Taking the bunch of modern cars which have been referred to above, that is to say those ranging in volume from 2,000 cc. to 3,000 cc., it will be noticed that they nearly all carry approximately four gallons of water weighing, of course, 40 lbs., whereas the average for pump cooled cars in this section alone would be about 3 gallons. Probably the majority of these engines develop about 30 b.h.p., so a very rough and ready rule appears to be a gallon per ten maximum b.h.p. of the engine for pump circulation, and a gallon and a quarter per ten b.h.p. for syphon engines cooled by syphon circulation.

If the two lines are followed across the figure, however, this approximation appears not to be accurate; for instance the engines on the 7,000 cc. line would probably develop about 80 b.h.p., and the largest amount of water carried is only just over 7 gallons, the average being say $5\frac{1}{2}$ for pump circulation and six-and-a-half for syphon. This is reasonable because, as the size of passages and water ways generally increase, the ease of flow improves likewise, and a smaller proportion of water ought to give all the cooling which is needed. The absence of many large cars without a pump makes the upper end of the syphon line largely a matter for speculation, whereas there is at least some data for the pump line. Obviously the amount of water used is largely a matter of chance or of trial and error.

There is undoubted agreement as to the amount of water needed by a 30 b.h.p. engine, which can scarcely be accident, and one other thing may be taken as certain. This is that an engine needs no more water than is represented by the two curves, and can probably perform with complete satisfaction in this country with a considerably smaller amount if the radiator has sufficient area and the various passages and water ways are sufficiently large.

So far only practical figures have been considered, and the real theory of water-cooling has not been approached at all. To attack the subject mathematically it might be assumed that the cylinder had to be maintained at a temperature not exceeding two hundred degrees Fahrenheit, that the average temperature inside the cylinder was—so much (calculable from the horse power), that the mass of the cylinder was—so much, etc. Owing however, to the enormous number of approximations which would have to be made, this is scarcely a promising method of arriving at a practical conclusion. Even however, if one could find the exact number of British thermal units which would have to be dissipated to the atmosphere, one would be but little nearer to the solution of the chief problem because the efficiency of the radiator and the restrictions of free movement of the water would be quite impossible even to guess. Still the efficiency of the water as a heat distributor or transferer is an interesting, small and corollary point, and may be determined roughly from the diagram. Sometimes in testing engines on the bench the temperature of the cooling water is taken carefully, and from this the amount of heat wasted in the water may be determined. In certain particular instances with which the writer is acquainted, the amount of heat so lost is equivalent to nearly half the brake horse power of the engine, that is to say nearly half the heat value of each explosion is being deliberately thrown away. As most engines when undergoing a bench test are considerably over-cooled, it is probable that there is no need to dissipate more than thirty per cent. of the power of the engine, if so much even.

Returning again to the bunch of engines which we have mentioned as coming between the 2,000 cc. and 3,000 cc. lines, supposing some further assumptions are made and their horse power is allowed to be 30, then it appears that ten horse power is the amount of work which three gallons of water can conveniently dispose of continuously, with ordinary modern equipment.

It may perhaps seem curious that so little has yet been determined concerning the true efficiency of radiators. There are an immense number of different tubular designs and more than several different types of honeycomb. Some of these must be much better for the purpose for which they are intended than others, and yet reliable information cannot be obtained concerning any one of them. The laboratory testing of a series of typical radiators would be a comparatively simple task, and the conclusions could not fail to be valuable to designers. It is therefore to be hoped that some public-spirited investigator with the necessary time and equipment (this ought really to include some crude form of wind tunnel) will some day feel disposed to take the matter up.

In some rough private experiments which were once conducted by a large firm for their own information the honeycomb radiator did not prove the most efficient, which may cause a certain amount of surprise to those who have never given more than casual thought to the subject. As a matter of fact, the most efficient of the quite small number of radiators tested was a pattern with vertical flat tubes, bent into wave-like form so as to give the neat appearance of a honeycomb when viewed from the front. Next in order of merit came a honeycomb with some other flat tube types, and next a vertical flat tube variety without gills or similar surface additions. This last proved to be very considerably more efficient than a well-made example of the round gilled tube pattern.

The first-named of the series, that is, the "waved," thin, vertical, flat tube example, was far ahead of the remainder and seemed to be quite the best.

However, to leave the subject of actual radiator design there is another point deserving of mention. Those who have been associated with the driving of racing cars will be well acquainted with the value of fresh cold oil for obtaining a momentary burst of extra speed, after the whole engine is really hot. This is not owing solely to the greater lubricating power of the cold oil, but to its actual cooling effect upon the metal with which it comes in contact, and after all this is in a sense reasonable enough, because all extra heat in the piston has to pass to the jacket water via a film of oil. Therefore it is more than likely that the regular efficiency of an engine might be improved if the oil were kept at a low temperature. This could be done without much trouble and, in conclusion, two methods may be suggested as worth experimenting with. One would be to divide off one or two tubes in the radiator and let the oil pass through them, but this would need somewhat special pumping arrangements. Another and better method would be to pass the cool water, from the bottom of the radiator, through the sump on its way to the cylinders' intake, by way of a few small pipes.

SOME PECULIAR AND SIMPLE JIGS

Which are used by the St. Louis Motor Car Co., of Ohio, for various regular and occasional purposes.

By C. T. SCHAEFER.

MANUFACTURING methods are always interesting, and the purpose of the writer in this article will be to show how some points in manufacturing, that have caused considerable worry and which are being given a con-

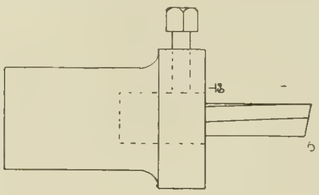


Fig. I.

siderable amount of study, may best be dealt with. The points to be discussed are the machining of square holes used for various purposes, the machining of eccentric piston rings, and a centre bearing for turning crankshafts to stiffen them

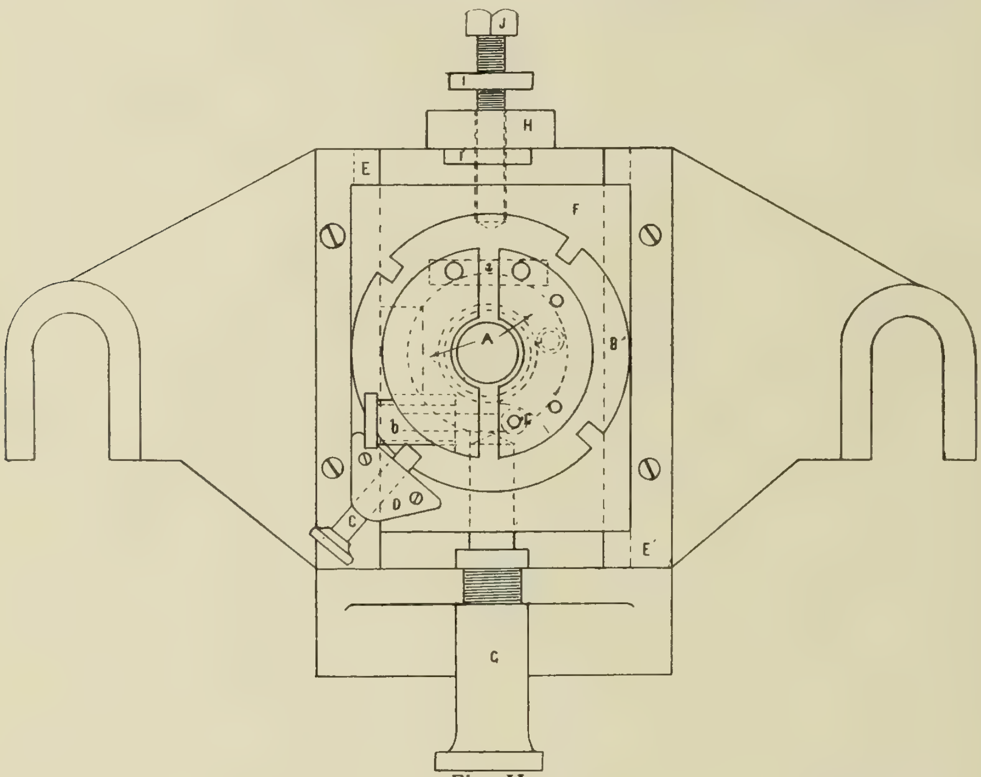


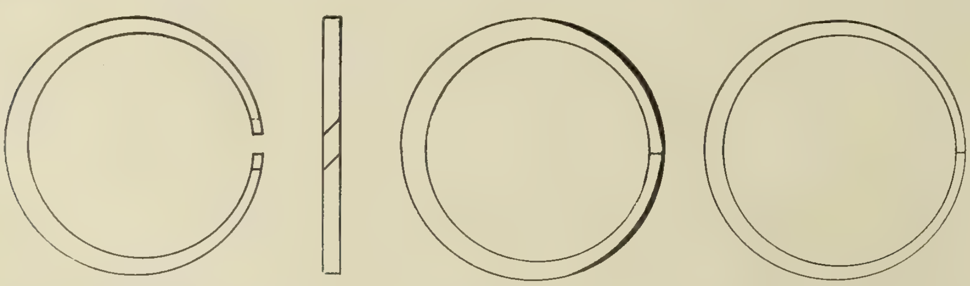
Fig. II.

while turning. These have been found to greatly assist in increasing the quality and quantity of work as well as decreasing its cost.

Machining a Square Hole.

Parts having square holes through them are quite common, and comprise such items as sliding gears, couplings and camshaft ends. In the past these holes have always been drifted, but it is certainly agreed that this is a very expensive operation. I will endeavour to depict a jig for machining square holes in a punch press instead of drifting them, and may add that couplings made in this way will run absolutely true if placed on an arbour and tested in a lathe. Fig. I. shows the punch for punching the hole, and this should be made with a clearance of 1-64in. from cutting edge to depth of cut, so that chips will not lead up the tool, while it should also have a shearing angle of 6°, this being derived from a series of experiments with various angles.

Fig. II. shows the jig, A being a vice to hold the work pivotting on a hinge and clamped by the nut B, which in turn is threaded on a rod end pivoting in C. The dividing device B is held in position by the pin C, which is supported by a spring in the piece D holding the dividing device in position. This dividing plate and vice are fastened on a plate F, which is grooved on the sides and guided by the small plates E. Its movement to and fro is controlled by the screw G, while H is a stop for the travel of the jig, having collars I locked in position by the set



A. B. C.

Fig III.

A. Imperfect eccentric ring. B. Black shading shows part which does not bear on cylinder wall. C. Perfect Ring.

screws J passing through a clearance hole in H. The vice is provided with split bushings to accommodate the various sizes and classes of work. This fixture is not very good for large work such as transmission gears and other considerable parts, but it has proved an excellent device for key-seating smaller parts in emergency cases.

Machining Eccentric Piston Rings.

For a number of years it has been the practice of manufacturers to put piston

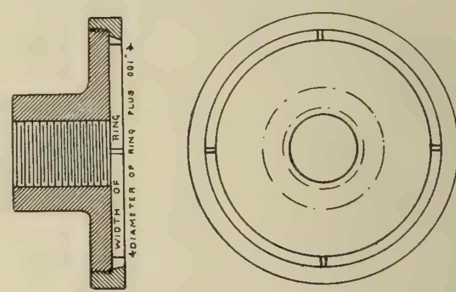


Fig. IV.

rings through quite a number of operations, some of them going so far as to grind their rings, which is quite expensive where the output is limited, especially in view of the fact that special machinery must be purchased to accomplish the job.

The series of jigs which are described hereunder will be found to eliminate many operations and also the objection mentioned above.

It might perhaps be thought that the jigs are scarcely so elaborate as they should need to be in order to produce a perfect ring, but they are really quite sufficient. The secret of a perfect ring is to eliminate the lump which is liable to present itself at the slot when the ring is ready to place on the piston and then into the cylinder. Such a defective ring, bearing only on two high spots of the cylinder wall, is shown in Fig. III. It is needless

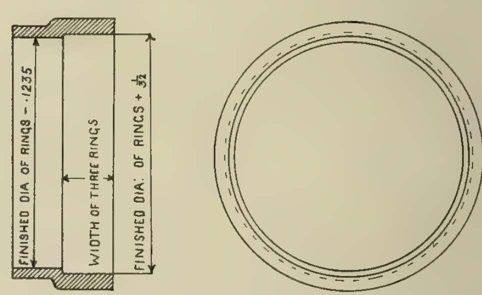


Fig. V.

to say how an imperfect ring may be detected.

The first operation is to turn the outside diameter of the ring eccentric, allowing 1-32in. in diameter for finishing, the inside diameter being finished already. The rings are cut off 1-64in. full of the required width, the 1-64in. being allowed to tone up the ring after it is slotted. For slotting, the rings are placed on a six-inch face plate which is recessed to centralize the ring; the slotting being accomplished by a hand miller, using a spacing collar to space the saws and allowing 1-64in. for filing the slots to the required opening. The rings

can then be placed in the jig, Fig. IV., and cut to the required width. I might say that with this device our rings vary but .005 in. in width, which is allowable. The hub of the fixture is, of course, made to fit the lathe spindle. The diameter

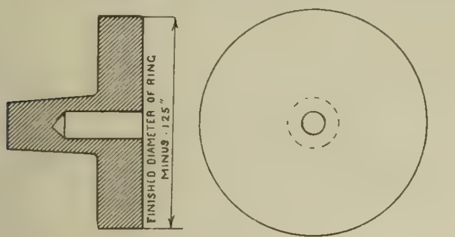


Fig. VI.

should be .001 in. larger than the outside diameter of the ring and have four 1-16in. slots cut into it to give the necessary tension to grip the rings when the nut is threaded on, the nut and fixture being tapered to get the action of a spring chuck, while the nut is also knurled to permit the jig to be used without the aid of any wrenches, making the operation quick and simple. When the ring has been placed in the jig a very light cut is taken, just enough to tone it up, and it is then reversed and finished to the required width.

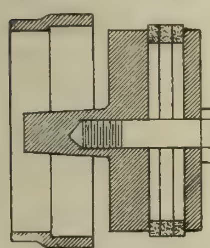


Fig. VIII.

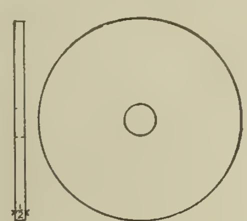


Fig. VII.

This should be done at high speed with a sharp pointed tool if it is desirable to get the best results.

The last operation is finishing the outside diameter, which is accomplished in the following manner. The rings are placed in the fixture shown in Fig. V., which can be made to hold from one to five rings, but I would advise finishing three rings at a time. The fixture, with the rings, is slid on the jig, Fig. VI., which is fitted to the lathe spindle, the rings being centralized from the outside diameter. The plate fixture is then placed against the rings (as shown in Fig. VIII.) and the plate and rings are clamped by a 1in. bolt.

Steady Bearing for Crank Turning.

This bearing has been found to be very useful in shops, and particularly in repair

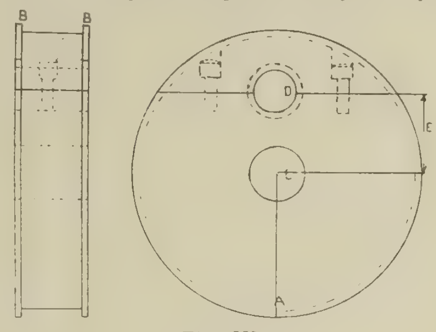


Fig. IX.

shops where crankshaft grinders are not available. It is very simple and made of cast iron, the outside being a running fit in the steady rest, while the shoulders hold the bearing in position (Fig. IX.).

The smaller hole C in the centre is merely for holding the bearing while the diameter is turned and the hole D is made to fit the diameter of the centre bearing and the distance E is equal to the throw of the crankshaft, or, in other words, the crank radius. The bearing is split on the centre line of the hole D, and the two pieces are held together by round head machine screws as shown. Where crankshafts of different throws and diameters are to be turned the hole D should be made larger so that split eccentric bushings can be used to make up this difference. When this is done the bearings will have to be recessed on both sides at D (as shown by the dotted lines) and the eccentric bushings will require a collar on each side to hold them.

The feature of this bearing is that the roughing cuts are eliminated, and the crank finished to proper grinding diameter in one operation. This is quite a reduction in cost, as setting up a machine for each operation means that the crank will have to be handled twice, both in turning and in inspection.

Jig for Connecting Rod Splitting.

The following is a simple fixture used in splitting connecting rods in a milling machine. The splitting of the rod is invariably the last operation to be performed and, as all surfaces are finished, there is very little trouble in finding a suitable locating point, but in view of the fact that this work is done by an apprentice, the fixture must be so constructed to permit the rod to locate itself.

A fixture having these features is depicted in Fig. X., being designed by the writer to meet these requirements. It is nothing more than a cast-iron angle plate finished all over. The pin A is driven into the plate (the projecting end is .001in. smaller in diameter than the gudgeon pin)

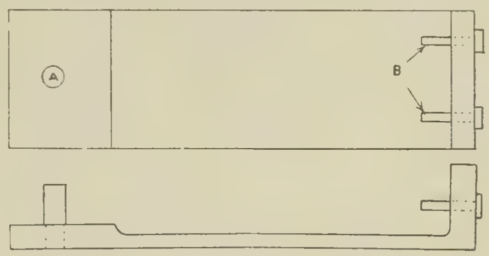


Fig. X.

and the two pins B locate and centralize the rod in position, as well as holding it and taking the thrust of the saw. This thrust will readily hold the pin in position and therefore no lockng device is required. As shown, this fixture requires the use of no wrenches.

STANDARDISATION OF SMALL DETAILS.

A brief account of some of the steps which have been taken by the Society of Automobile Engineers (U.S.A.) towards the standardisation of the smaller detail parts used in automobile construction.

THE now defunct Association of Licensed Automobile Manufacturers of America established a technical department to deal with matters of manufacturing interest, and especially questions relating to standards. This department was not long ago handed over to the S.A.E., who have always advocated very strongly a reasonable degree of standardisation, and the whole subject is so interesting that it is proposed to quote at fair length some recent remarks of two of the leaders of the Society, especially as it appears the Society's aims have been misunderstood in many quarters; some manufacturers appearing to hold the word "standardisation" in abhorrence under the mistaken impression that to assist in standardising anything would mean sinking their own individuality.

In a recent circular issued from the head office of the S.A.E. to the membership, it was suggested that very few manufacturers realised that there are in existence *three or four hundred* more dimensions and designs for lock washers alone, between the sizes of three-sixteenths and half-an-inch, than there ought to be, and the circular goes on to explain this as follows:—"A draughtsman is called upon to make a drawing of, say, an axle; his instructions are to draw an axle capable of carrying a car of some given weight, and that in a general way it should be an I beam axle of certain length and style. Entering into this axle are very many details, such as the size of hole for spring clips, size of pin and steering joints, etc., to accommodate the rest of the steering mechanism. Supposing the draughtsman exercises his own judgment as to these details: not having before him any standards, he may choose a size of tubing for the steering tie rod that does not exist; whereas if he has a table of sizes before him he would choose the tube the nearest possible to his ideal. Precisely the same argument applies to the shapes of the heads of screws and bolts for spring clip details and so on. Supposing the drawing be made, it will be passed, if the dimensions correspond and the general appearance is satisfactory. From the checker it goes to the purchasing department and the different pieces are ordered. It is then it may be found that the screws (say) ordered are special, and when this occurs usually the screw makers will enquire whether their nearest standard size will not be suitable. The only possible answer is in the affirmative except for the fact that other parts, to fit which the special screws were designed, will be already in the machine shop, and to alter them to suit a standard screw would cost more than the purchase of special screws."

Nor, remarks the circular, does the thing end here, because once a design has been accepted and become part of a motor manufacturing company's standard type, it is likely to be retained for several years in order to avoid the expense of changing jigs and so forth. Through all this time the special screws or other pieces will still have to be made and paid for at the special rate which must commercially be demanded for sizes other than one of the many general engineering standards.

The Society, with the assistance of the car-makers, are therefore at present engaged upon getting out a handbook, giving in tabular form the standard sizes of screws, tubing, lock washers, rod ends, grease cups, etc., within the limits of sizes used in automobile work. Such books have, of course, been published before in connection with other branches of engineering, and this additional volume ought to be of very great value to the American car designer.

Some interesting personal views.

Even though European readers may not agree with them in their entirety, the following views of the President of the Society, Mr. Henry Souther, and Mr. Howard E. Coffin, the ex-President, are distinctly interesting. In discussing the whole subject generally, the former recently said that in taking up the work of the S.A.E. with those who were not closely associated with that body, he discovered that many had a wrong idea of the form which standardisation should take, and also what it would accomplish, many having the idea that the Society desired the form, the design, and the shape of large portions of machines to be made all alike. Nothing could, of course, be further from the truth, for such a procedure would throttle originality and hamper work. Such a kind of standardisation could never be accomplished, even if it were attempted. What was aimed at was the standardisation of detail, such as screws, washers, spring parts, bearing parts, water connections and many other small pieces too numerous to mention, as it had been found that the number of those parts continued to multiply because a draughtsman exercises his ingenuity and his fancy in introducing something distinctive. There could be no possible gain from originality in such details as these, and multiplication of such parts ought to be minimised.

Mr. Coffin, speaking to a gathering of manufacturers on another occasion, said that any standardisation at the present time should not be an attempt to standardise the practice of engineers or the exercise of engineering ability, but should be an effort to standardise the practice of the junior designer by the method of placing upon the walls of the drawing office such reference tables as would guide him in working out the various problems that confront him. The things which should be standardised were the things which might just as well be standard as not, and these were the things which gave the maximum amount of trouble to purchasing departments, therefore costing most money in proportion to their real value. It was no more trouble to have articles of this kind designed from the point of view of convenience, so as to be correct when they left the drawing office, than to have the builders of component parts complaining that if such and such had only been made slightly different, the price quoted would have been half of what it actually had to be, because the specifications were not standard.

Continuing, on the subject of heavy vehicles, Mr. Coffin said :—"The complications under which the heavy car builders are going to labour will be far greater even than those under which the pleasure car makers are labouring at the present time. Everyone of you, or everyone of your engineering departments, will probably be specifying a different gauge of metal for every particular part of your wagon, not because there is any engineering reason why it should be so, but because the draughtsman whom you have employed will use his own good judgment. Your checker will pass it over because the dimensions will check, and it will go to your purchasing department. Your purchasing department will go to your steel maker to purchase it. Your steel maker will say, "That is a gauge that I do not carry regularly in stock," or "that is a size tubing that you will have to wait six weeks for." "That is a broach for a square hole that we do not carry in stock; it is two-thousandths different from anything we have, and therefore you will have to wait for it six weeks, and pay a special price."

Now, is it not just as easy for us to cut down the number of gauges that are in the market, the number of sizes of tubing, and the number of dimensions that are called for in broaches by getting these data together now and embodying them in engineering tables of reference which can be used just as are the rolling mill steel section tables, for instance, at the present time? Everyone of you who builds vans and employs a rolled section, will get out one of the steel makers' handbooks and look at the properties of the channel before you think of laying it out on the drawing board. Why should you not do that on the thousand and one other things that are absolutely within the draughtsman's jurisdiction, or within his ability to make either one way or the other? In other words, let us get the little things out of the inexperienced man's hands, and let us make it a matter of engineering practice. Some of you know, for instance, that the National Lock Washer Company are at the present time supplying to touring car builders six hundred different sizes of lock washers, most of them made to order, between the sizes of 3-16 diameter bolts and ½-inch diameter bolts.

Three years ago we found 1,600 different sizes of steel tubing specified by motor car makers, and being rolled by steel mills. At least one builder was himself specifying eighty different sizes of tubing. The statement has been made yesterday by the Standard Committee that within a year they would have the number of steel tubes necessary for motor car construction probably at fifty sizes, and certainly not above seventy-five. The difference bteween seventy-five and one thousand six hundred, it seems to me, is worth considering. It means quality, quantity output, and economic output. It means the cutting out of delay in every phase of your business."

The European View.

Although European manufacturers would probably consider that the scheme of standardisation outlined in the above speeches was rather too sanguine, it is to be doubted whether this is really so. Far more depends upon the spirit in which the information collected by the S.A.E. comes to be applied, than upon the actual information itself. It would be wrong for a man to determine that he must never use a bolt or a channel or a tube in a chassis that was not standard. It must not be forgotten that standards are made by demand. A steel maker's standards cover the requirements of the bulk of his customers, and *are* his standards *because* they are the requirements of the bulk of his customers.

Undoubtedly, automobile builders all the world over are using less of the special size year by year, and, so far as British factories are concerned, it is probably safe to say that far more material is now bought in a finished or semi-finished condition than was recently the case. Commencing by being almost entirely a special product the touring car chassis has (at a guess) come to be, in England, somewhere about one-tenth standardised (there are not quite so many varieties of screw threads as there were). On the Continent probably standardisation has advanced a little further; in America certainly it has advanced much further still.

Up to the present in the majority of cases designs are got out to meet certain conditions. The wise manufacturer, when the design is roughed out, but not before, examines his list of standards to see what of them will suit the particular design in hand. If it is then found that very slight alteration will make a great difference in the cost of the part, undoubtedly the cheaper way is the better way from the point of view of everybody concerned, including the ultimate owner. To work things the other way about however, to begin with the standard and to design as few special parts as possible to string standard pieces together, could not fail to debase the average excellence of the work.

A great deal of pure nonsense has been talked and written from time to time on the advantages of "standardisation" to users of cycles, motor cars and other common types of machinery. Parts which are liable to destruction by fair wear and tear should advantageously be easily obtainable anywhere, but if that easy obtainability is made a god of, it is quite simple to see that replacements would be needed far more often than would be the case were the machine designed primarily on engineering principles, by proportioning it to the work it had to do and disregarding all standards. It is easy to become extremely enthusiastic eitherfor complete standardisation or complete individualism. The ideal lies of course, somewhere between the two, but the automobile industry is now in a state in which individualism still requires to be given free scope.

We think the Society of Automobile Engineers is to be congratulated upon the work which it has undertaken. We believe the results of that work will be extremely useful, but we would urge again, as we have urged before, that designers in this country should not make too great haste to copy the procedure of factories with far larger average outputs than are common here. The British automobile industry holds its position today by virtue of the high quality of British cars, and it is of far greater importance to maintain that reputation than to reduce costs of production at the expense of ever so little merit.

THE 16 H.P. ADAMS CHASSIS.

A well balanced engine and an epicyclic transmission gear are the principal features of this chassis, which has several other points of interest.

By comparison with the majority of the 80 mm. by 120 mm. class the 16 h.p. Adams follows standard lines externally, but in detail there are many points of difference, and the additional five millimetres on the bore has an appreciable effect upon the behaviour of the engine. Taken as a whole the design is of a clean nature, and merits some praise on the score of the accessibility of the principal parts which are liable to need adjustment, while there is throughout a substantiality which suggests ability to withstand considerable hard usage. Simplicity of control in particular has always been considered to be of great importance by those responsible for Adams designs, and this fact accounts for the retention of the epicyclic transmission which is, of course, decidedly more costly to manufacture than a sliding gear. So much so it may be remarked, that the cost of erection alone is more than double that for the standard type. On the other hand the Adams gear is certainly extremely durable, so the price which has to be paid must not be regarded as being for convenience in use alone.

Leaving the question of the gear however, the engine exhibits several departures from standard practice as may be seen by reference to Fig. I. The cylinders are cast in pairs, and have rather large water spaces especially in the neighbourhood of the valve pockets, while the spherical section of the space immediately beneath the valve should be noticed, as it ought to afford unusually free passage for the gases. For the machining of the cylinders only two jigs are used, the first operation being to face the tops. As soon as this is done the casting is bolted down to a jig consisting of two plates, of which the smaller is tongued and slides in grooves in the larger. There are three holes in the plates, and a stop pin enabling the centre of either cylinder bore, or the midway position between these points, to be brought under the spindle centre of a vertical boring mill. In this machine the cylinder base is first faced up, starting from the midway centre, and then the cylinders are rough bored in succession. Finally a variety of floating reamer is run down each bore and leaves a really excellent finish so that grinding is not needed. When they have been bored the castings are put into a box jig which locates the valve chambers for boring, and every hole which has to be drilled. Cast iron is used for the pistons, and the gudgeon pin is secured by the much criticised set-screw. All the pistons and the connecting rods are balanced and the total reciprocating weight (the piston plus half the connecting rod) is less than three pounds, which is decidedly on the light side for an 85 mm. engine. The connecting rods are unusual in that the big end bearings are housed in bronze blocks, the connecting rods having flat ends (see Fig. I.). While this method makes for simple machining it is probably no better than the more usual system as regards efficiency, and it would be more expensive than running white metal directly into the steel of the rod and its cap, which is the most up to date practice.

For the crankshaft chrome-vanadium or nickel-chrome steel is used, and the peculiar design is shown in Fig. I. The purpose of extending the webs in the manner there shown is to balance the shaft more perfectly than usual by reducing the value of the couple set up as soon as the rotation commences. This crankshaft is in fact the equivalent of a disc-web crank, and it can certainly be said that the balance of the engine is above the average of merit. This is probably also partly owing to the substantial size of the shaft, which is 35 mm. in diameter, while the main bearings are supported in the crank case by thick webbing, and are therefore very rigid.

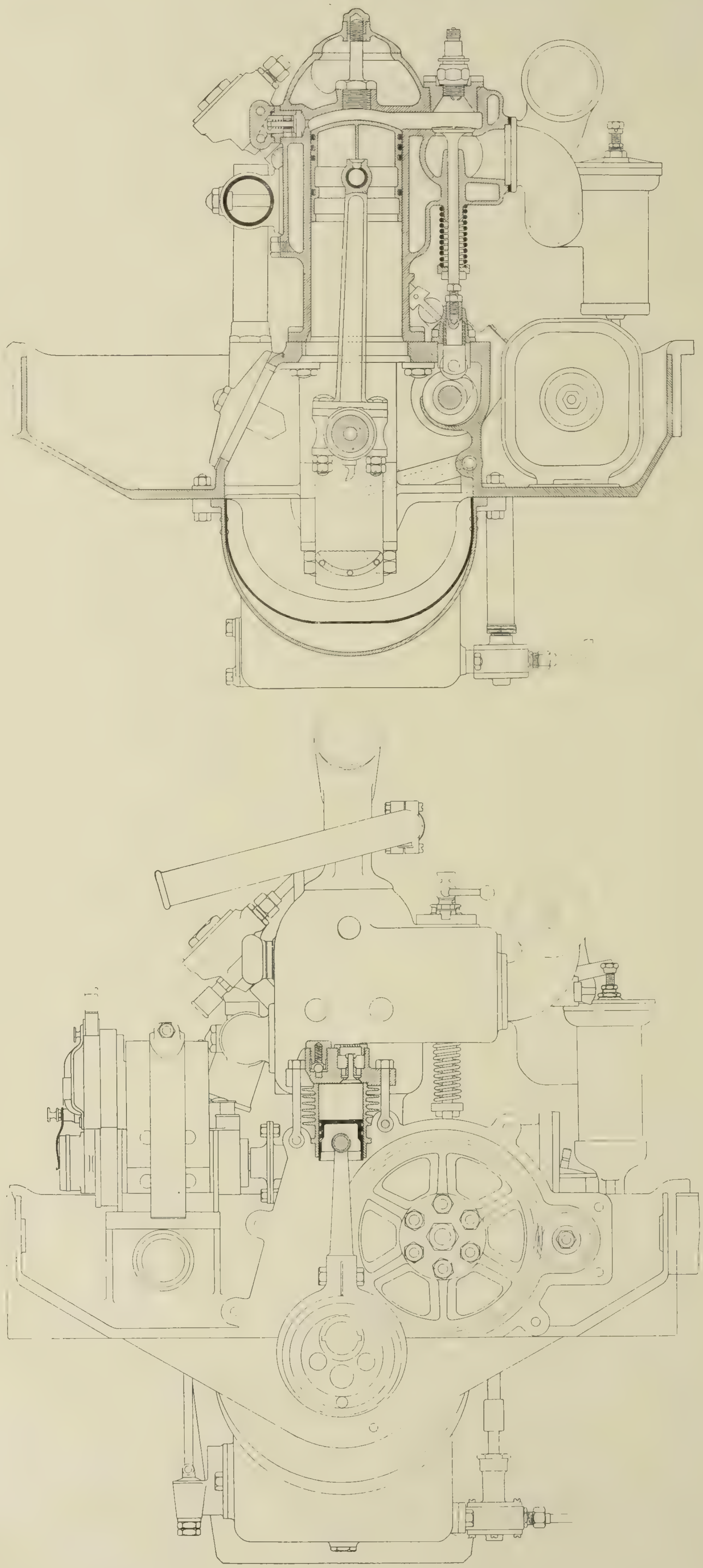
Loose cams are used made of Ubas case-hardening steel with a shaft of three per cent. nickel steel, and this shaft has three bearings like the crankshaft: it is 22 mm. in diameter, the tappet rollers being much larger than the normal. The valves are 35 mm. diameter at the bottom of the seating, and have a maximum lift of about 8 mm. Developing from thirty to thirty-five horse power at 2,500 r.p.m., the engine shows a fair efficiency and runs smoothly, as has already been remarked. As regards noise the exposed tappets are quiet, though still quite audible, but as the large cam rollers ought to help to reduce sound, it might be anticipated that the valve gear altogether would become quite quiet if it was enclosed.

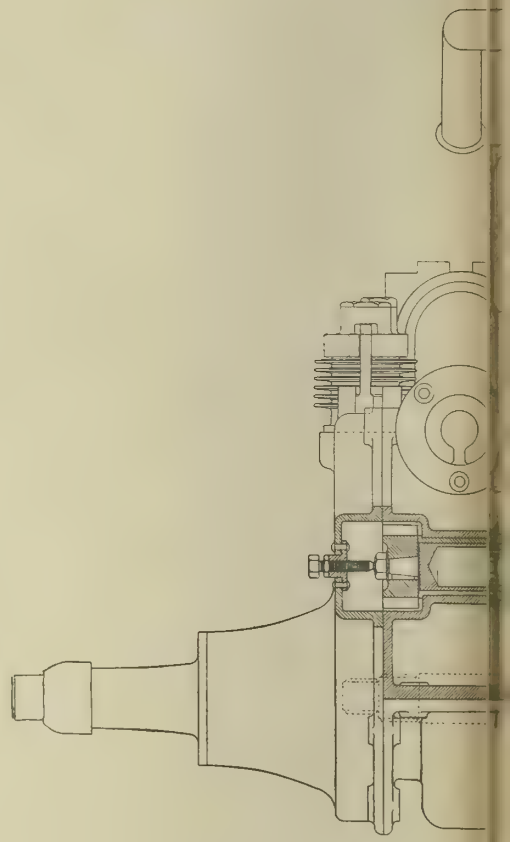
For lubrication a double system is employed, which is a combination of pressure feed and trough-controlled splash. The oil pump is external to the crankcase, and sucks from inside a large cylindrical filter at the bottom of the sump, while it is driven by a vertical shaft and a spur gear of the usual type at the rear end of the cam-shaft. This filter can be removed, without disturbing the pump, by unbolting a flange on the opposite side of the engine as may be seen in the cross-sectional views in Fig. I., and the pump can, of course, be detached with equal ease. From the pump oil is taken to a phosphor bronze pipe cast in the aluminium of the case, and from this it flows to the three main bearings, to a dashboard tell-tale and to the troughs (by way of small holes and corresponding jets). Contrary to usual practice the troughs are separate castings in aluminium which are rivetted into place, thus making a simple foundry job of the lower half of the base chamber.

At the last show the prevalence of the tray-form webs connecting the crankcase arms was very marked, and few designers have taken greater advantage of this form of construction than the makers of the Adams, for in this case there is a deep and wide channel at each side of the casting, running its full length so that the magneto and a C.A.V. lighting dynamo can be housed entirely below the cylinder level. Behind the latter the float chamber of the carburettor is situated, and the whole arrangement is extremely neat, while making for accessibility. A standard fitting is the Saurer or S.C.A.T. type of automatic starting device, and the exact method of application is such that the apparatus is not obtrusive and does not detract from the accessibility of other parts. For driving the pump there is a ball-bearing eccentric at the front end of the crankshaft, this being of the type made by the Auto Machinery Company without a cage and almost entirely filled with balls. The air valve distributer is driven from a skew gear at the centre of the cam-shaft by means of a shaft (with small universal joints) placed angularly between the cylinders, as shown inset in Fig. I. The air valves are horizontal, and are fed by a small cast iron pipe on the off side of the cylinders, which is placed just above the water intake pipe and below the distributing valve. For the magneto drive there is a cross-shaft driven by a skew gear situated immediately behind the large timing wheel, and it may be remarked that the usual material for the latter and the crankshaft pinion is good quality cast iron, though various fibre substances have been tried.

The carburettor is quite a simple type as is shown in Fig II, though the air orifice below the jet is fixed as regards its diameter, the area of the curved passage leading to the mixing valve can be adjusted by screwing down the cap above the jet: the adjustment for this can easily be marked so that the cap may be taken out when it is necessary to remove the jet. The mixture from the jet passage passes up the inside of the piston throttle and thence through a branched pipe to the cylinder pairs while, on top of the throttle chamber, there is a spring-controlled air valve faced with leather for quietness in running. As regards this valve, it should be noticed that the spring and the amount of lift can be adjusted separately, and this is claimed to be of great assistance when setting the carburettor.

Unlike most cars with epicyclic gearing the Adams has an ordinary pedal operated clutch, the details of which can be observed in Fig. I. It will be seen that it consists of a cast aluminium leather-faced cone, but is however, quite peculiar in some other respects. Firstly it will be noticed that the end of the crankshaft is a hollow cylinder inside which there slides a separate piece carrying the outer portion of two cup and cone ball bearings, while the clutch cone is mounted on the spindle and is released by being pressed inside the fly-wheel. The necessary lubricant is supplied by a small grease cap, seen in the illustration, and the bearing may be adjusted by screwing up the inner cone. At first sight this design may appear to be some-





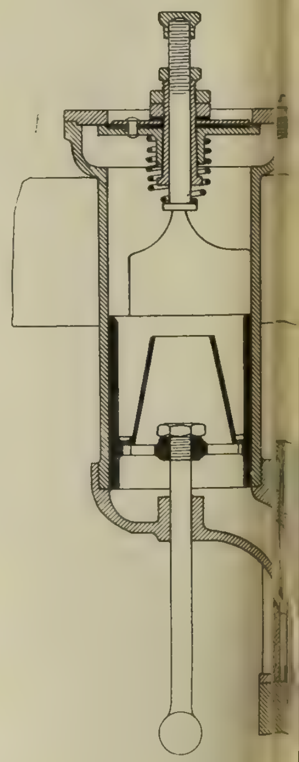
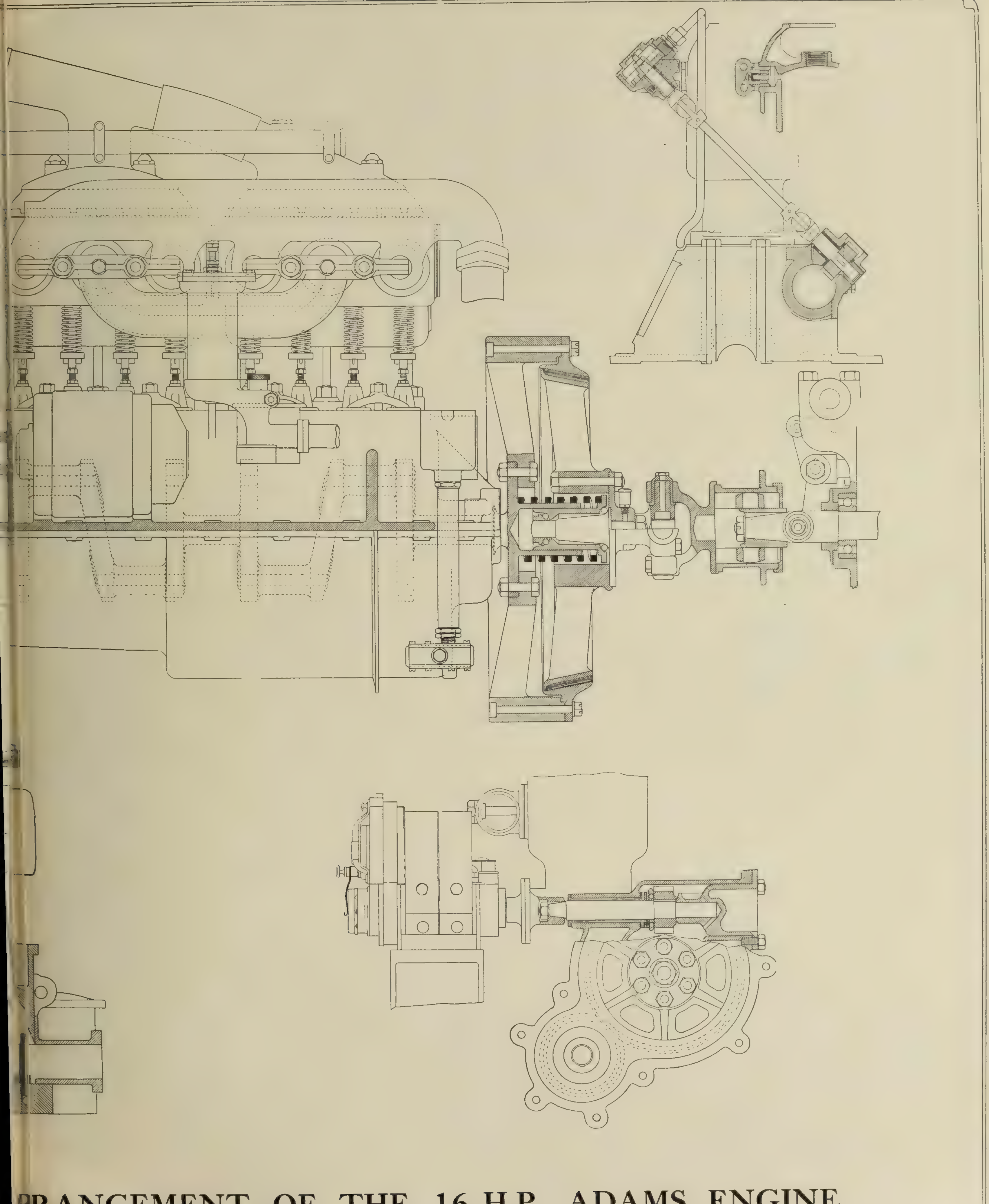


Fig. 11

Fig. I. GENERAL AR



RRANGEMENT OF THE 16 H.P. ADAMS ENGINE.

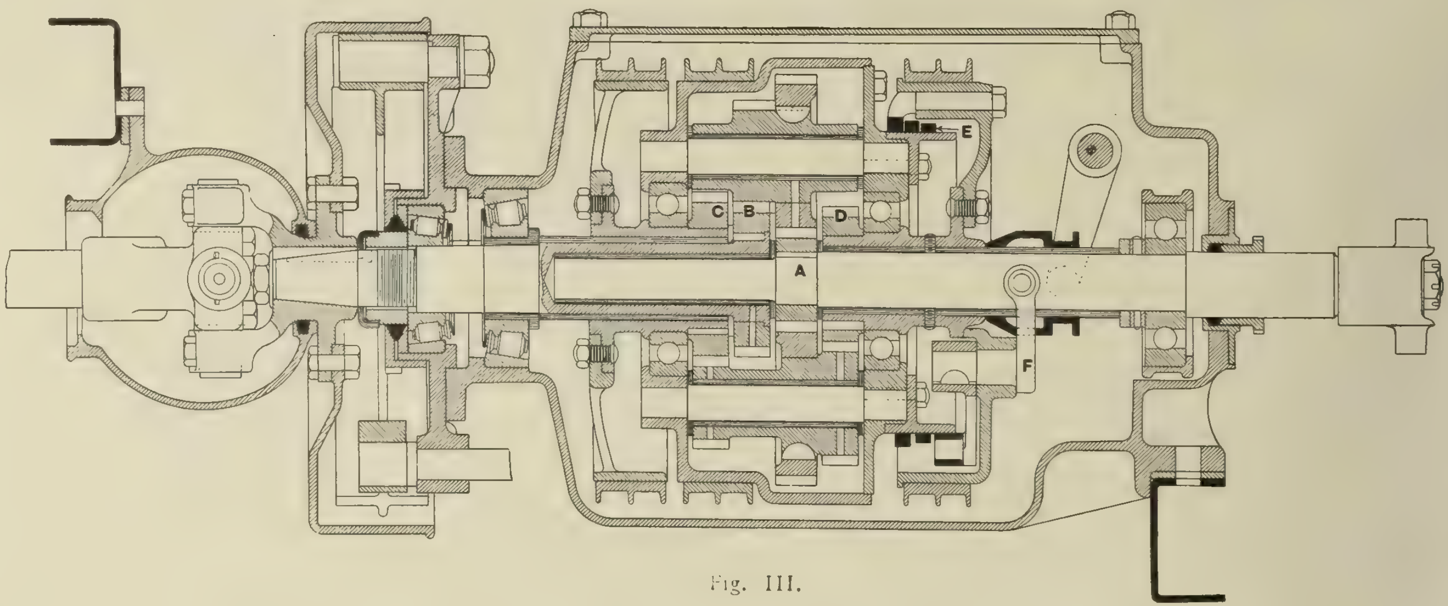


Fig. III.

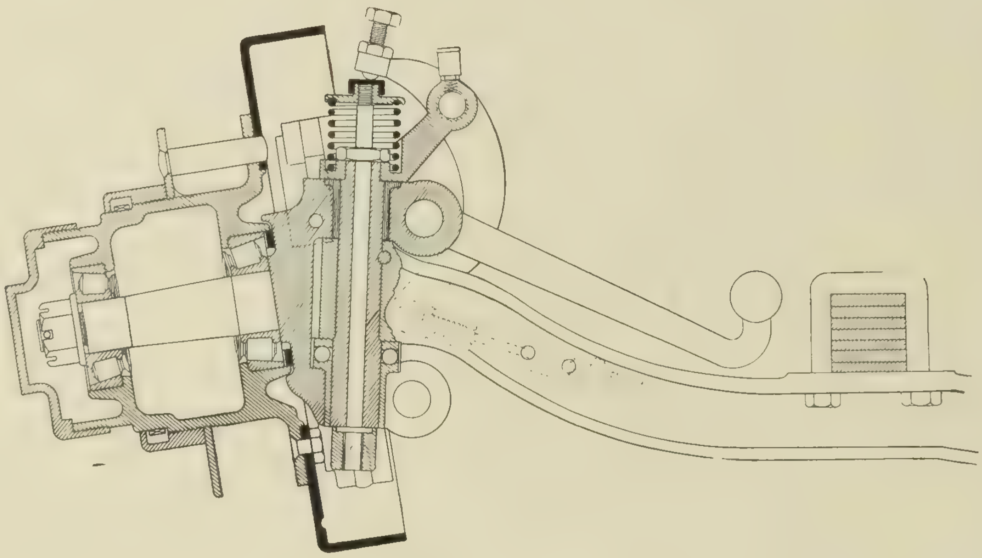


Fig. IX.

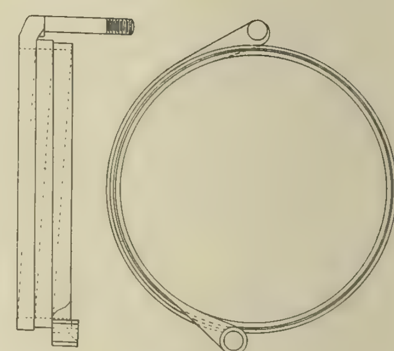


Fig. IV.

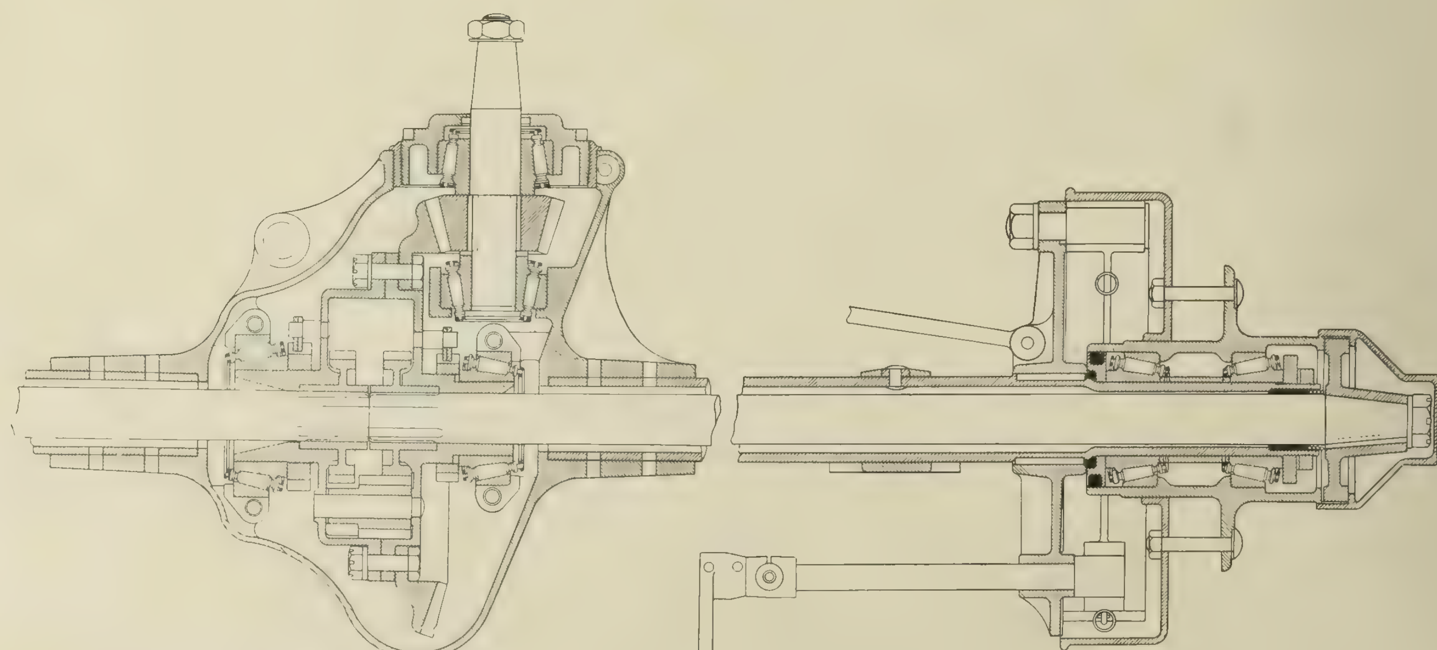


Fig. VII.

16 H.P. ADAMS GEAR AND AXLE ARRANGEMENTS.

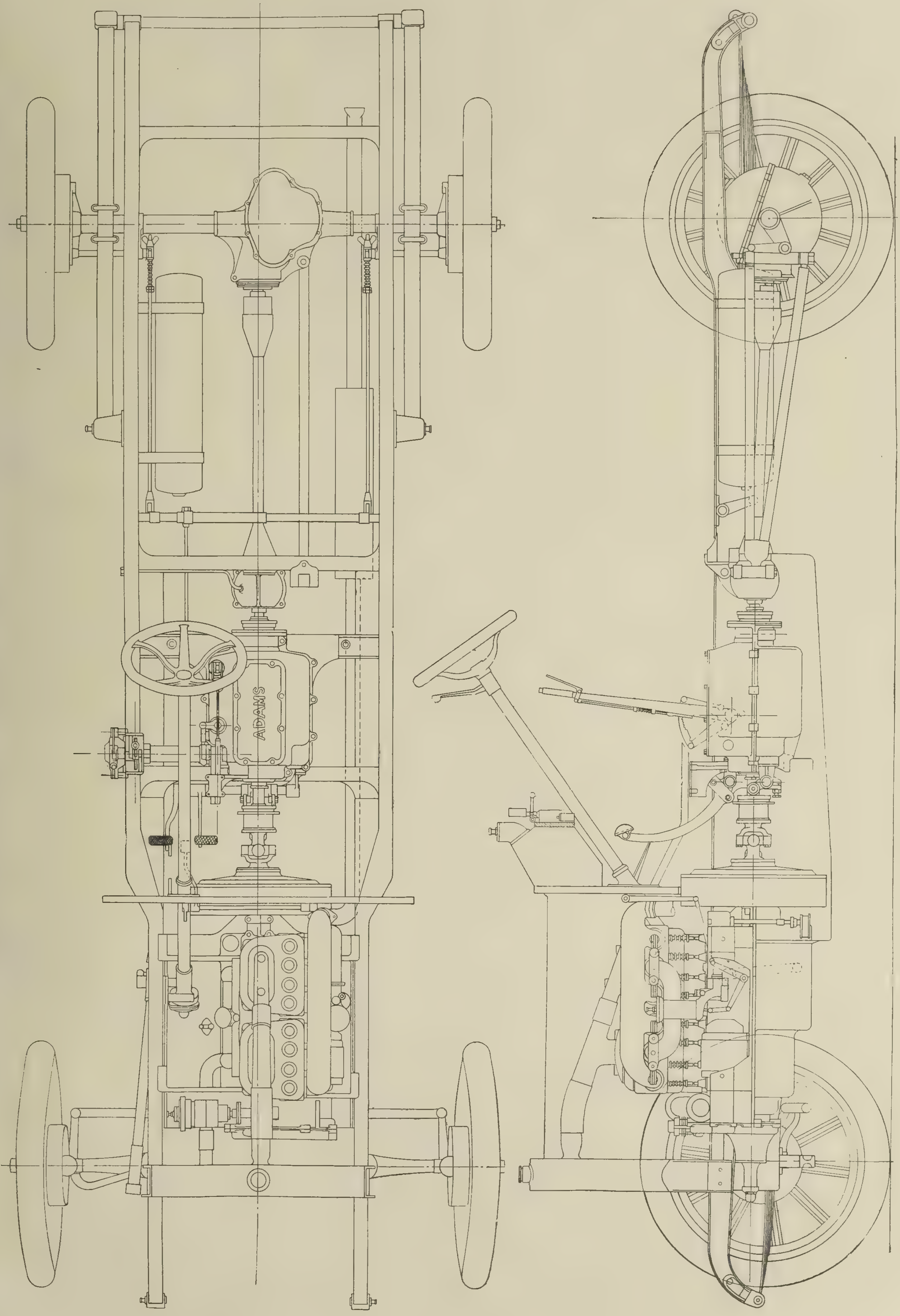


Fig. XI.

THE 16 H.P. ADAMS CHASSIS.

what complicated, but it is probably not more costly than standard practice, while being very strong and rigid. From the clutch a drive passes to the gearbox through a double universal joint, of which the details are explained sufficiently clearly by the illustration, it being only necessary to add that the De Dion type joint has a malleable cast iron pot provided with case-hardened steel working surfaces held in place by pegs, and a steel back plate. The clutch pedal operates through a link and ball-crank lever, the striking fork being provided with two small D.W.F. ball bearings which are brought in contact with the back plate of the rear universal joint.

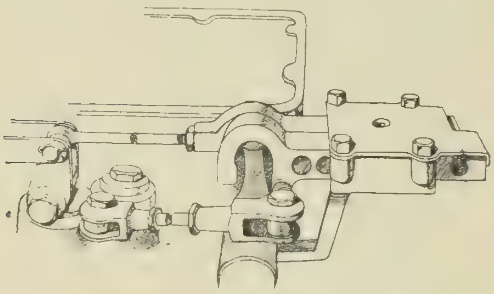


Fig. V.

Three speeds and reverse are given by the epicyclic gear, the reductions in the gearbox alone being as follows :—First speed, 4.2 to 1, second speed 2 to 1, third speed, direct and reverse 4.2 to 1. To follow the action, it must be remembered that only one sun wheel is in permanent connection with the clutch shaft, and this is marked A in Fig. III. A second sun wheel B is keyed to the tail shaft, while the two wheels C and D, are fixed to the first speed drum and the reverse drum respectively. There are three planet wheel shafts, but the planet pinion sets are not all precisely alike, as can be seen in the figure. This means that they are not all in operation on each of the reduced speeds and by this method of construction the cost of production can be lessened considerably.

For the first speed, when C is held and all other parts are free, the planets are forced to roll round on C thereby driving B at the reduced rate already mentioned. For the second speed the planet box is held and the drive then passes from A to B direct through a pair of planet wheels. When D is held, since D is smaller than C, a reverse motion is imparted to B by the revolution of the planets. For the direct drive all the brakes are freed and the pinion-carrying box is clutched to D through the scroll marked E and shown separately in Fig. IV. The operation of the scroll is controlled by a conical collar which lifts the lever F so turning the shaft to which it is attached, and putting the scroll in tension through the short lever inside the drum. On withdrawing the collar the natural spring of the scroll is sufficient to cause instant release. Operation of the brake bands for the lower speed is performed by toggles which, together with the top speed control, may be connected up in two ways, the most usual being to employ a side lever working in an ordinary gate in connection with the mechanism shewn in Fig. V. which illustrates the selecting and the striking mechanism. The brake bands are hinged on a shaft mounted in lugs inside the aluminium case, and the free ends are connected up to the cam mechanism shown in Fig. V. Thus these two parts are drawn together equally and there is no tendency to bring one shoe into operation before the other. Adjustment for tightness merely controls the total opening between the free ends of the shoes and does not in any way affect their concentricity. The alternative control is somewhat similar, but in place of the hand lever and gate a pedal is provided, itself working in a gate with four slots. To engage a gear the pedal may be moved sideways by the foot and pressed into any one of the slots, but before another gear can be engaged it is necessary to return the gear pedal to the free engine position by depressing the clutch pedal. The change is therefore difficult to perform so quickly as with hand lever, and it is not easy to see that the foot control possesses any important advantages.

When in operation this transmission is slightly less noisy than the average sliding change speed gear particularly on the second speed. It is practically noiseless on the top speed, because none of the gears are in relative motion, while in a sliding gearbox the layshaft drive at least is always in mesh. On the other hand, when in the free position many parts of the gear are revolving and the gear is then inferior to the standard type on the score of noise To overcome this difficulty the clutch pedal is fitted with a catch shewn in Fig. VI., and this is usually employed when running the engine with the car standing, while it can also be used for running down hill with the engine shut off. For changing speed it is not necessary to use the clutch, but those who are accustomed to a sliding gear will find it more convenient to use it for starting if not for changes, and some shock is avoided if the clutch is freed slightly during the engagement of the top speed because the scroll clutch takes hold very quickly and cannot be slipped to more than the very smallest extent.

From a manufacturing point of view, most of the parts of the gear are simple, and although they are fairly numerous their number is not so great as is the case with some other epicyclic transmissions. Assembling is, of course, rather a slow job, and requires to be carried out with considerable care. It will be noticed that the pinion carrier runs on large ball bearings and that the main shaft has a ball bearing in front with a Timken combination at the rear end, the latter taking all end thrust from the top speed clutch actuating gear or from the propellor shaft. Before leaving the subject of transmission it ought to be remarked that a sliding gear is supplied with Adams chassis if so desired, but the gear we have just described is recommended by the manufacturers and the sliding box which has been fitted up to the present time exhibits no peculiarities which make its description necessary.

As has already been explained, the engine is supported directly from the side members of the frame and the gearbox rests upon two dropped cross members, lining up being performed by placing fibre packing strips beneath the gearbox feet. The universal joints on the propellor shaft are precisely similar to those between the clutch and gearbox, only in this case the forward joint is enclosed in an aluminium box separate from the gearbox casting and hung from another cross member which comes behind the gearbox and is not dropped. This construction can be seen in Fig. III., which also shows the construction of the foot brake. Experience has shown that this method of joint protection is satisfactory and it will be observed that quite a large quantity of oil can be carried without much danger of leakage, although a piece of leather is used to connect the casting and the shaft to prevent the ingress of dirt. Both the foot brake and the hand brakes are operated by toggles instead of being cam-expanded, which is commendable because it reduces the risk of the brake sticking that is always present when a spring alone is relied upon for the return. The actuating shaft for the foot brake is carried on lugs solid with the gear box casing, but the plate which carries the shoes is an entirely separate piece, so no unnecessary stress is applied to the casing

For the rear axle (shown in Fig. VII.) steel sleeves, on which the road wheels are mounted, and a malleable cast iron centre casing are used. The method of construction is shown fairly completely in the illustration, where it will be seen that the separate spring pad is slipped on and located by means of a rivet, the pad itself being slotted to allow rotational movement. Keys are used to attach the brake bracket, and the tube is secured to the casing by rivets, the two parts being a press fit with each other. The most interesting feature of the axle is the manner in which adjustment is provided for the depth of engagement of the bevels. It will be noticed that each of the Timken bearings on the differential box is backed by an adjustable threaded ring : the outer portions of the bearings are, of course, held in the casing and in assembling the two rings are screwed up until there is no end play on the box. When this is done the position

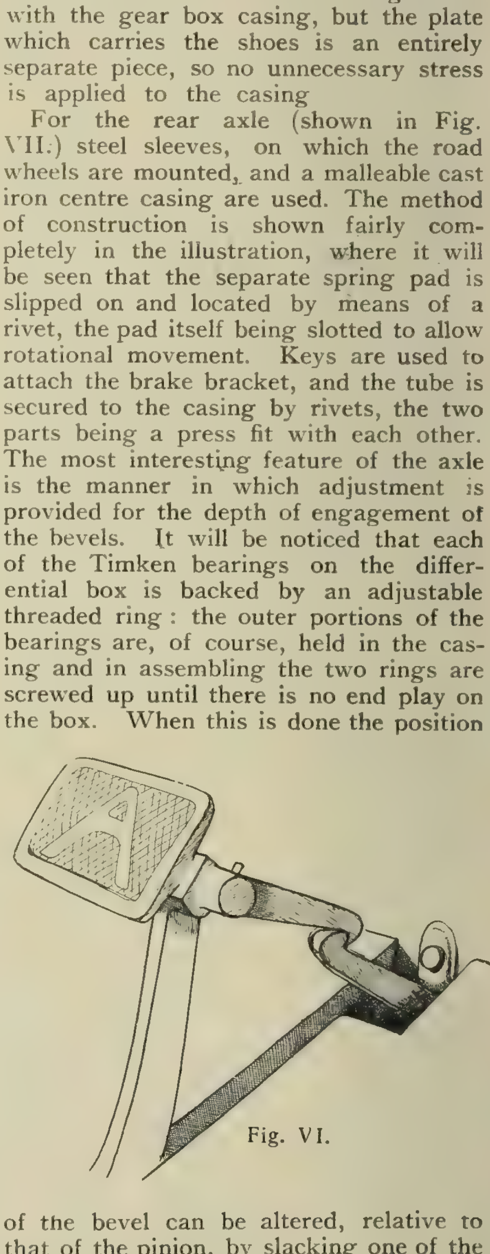


Fig. VI.

of the bevel can be altered, relative to that of the pinion, by slacking one of the rings while tightening the other to an equivalent extent, and this adjustment is claimed to be a very useful one when trouble arises with the drive through noise. Similarly the pinion is adjustable axially by means of the threaded cap carrying the outer bearing, although if this adjustment is used it is necessary to vary the thickness of the washer seen between the rear end of the pinion and the inside bearing. Owing to the stiffness of the case the differential and the

driving wheels are supported with unusual rigidity, which should make them very durable, while in actual practice these axles are certainly not noisy. Needless to say, a stub type of tooth is used for the spur differential, and the proportions of the axle gear are varied according to the purpose for which the chassis is required.

A small point which seldom seems to receive attention is shown by the arrangement of the torque rod which is of the ordinary triangulated pattern, but so set that the long bolt through the axle case normally is vertical. This gives freedom to the axle allowing it to swing a little, and is claimed to prevent undue stressing of the torque rod and its connections by road shocks.

Also, in the plan of the chassis, Fig. XI., the method of attaching the covet half of the centre of the axle can be observed, and the very large opening pro-

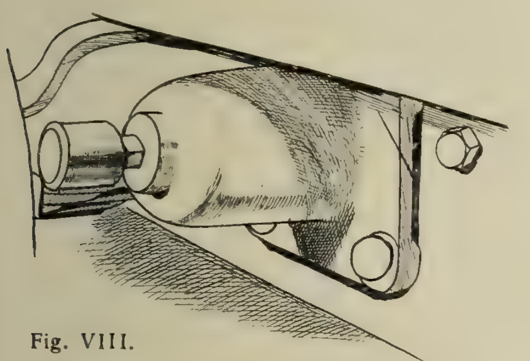


Fig. VIII.

vided by removal of this cover should be extremely convenient for cleaning purposes as well as being useful to the manufacturers for bevel adjustment. The driving shafts can be withdrawn from the hub if it is desired to remove the whole differential case.

Concerning the frame and springs, only quite small departures from usual practice are to be noticed, but as the method of attaching the spring hangers are more than usually neat this is shown in the sketch Fig. VIII., where it may be noticed that the hanger has four small feet which are drawn against the frame side by tightening up the inner plate: this gives a better bearing on the comparatively irregular surface of the frame side, while also reducing the cost of machining the piece. So far, when referring to brakes, we have mentioned only those on the rear axle and propellor shaft, but for the latter it is proposed to substitute front wheel brakes if required, and the arrangement of these is shown in Fig. IX. This figure in any case holds good so far as the front axle and hub arrangements are concerned, the brake being but little more than an attachment: in fact it could be fitted to any chassis of the type now being described. The front wheel and pivot are raked at fairly wide angles with the idea that their centre lines shall coincide at the point where they cut the road surface, and this we believe to be a very desirable feature with front wheel brakes in the absence of a truly central pivoted front hub. The whole of the operation of the brake can be followed in the drawing, with the exception perhaps of the connection with the pedal which is performed by means of ordinary Bowden tubes and cables, the wire being linked up to a compensating mechanism at its point of attachment to the pedal.

At the present time the Adams Company are using a number of Sankey detachable pressed steel wheels, and the front hub shown in Fig. IX. is arranged for a wheel of this type, although the hub shown on the back axle is for the ordinary fixed artillery wheel. It is of course only necessary to change the back wheel hub shell to make the pressed steel wheels interchangeable on both the axles.

The steering gear is noticeable on account of the extremely neat way in which

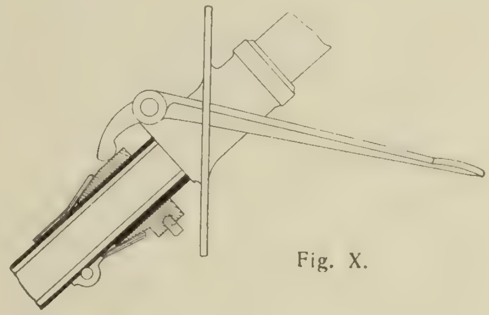


Fig. X.

the control is arranged as shown in Fig. X. A Renault pattern of throttle lever is used, rotating the outside tube, and this tube ends in a cam. As the tube is turned the cam lifts the short end of the accelerating pedal, turning the shaft to which the latter is pinned and actuating the throttle through a lever also pinned to the same shaft. Thus the pedal controls only beyond the point set by the hand lever. To lock the setting in any position, without the use of a ratchet, the cam ends in a conical piece which fits inside a brass cup clipped to the dashboard bracket, and there is a spring above the cam to keep it forced into the cup. This provides just sufficient friction to hold the lever while giving perfectly smooth movement, and it is also by no means an expensive device. A separate Bowden ignition control may be fitted on the steering column. On the road the car steers well, and this is probably partly traceable to the inclined pivots. Doubtless an improvement would be noticed if ball bearings were used in the worm box, and it seems curious that they should have been omitted on a chassis which, taken as a whole, is extremely carefully constructed and very well thought out.

A DOUBLE ROTARY ENGINE.

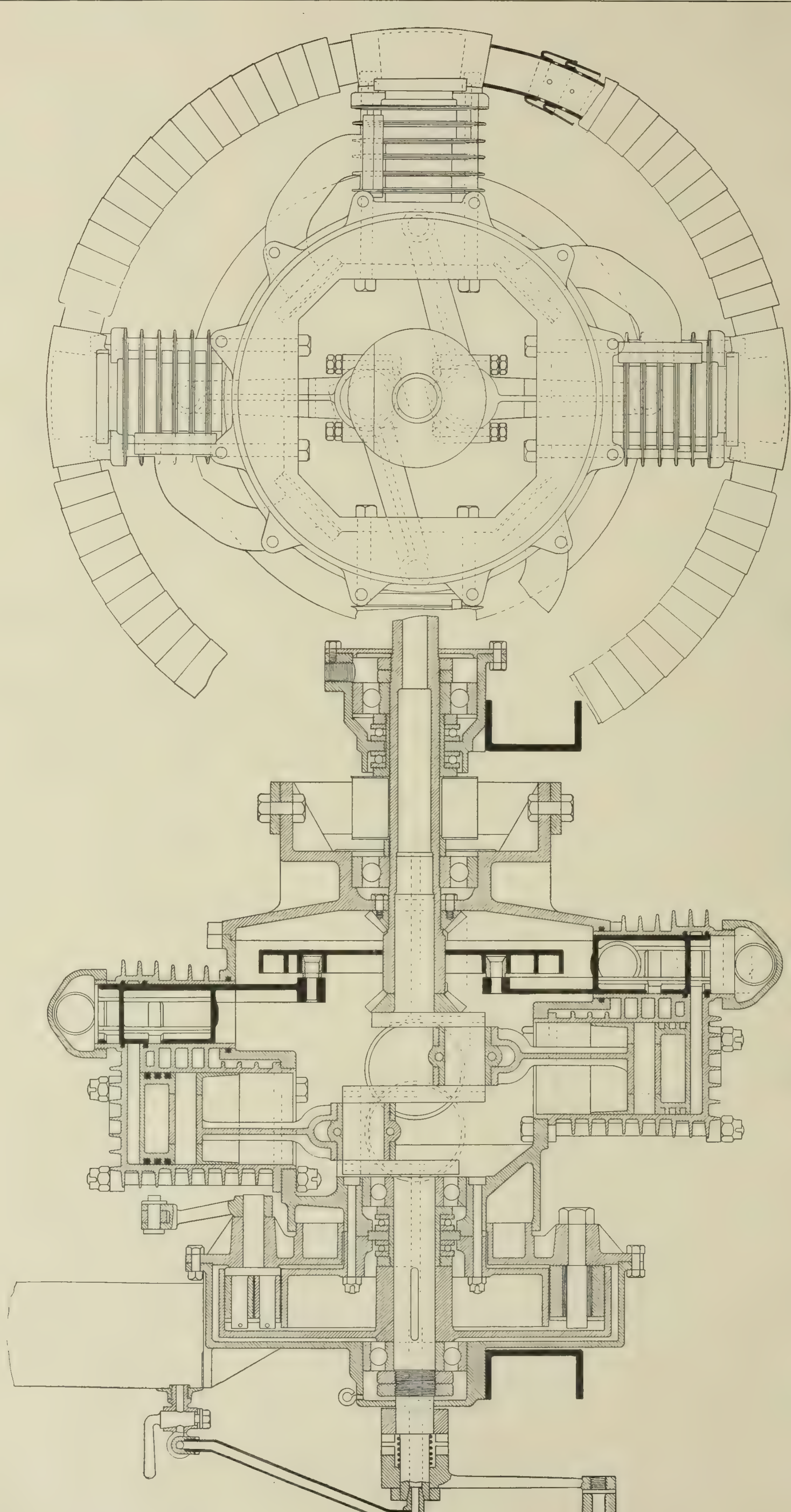
An interesting and original design for either car or aeronautical work.

On the following page are given some illustrations prepared from the working drawings of an engine which has recently been designed by Mr. J. D. Roots. This particular design has grown from a somewhat more simple idea intended solely for aeronautical use, and the main feature of the construction is that either the cylinders or the crankshaft may rotate; or they may both rotate at the same time in opposite directions. To obtain this end the shaft is mounted in bearings on a frame or bed plate, and the cylinders on ball bearings, outside the crankshaft. Supposing that the crankshaft is held and the engine started by revolving the cylinders (precisely after the fashion of the Gnome engine), but that the crankshaft be released as soon as the engine has commenced to fire, then, since action and re-action are equal and opposite, assuming the frictional resistance to be the same for each part, obviously the cylinders and shaft would continue to move relatively to each other at the same speed as before, but the cylinders would only make half the number of revolutions per minute which they did when the crankshaft was held stationary, while the crank would make an equal number.

Thus in an aeroplane it would be possible to drive two propellers (of course of opposite pitches) at the same speed, and, when the propellers were revolved at say 1,000 r. p. m. the relative motion of the cylinders and pistons would be equivalent to 2,000 r. p. m.

In the engine which we illustrate an endeavour has been made to utilise the same principle to give a variable speed engine, or rather a variable speed shaft coupled directly to an engine without any intermediate change gear mechanism. This is accomplished by means of the differential gear seen in the side sectional view. It will be noticed that one bevel pinion is fixed to the cylinder case and another to the crankshaft, these being connected by means of three mitre wheels so forming an ordinary differential gear. If therefore the cylinders and crank rotate with equal velocities in opposite directions the intermediate pinions will revolve on their axes, but the cage which holds them will have no tendency itself to revolve in either direction. As soon however as either crank or cylinders are retarded, motion of the pinion carrier must begin to take place, for it can only remain stationary so long as the cylinder and crank wheels revolve at equal speeds.

At the other end of the engine, in the same view, it will be seen that there are two drums connected to cylinders and crank respectively, and between them a special form of brake ring is situated which can be either contracted or expanded. Supposing therefore, that the engine is running, and that the brake is applied gently to the crankshaft. As the relative motion between the cylinders and crank will be the same as before, if we assume them both to start revolving at 1,000 r. p. m., then if the crank is slowed down to 800 r. p. m., the cylinders will revolve at 1,200 r. p. m., and the cage carrying the differential pinions will revolve at a speed of 200 r. p. m. or half the difference between the crank and cylinder speeds. The pinion carrier is mounted on what Mr. Roots prefers to call the ultimate shaft, that is to say the drive is taken from this shaft, and the idea is that the brake should be applied to the crankshaft, first very gently and then gradually more and more fiercely as the car gathers speed, so raising the gear ratio up to the point where the crankshaft is locked, and the ultimate shaft revolves



THE ROOTS ROTARY ENGINE.

In this engine either the cylinders or the crankshaft or both can rotate, while variable speed in either direction is obtained from an ultimate differential driven shaft by means of braking the movement of either of the revolving members.

at half the speed of the cylinders. For reversing, the brake ring is operated in the opposite direction, that is to say the cylinders are slowed down and the crankshaft allowed to revolve freely. In this way the full range of speed, from zero

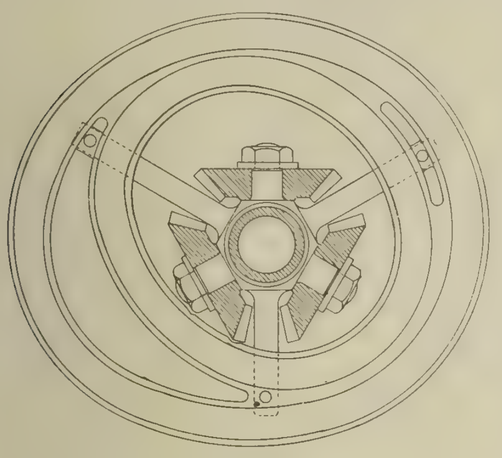


Fig. II.

to the maximum of which the engine is capable, is obtainable in either direction.

Of course a considerable amount of power would necessarily be lost through the friction retarding device, but it is claimed that this amount would not greatly exceed that absorbed in an ordinary transmission gear. The claims made for the engine are considerable, and it will be sufficient to state the most notable amongst them, which are that its use would enable a change speed gearbox to be done away with as well as the usual water cooling arrangements. Further, the engine is said to be intrinsically light, as indeed would be expected, and therefore the total weight of a chassis of any given horse power fitted with this engine might reasonably be extremely low. However, apart from the question of advantages and disadvantages, the details of the engine are worth considering. First of all in respect to the valve gear there is a single piston valve to each cylinder, but instead of being driven by an eccentric each piston is actuated through a roller running in a cam-shaped groove in a piece which is attached to the inner end of the ultimate shaft. There are two cam grooves cut concentrically in the same piece, and each groove operates two of the opposed pistons. See Fig. II.) The exhaust gases are allowed to escape by passage over the top of the piston, whence they pass into the exhaust pipe and are disposed of in a manner which will be described further on. The intake passes through the inside of the valve piston (thereby cooling the head) and the difficulty of supplying mixture to the valves has been overcome by the arrangement seen between the brake gear and the cylinders in the side sectional view. The stationary plate which carries the brake fulcrums is shown in Fig. III. which is an end view, and it is proposed to use two carburettors both feeding to the annular channel, which can be seen partly in the stationary plate and partly in the revolving crank case, and from the latter part pipes lead to each of the cylinder valve ports. No special arrangement has been designed to prevent air leakage at the junction in this mixing chamber, it being surmised that any leakage which does take place will be constant and can therefore be compensated for by carburettor adjustment.

The construction of the exhaust pipe, which is also a silencer, is interesting quite apart from its application to this particular and peculiar engine. It is shown in section in Fig. I., and is composed of an inner steel pipe pierced with a number of small holes surrounded by cup-shaped stampings, which are slightly conical in form and arranged as shown.

Fig. III.

Exhaust gas therefore passes through the holes in the inside pipe and reaches the small annular chambers formed between any two cups. Thence it escapes to the outer air, a very small amount of clearance being allowed between each pair of the cups for this purpose. With a revolving cylinder engine, and with the cups arranged facing the proper way, there would, of course, be continual suction on the little annular spaces, so assisting the exit of the exhaust while, as the total length of outlet corresponds to the

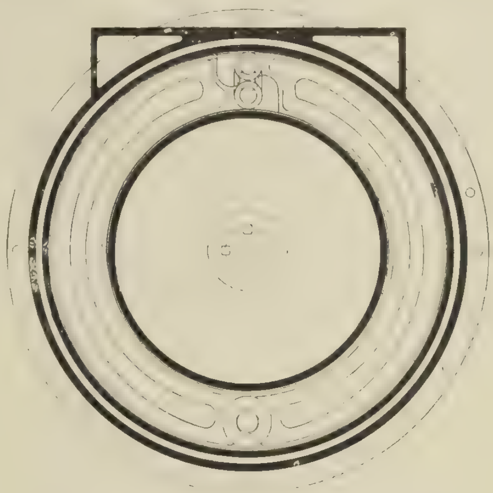


Fig. IV.

sum of the peripheries of all the cups and the individual clearance is in each case extremely small, the exhaust should be silenced efficiently.

The next point calling for a more detailed description is the brake gear shown in Fig. IV., which is included for the purpose of illustrating the cast iron ring which acts in both directions. This ring is solid, its natural spring being relied upon to retain it in its normal position, the lower fulcrum pin taking the thrust while the upper piece is used for either expansion or contraction according to the way it is applied. In the two ends are holes for pins, and a pair of pins to correspond are fitted in the cam plate attached to a shaft which in turn is linked up to the hand control. Movement in one direction therefore causes contraction, movement in the opposite direction expansion, while on release the ring clears the drums automatically.

Speaking broadly this engine is too peculiar for it to be possible to offer really useful criticisms. It would be easy to enlarge upon the small number of minor mechanical defects, but it would serve no useful purpose to do so and therefore further consideration will be left until such time as the 15 h.p. engine (which is now in course of construction from the designs shown in the illustrations) is ready for testing, which we understand it should be some time during the summer.

WHEEL AND ROAD.

A paper read before the Incorporated Institution of Automobile Engineers.

By A. Mallock, F.R.S.

If we examine the history of wheeled vehicles from the earliest times to the present day, we can recognize three distinct periods of progress, each depending on the way in which the pressure of the useful load is transmitted to the ground, and each also marked by a large increase in speed. The first period extends from remote antiquity to the seventeenth century of our era, and throughout this time, although many forms both of two-wheeled and four-wheeled vehicles were produced, they all had one feature in common, namely, that the load was carried directly on the axle without the intervention of any kind of spring except that due to the natural want of rigidity of the structure. The second period dates from early in the seventeenth century, when springs of various kinds were first placed between the body of the vehicle and the axles of the wheels. The third period began some time between 1870 and 1880, when springs (in the shape of solid elastic tyres) were placed between the rim of the wheel and the ground. In this paper I propose to examine the distinctive features of the vehicles of each of these periods in so far as they affect the forces exerted on the load carried, and on the surface of the ground.

If the wheels are truly circular and the surface of the ground is smooth (by "smooth" in this connection it is meant that, at the speed of the vehicle, the vertical accelerations are small compared with the statical forces), springs of any sort are of no advantage, for there is nothing in the motion to cause them to alter their shape, and they might therefore be replaced by rigid structures. Such smoothness as this however, is never realized, and even on railways where it might be expected to be most nearly approached, not only is the vertical acceleration of the wheel comparable to gravity, but it often exceeds it, so that the wheel at times actually rises off the rail. The real problem therefore, is to determine the motion and forces resulting when a load carried by a wheel travels over a rough or uneven surface. Before considering this dynamical problem it will be interesting to give a short sketch of the history of wheeled vehicles, in particular with reference to the speed attained.

The use of wheels is of unknown antiquity. War chariots are represented in Egyptian paintings, and chariot races were common in Greek and Roman times. I cannot find that any information is to be had as to the speed attained in such races, but since, from classical allusions to such events, the ground of the race-course appears to have been sandy and yielding, it may be taken that it was limited by the power the horses

could develop on such ground, and probably did not exceed that of an old English mail coach on a good road, namely 12 or 14 miles an hour. Although no records are available as to these racing speeds, there is some information as to the rate of progress made by Roman mails and despatches, and by various important people in journeys where time was an object. The speeds, as might be expected, were very low, averaging under 3 miles an hour, on the assumption that when the journey took days, half of the time was given to travelling. I may refer anyone who is interested in the matter to Ludwig Friedländer's "Sittengeschichte Roms in der Zeit von August bis zum Ausgang der Antonine," where references are given to many classical authors. From those times down to the seventeenth century the speed of transit hardly increased. About then springs were introduced for private coaches (Louis XIV., of France, had a spring-borne carriage in 1643), but ordinary traffic was still carried in springless carts or wagons.

In the eighteenth century the wagons from Edinburgh to London, a distance of 400 miles, took a week on each journey, and counting twelve hours to a travelling day, this is equivalent to less than five miles an hour. Stage coaches with springs were introduced in 1750, and at a rather later period great improvements were made in the roads themselves.

The badness of the old roads made comparatively little difference to the speed of the springless vehicles, since with them anything but the accustomed slow speeds produced an intolerable jolting, even on what would then have been called a good surface, but with the advent of springs the advantage of a hard and fairly even road became apparent. With the improved roads the mail coaches used to average something like ten miles an hour, with a maximum of thirteen or fourteen perhaps, nearly three times the speed at which the springless vehicles could be driven. From that time onwards private carriages and other vehicles took an immense variety of forms, some of them merely the result of fashion and some introducing real improvements in structure, but in all cases depending on the springs for the comfort of the occupants and the safety of the vehicles when driven fast.

Between 1870 and 1880 solid rubber tyres were first introduced, and this date is the opening of the third period above referred to. The first patent I can find relating to india-rubber tyres belongs to the year 1855, but this did not go beyond a provisional protection. As has often occurred in other matters, the object aimed at by the inventor is one of the least of the merits of the invention. The idea in this case was, I believe, to reduce the noise made by the ordinary iron tyres on town pavements. The solid tyres were first applied to hansom cabs in London. Their use soon spread to other vehicles, and is still being widely extended.

In 1888 the first patent was taken out for pneumatic tyres, *which for many purposes (especially as regards the comfort of occupants of the carriage) were as superior to the solid rubber tyres as the latter were to iron tyres. In fact, if it were not for questions of expense and liability to damage, pneumatic tyres would everywhere supersede the solid ones. The reason for this will be

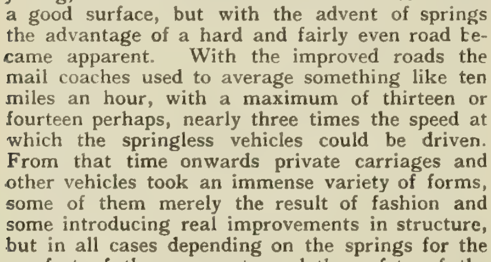


Fig. I.

apparent after the dynamical problem before stated has been considered, but it may be noticed that, as in the case of the solid tyres, the inventor was quite unaware of the chief merit of his invention. With the advent of pneumatic tyres (and light engines for use with them) speeds were again largely increased.

The problems considered in this paper are three in number. It is supposed that a load borne on a wheel travels over uneven ground at a uniform horizontal speed. Between the load and the wheel and between the wheel and the ground it is supposed that springs of known stiffness intervene. The weights of the load and wheel are supposed to be known, and in comparison with them the weights of the springs are supposed to be negligible. These suppositions are fairly close to the actual conditions, and by assigning different values to the ratio of the weights and to the strength of the springs they can be made to represent the case of any vehicle, whether springless or fitted with ordinary springs, and with hard or springy tyres. For instance, if both springs are infinitely stiff, the case is that of a springless vehicle with hard tyres. If, on the other hand, only the spring between the tyre and the ground is infinitely stiff, the case represented is that of a carriage with ordinary springs and hard tyres, and so on.

The three points which it is proposed to examine are: the forces experienced by, and the motions of (a) the tyre, (b) the axis of the wheel, (c) the load while the vehicle travels over uneven ground (of specified quality) at various velocities.

As long as the conditions as to velocity and ground surface are such that the wheel never leaves the ground, the problem is comparatively simple. When, however, the speed is so high or the ground so rough that jumping occurs, the exact treatment of the motions and forces would be very complicated, and of little practical importance, but it is easy to examine the "worst cases," and find the average effects. (See Appendix). In many forms of roadway the surface is such that, although the wheel never leaves the ground, the line on which it touches it is not continuous, but forms a succession of discontinuous lines with intervening gaps (for example, on stone setts). These cases also are considered. It

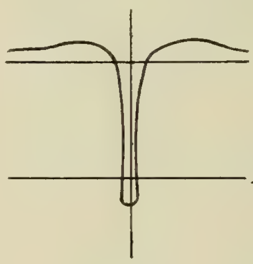


Fig. II.

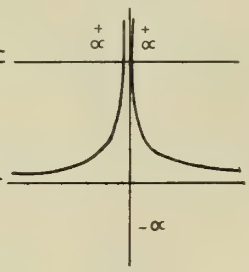


Fig. III.

would be out of place in a paper of this kind to introduce mathematical symbols. In the Appendix, however, a sketch of the process by which the results were obtained is given, but here the results only will be stated.

In Fig. I the curve A represents the surface of the ground, which is supposed to consist of regular series of low waves of the ordinary harmonic type. (Wave length λ, amplitude The regularity thus supposed in the nature of the uneven surface does not really detract from the generality of the results stated, since it is well known that any single-valued arbitrary curve can be represented as the sum of various simple harmonic curves, and the total effect of the series as the sum of the effects of each component. The curves B_1, B_2, B_3 shew the path of the axis of a wheel of radius R_1, R_2, R_3 respectively rolling on the ground surface, and if there are no springs, or infinitely stiff ones, the curves B_1, B_2, B_3 are also the path of the load carried. The radii R_1, R_2 and R_3 are chosen so that their relations to the wave length and amplitude of the A curve are such that R_1 is less than the least radius of curvature of the ground surface, while R_2 is equal to, and R_3 greater than, this radius of curvature. R_2 is a critical value. It will be noticed that while for any wheel whose radius is less than R_2, the B curves differ from the simple harmonic type by having flatter tops and more localised depressions, the depression for the critical value becomes a cusp, so that the concave part of the curve disappears, and that where the radius exceeds the critical value there is a loop in the curve B_3, at the depression. The loop itself corresponds to no real part of the contact of the wheel with the ground, for when the centre of the wheel is at the top of the loop the tyre is in contact with the ground at two places, P and P^1, and the ground between these points is not touched at all, so that the actual line of contact is discontinuous.

From this it will be seen that for a load borne by the wheel at a constant horizontal speed, the dynamical force acting on the load owing to the curvature of the path is directed alternately upwards and downwards where the radius of the wheel has less than the critical value, and that the force acting upwards is less in amount than that acting downwards, but it acts for a longer time, the time-integral, however, of upward and downward force during the complete cycle being zero.

When the radius is equal to or greater than the critical value, the upward force acts for the whole time, except at the cusp or the commencement of the loop. At this point the downward force becomes infinite (on the assumption of complete rigidity in the structure), that is, the downward force is of the nature of a blow which only lasts for an instant, but which is capable of generating the same momentum in that time as the upward force generates while the axis passes over one complete wave length. Diagrams representing the variation of force on the ground corresponding to the three radii of Fig I are given in Figs. II. III., and IV. The forces are expressed in terms of the total load for a velocity of one foot per second.

It should be noticed also that the slope of the path in the neighbourhood of the concave part becomes steeper as the wheel approaches the critical dimension, and is vertical at the cusp when the critical dimension is reached. In order therefore, that the load should follow the path geometrically indicated, it would require, in this part of the curve, to be acted on by an infinite downward accelerating force, whereas in reality the only downward acceleration is due to gravity. In real cases therefore, the wheel will leave the ground at some small distance (depending on the horizontal velocity (see Appendix) before the axle reaches the point vertically over the lowest point in the hollows.

Up to the present we have referred only to the dynamical variations of the vertical forces acting on the load and ground, but horizontal variations occur also, the expressions for which are given in the Appendix. We may now examine the manner in which these forces are altered when a spring is interposed between the load and the wheel. It will be supposed that the spring is of such stiffness that the period of the load oscillating vertically on it is T, and it may be remarked that this period can be arrived at in a very simple way, for if the weight of the load is large compared to that of the spring (as it is in practice), the distance through which the statical application of the load compresses the spring is the length of the simple pendulum which has the period T. In the first place we will examine the forces acting between the wheel and the ground, and then those acting between the wheel and the load, the two being no longer identical. The forces between the wheel and ground can be found from the case of a rigidly-supported load by substituting the weight of the wheel for the load and remembering that the downward accelerating force acting on the wheel is not gravity only, but gravity plus the action of the spring above it. To find the effect of the spring on the wheel it may be observed that the natural period of the wheel on the spring is the period of a pendulum whose length is the compression produced in the spring by a weight equal to the weight of the wheel, and since the weight of the wheel is generally much less than that of the load it carries, the period in question will be considerably less than the period of the load on the spring. As regards the vertical motion of a wheel which is pressed on the ground not only by its own weight but also by the loaded spirng, this is only affected in so far that the path which can be followed by it without jumping may be more sharply curved than when the load is carried directly on the axle. So far therefore, as the variation of pressure on the ground is concerned, the chief difference between the spring-borne and the rigidly-carried load is, that in the former case the variation of pressure is diminished in the ratio of the weight of the wheel to the weight of the wheel plus the load carried. (In practice something like 1/10.)

If we turn to the load itself, and examine the variations in the vertical force acting on it, we

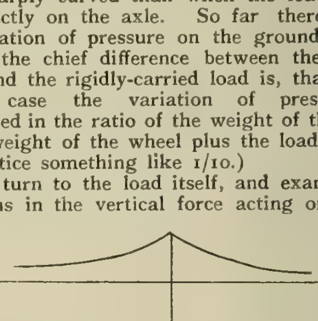


Fig. IV.

see that this is measured by the variation of the compression of the spring. The problem involved is the same as that involved in determining the motion of a pendulum whose point of support is shifted laterally in some prescribed way. The point of support here corresponds to the axle of the wheel, and the bob of the pendulum to the load carried. The calculation will be found in the Appendix, but the general result is that unless the speed is such that the time taken in passing over a complete wave length, or a sub-multiple of it, is equal or fairly close to the period of the load on the spring, the vertical motion given to the

*The inaccuracy of this statement was pointed out in discussion.

load will be small compared to the vertical motion of the axis, that is, that the load will be subject to only small vertical accelerations. In practice the springs of carriages are generally made so that the compression due to the statical load lies between one and three inches. The period of the load on the spring, therefore, lies between $2\pi\sqrt{1/12g}$ and $2\pi\sqrt{3/12g}$, that is, between 0·32 and 0.55 seconds.

Now, take the case of a spring-cart travelling at 10 miles an hour or 14.6 feet per second. In 0.32 seconds it will have moved 4.6 feet and in 0.55 seconds, 8 feet, and if the pitch of the inequalities of the road has these lengths, the body of the cart would have its vertical motion much exaggerated. In fact, the occurrence of a few hummocks, regularly spaced at intervals which agree with the natural period, would make it difficult for the driver to keep his seat. The road inequalities, however, have usually a much shorter pitch than this, and synchronism between the two fundamental periods rarely leads to trouble in the case of ordinary carriages.

I have computed the forced vibration which would be caused in the body of the spring cart referred to (on the assumption that the wheels are five feet in diameter and the speed ten miles an hour) in passing over a waved surface with the ridges spaced at intervals of two feet, and in the result it appears that variation of level of the load carried is less than one-tenth of the height of the ridges, but that, although the up and down motion of the load is very much reduced by the introduction of the springs, there are very quick changes of accelerative force at places corresponding to the cusps or discontinuous change of direction in the path, though the actual amount of movement produced is small. The variations of the horizontal force (still assuming the horizontal velocity constant) are reduced by the introduction of the springs in the same proportion as are the vertical variations.

We may now consider the case of elastic tyres. The introduction of a spring between the wheel and the ground of course affects the path of the axle, and in general will make that path less curved than it would otherwise have been. We have said that the natural period of the load on the spring is the same as the period of a pendulum whose length is the distance through which the spring is compressed by the load. In the case of the spring tyre this period is changed to that of a pendulum whose length is the sum of the compressions of the spring and the tyre. A new period also has to be taken into account, namely, that of the wheel and axle on the combined elasticity of the spring and the tyre. This is generally between one-quarter and one-fifth of that of the load. (See Appendix, where the formal relation between the two is given.)

As the ratio load/weight of the wheel increases, the vertical period of the load increases and that of the wheel diminishes, but the latter variation is not important.

If the elasticity of the tyre was due to a rigid but massless rim connected to the wheel by springs, the problem would be exactly similar to that treated in the first two cases, and would merely require the determination of the forced oscillations set up by the irregularities of the ground in the modes defined by the two free periods above mentioned. But in the case of india-rubber tyres, whether solid or pneumatic, account must be taken of another consideration depending on the nature of the contact of the tyre with the ground. In the case of a rigid tyre, whether backed by springs or not, the area of contact is so small that it may for most purposes be considered as a point. With rubber tyres, however, where the length of the contact is six inches or more, along which the pressure is fairly though not quite uniformly distributed, this supposition no longer applies. Inequalities of road surface (for instance, stones), which are small compared to the length of the contact, may make deep local impressions in the soft tyre, but have little effect on the mean pressure over the whole area of contact, and little effect, therefore, on the path of the axle of the wheels.

When the wheel with pneumatic tyres travels over ridges spaced at intervals of two feet, computing, as before, the forced oscillations of the axis, but assuming (in order to make the conditions conformable to those of a motor car) that the diameter of the wheel is three feet and the speed twenty miles an hour, I find that the variation of level of the axis is about one-third of the depth between the ridges, and that the consequent variation of the level of the load is little more than one-hundredth of that of the axis.

If we turn from the effect of the road on the vehicle to that of the vehicle on the road, the benefit of the introduction of springs between the wheel and the ground is equally conspicuous. The variation of the force on the ground is reduced in the ratio of the effective weight of the tyre (i.e., that part of the tyre which is deformed by contact with the ground) to the weight of the wheel. This ratio will vary for different types of tyre, but is of the order of one to a hundred. The contrast between the pneumatic and the hard tyres is striking, and indicates clearly why the former make high speeds not only practicable, but more comfortable than an ordinary spring carriage would be when travelling over the same surface at half the speed.

To convert the force acting on the ground into pressure, the force must be divided by the area of contact, and since with pneumatic tyres the area of contact is large in comparison with that of hard tyres, the use of the former not only reduces the variations of pressure but the pressure itself. It must be remembered however, that for rigid tyres on uneven ground the contact is no longer between a plane and a cylinder, but that both surfaces are curved, and that thus the area of contact for a given load is increased where the ground is hollow and diminished where the ground is convex, the pressure consequently being in this respect more intense on the prominences than in the hollows though the dynamic variations act in the opposite sense.

The damage done to a road by traffic, depends on the excess of the applied stress above the limit of stress which the road in question will stand without disturbance. The limiting stress for any given road depends in part on the nature of the road material, and in part on the manner in which the material is used. For example, if the road is made up of angular fragments of stone without any filling between them, then even if the stone be strong enough to bear the surface pressure and the surface were ground flat, there would be crushing below the surface at the places where the fragments touched one another. Soft filling material which separates the fragments at once reduces the pressures between them by enlarging the area of contact, and if the filling material be elastic as well as soft and of the proper consistency, no permanent effect on the interior of the road is caused by the passage of a load over the surface. Such wear as does occur in this case is confined to the surface. The degree of hardness requisite for the material of the road surface if wear is to be avoided, depends on the intensity of pressure to which it will be exposed, and the small intensity of the pressure produced by elastic tyres would allow much wider choice in this respect for a road where such tyres and mechanical traction made up the whole traffic. As long however, as iron or steel or indeed any metal is employed, whether on the feet of draught animals or on the wheels, it will be necessary either to surface the road with a substance strong enough to stand high pressure, or, as in the case of asphalt and wood, capable of yielding sufficiently without disruption.

Appendix.

In Fig. V let o be the axis of the wheel.

AB = L_1 = the compression in the spring due to the load w_1.

CD = L_2 = compression of tyre due to loads w_1 and w_2.

OD = r = radius of wheel.

v = constant horizontal velocity of axis.

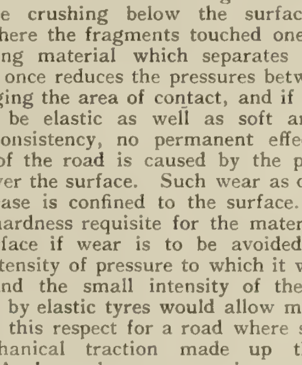


Fig. V.

The mathematical statement of the problem is:

A system consisting of two loads, w_1 and w_2, in which w_1 is sustained by a massless spring, and has the natural period T_1 on that spring, while w_2 rests on a second massless spring (the tyre) supported on the ground, and under the load $w_1 + w_2$ has the period T_2.

An arbitrary periodic vertical displacement is applied at the contact of the second spring with the ground. It is required to determine the vertical force on the ground and on the loads w_1 and w_2, and the vertical motion of the latter.

To do this, we may first calculate the forced vibrations of w_1 due to the action of the periodic displacement of the axis when the tyre is rigid, and then those of w_2 when the period of the tyre spring is not infinitely small.

Before doing this however, it is necessary to examine the nature of the arbitrary applied force. The origin of the applied force is the vertical component of the acceleration given to the tyre by the roughness of the ground, the surface of which is supposed to be represented by the sum of a series of parallel waves, each of the types—$a \sin p(x + x')$.

It will be supposed also that a/λ, (λ being the wave length) is small enough to make the maximum value of the angle between the perpendicular to the ground surface and the vertical such as to allow of the cosine of this angle being taken as unity. If $y = a \sin px$, be one of the terms of the series (where y is the ordinate of ground surface at x, and $p = 2\pi/\lambda$): then taking η and ξ as the co-ordinates of the axis of the wheel, we have

$$\eta = r + a \sin px \quad \ldots\ldots\ldots\ldots \quad (1).$$

$$\xi = x - rap \cos px \quad \ldots\ldots\ldots\ldots \quad (2)$$

These are the co-ordinates of the curves B_1, B_2, B_3 in Fig. 1.

The vertical acceleration of w_2 is $d^2\eta/dt^2$, and

$$d\xi/dt = v \quad \ldots\ldots\ldots\ldots \quad (3).$$

From the above equations we find

$$d^2\eta/dt^2 = -v^2ap^2\,(rap^2 + \sin px/1 + rap^2 \sin px) \ldots (4).$$

If we expand η as a Fourier series in terms of ξ, then v for every simple harmonic term in the series for the ground surface, a series of terms of the form $A_n \sin np\,\xi$ and $B_n \cos np\,\xi$, is required to express the arbitary displacement of the tyre.

The co-efficients A_n and B_n are very complex in form; and it is unnecessary to give their values here.

The force acting on w_2 due to the vertical acceleration is

$$w_1/g \cdot d^2\eta/dt^2 \quad \ldots\ldots\ldots\ldots \quad (5),$$

and the variation of tractive force necessary to keep v constant is

$$rap\, w_2\, w_1/6 \cdot d_2\eta dt^2 \quad \ldots\ldots\ldots\ldots \quad (6).$$

If the radius of the wheel is very small compared to λ, the path of the axis η, ξ, differs little from the curve of the ground surface, but as r/λ increases, ξ departs more and more from the corresponding values of x, and the path of the axis becomes flatter on the ridges and steeper in the hollows than on the ground surface.

The least radius of curvature of the curve $y = a \sin px$ is $1/ap^2$ or $\lambda/4\pi^2a$, and if $r = 1/ap^2$ the path of the axis has a cusp at $px = 2i\pi + 4\pi/3$ where i is any integer.

When $r > 1/ap^2$ there is a loop in the path of the axis which begins when ξ (or $x - rap \cos x) = \lambda$.

At this position the tyre touches the ground at two points $\lambda - x$, and $\lambda + x$ (P and P^1 in Fig. 1), and the part of the curve within the loop corresponds to no real contact of the tyre with the ground.

The vertical forces acting on w_2 in virtue of the curvature of the path of the axis, are illustrated in Figs. VI. and VII. for values of rather less and rather greater than the critical value $1/ap^2$.

When r is rather less than $1/ap^2$ the force acts upwards and increases rapidly as ξ approaches $(3/4 + i)\,\lambda$, and then rapidly diminishes, becoming a large downward force while the axis crosses the short concave part of its path and again acting upwards when the path becomes convex.

When r is greater than $1/ap^2$ the force acts upwards along the whole path except when $\xi = \lambda\,(3/4 + i)$, and then for an instant there is an infinite force acting downwards.

In what has been said it is assumed that the axis *does* follow the geometrical path prescribed by the ground surface, but, while it is possible that a force of any magnitude should act upwards on the axis, or downwards on the ground (the rigid ground supplying the necessary reaction), the only force which keeps the wheel in contact with the ground when the axis is being accelerated downwards is the weight w_2 + the force due to the compression of the upper spring.

Hence since the total force on the ground during the downward acceleration is $(w_1 + w_2)(1 + 1/g)(d^2\eta/dt^2)$, if $d^2\eta/dt^2$ exceeds g, the wheel will leave the ground and continue its path simply as a free body under the action of gravity and of the upper spring.

When the radius has the critical value $1/ap^2$, this must happen as ξ approaches $\lambda\,(3/4 + i)$ however small v may be, but the distance jumped decreases with v.

Wherever the wheel strikes the ground after a jump an infinite downward force would act if the ground was rigid, but, of course, the hardest blow is struck when the speed and slope of the ground make the impact take place at the bottom of a hollow.

We may now proceed to find the acceleration acting on w_1 (the body of the carriage), when travelling at velocity v over ground defined by $y = a \sin px$.

Suppose in the first place that the tyre spring is infinitely stiff (i.e., the tyre is hard). The

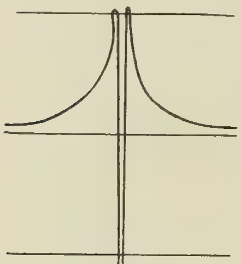


Fig. VI.

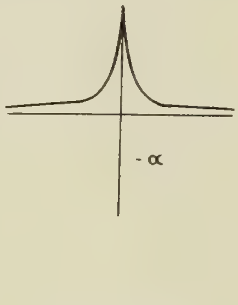


Fig. VII.

path of the axis in this case is that defined by η and ξ above.

This path being expressed as a Fourier series, consider the effect of any one of the terms in producing a forced oscillation in w_1.

The period of w_1 on the spring T_1 and if v $T' = \lambda$, T_1, $T_1/2$, $T_1/3$, etc., are the periods in which the various amplitudes, defined by the co-efficients of the series, act on the system; if $T' = q\,T_1$ and if A_n is the co-efficient of the nth term of the series, the amplitude of the resulting forced oscillation of w_1 can be shown to be $A_n \cdot n^2 q^2 / (1 - n^2 q^2)$.

To get the actual motions of w_1, however, this oscillation has to be combined with the free oscillation of period T_1, which must in general be introduced to represent the initial conditions of the motion, but the most important point is that unless $n^2 q^2$ lies between 1/2 and 1 the forced amplitude is less than A_n.

If $n^2 q^2 = 1$ the amplitude is only limited by the natural extinction, of which account is not here taken, and, of course, the conditions are entirely changed if the amplitude is great enough to make the wheel jump.

On most roads at ordinary horse speeds $n^2 q^2$ is much less than 1/2.

Proceeding to inquire into the effect on the path of the axis caused by the introduction of an elastic tyre, imagine first that the tyre consists of a rigid ring of weight $w3$, connected with the wheel by massless springs whose combined stiffness is the same as that of the actual tyre. The centre of this imaginary ring then follows the path defined by η and ξ and the variations of the force acting vertically on the ground are $w_3/g \cdot d^2\eta/dt^2$.

The period of w_2 on the combined spring of the tyre and the carriage spring is easily shown to be equal to $2\pi\sqrt{L3\ L4/g\ (L3 + L4)}$ where L3 and L4 are the compression caused by a force equal to w_2 acting on the carriage spring and tyre spring respectively. This period which we will call T3 is in general much less than the period of T_1.

Take for example the following case in which the various quantities are such as actually occur in practice. Let the compression L_1 of the carriage spring by w_1 be 3in., and let the weight of the wheel w_2 be one-tenth of w_1. Then L3 = 0.3 in. The compression of the tyre by the weight of the wheel alone L4 will probably be about 0.25 in.

Hence the period $T3 = 2\pi\sqrt{0{\cdot}3 \times 0{\cdot}25/g(0{\cdot}3 + 0{\cdot}25)}$ or 0·118 sec. nearly.

The period $T_2 = 2\pi\sqrt{(L_1 + L_2)/g} = 0{\cdot}55$ sec., so then T_2 is about four and a half times T_3.

We have to find what are the forced vibrations set up in w_2 by the relative displacement of the centre of the imaginary ring which follows the path whose co-ordinates are η and ξ and the axis of wheel which now only departs from motion in a horizontal straight line by virtue of the force exerted by the variations of η.

Taking, as before, A_n for the amplitude of the nth term of the series for η, the consequent amplitude of the forced vibrations of w_2 is $A_n \cdot n^2q'2/1 - n^2q'2$, where q' is the ratio $T_1/T3$.

The effect of the spring tyre in general is to considerably reduce the vertical motion of the axis of the wheel, and thus also that of the forced vibrations of the load it carries.

That is what chiefly affects the occupants of the carriage, but, as regards the effect on the road, the most important point is the immense reduction of variation of pressure which is due to the substitution of w_3 (the effective mass * of the tyre) for w_2 (or $w_1 + w_2$ in the case of springless vehicles) in the co-efficient of $d^2\eta/dt^2$.

Although we have been dealing only with forced vibrations, the free vibrations of the load which are necessarily present must not be forgotten. Their influence depends largely on the rate at which they tend to die out, but their existence is most readily noticed when the vehicle passes over a solitary irregularity, such as an elevation or depression on an otherwise smooth road.

* This is the mass deformed at any one instant by contact with the ground.

CORRESPONDENCE.

No responsibility is accepted by the Editor for the opinions of correspondents, and the right is reserved to omit any portion of a letter if it is considered desirable to do so. Letters to be inserted with a nom de plume must be accompanied by the name and address of the writer. If publication in the next issue is desired letters should reach the Editor in no case later than the 20th of the preceding month.

HORSE POWER FORMULÆ.

Sir,—I have observed for some time past some discussion in your columns relating to horse power formulæ.

I would suggest a settlement of this vexed question in the following manner:—That your Journal appoints you, and possibly one other engineer, to test all the favourably known h.p. formulæ. Of these there are about half-a-dozen in the running. This could be done in a very simple manner, as follows:—

That you obtain information from the makers of a selected dozen of the best designed and most up-to-date automobile engines on the market. This of course will necessarily mean those which are most efficient, and which give the best results, and the highest power from a given cubic capacity of cylinder.

Then having set out the twelve makers' engines selected, say one or two of each size with the h.p. obtained on the brake, as given by the maker, and then test each formula by means of these actual results, publish in your journal which formula was nearest to a correct result in all cases. This would probably give you about twenty-four engines upon which to test each formula and would be a fair and complete test.

So far as my own formula is concerned against any and every other, I have no doubt as to the results. Particulars of my formula (which has been published from time to time in the R.A.C. Journal, *The Autocar*, and other motor journals) and which is used by many automobile engineers in this country is appended hereunder:—

Where d is 4 in. or less than 4 in., the maximum of b.h.p. $= \frac{s\ d\ N}{2}$, but where d is greater than 4 in., then the h.p. $= \frac{s\ (d + a)\ N}{2}$; s is the stroke in inches, d the diameter of cylinder in inches, N the number of cylinders, and a number by which the diameter exceeds 4 inches.

The latter part of the formula, that for engines having a diameter greater than 4 inches, may be expressed in another manner, thus:—

$$\text{H.P.} = \frac{s\ (2\ d - 4)\ N}{2}$$

This formula assumes revolutions of about 1,500 p.m. If the engine whose h.p. is to be found runs at more or less a proportionate increase or reduction must be made. I have not applied it to engines of larger than 6 ins. diameter.

Two examples, one below and the other above 4 inches diameter are as follows:—

Example 1.

s = 4 inches

d = $3\frac{1}{2}$ inches H.P. $= \frac{4 \times 3 \cdot 5 \times 4}{2}$

N = 4

H.P. = 28

Example 2.

s = 5 inches

d = $4\frac{1}{2}$ inches H.P. $= \frac{5\ (4 \cdot 5 + \cdot 5)\ 4}{2}$

N = 4

a = $\frac{1}{2}$ inch H.P. = 50.

J. D. ROOTS.

A PLEA FOR WOOD.

Sir,—Reading your editorial leader in the *Automobile Engineer* of March, 1911, I have been very much interested, and quite agree with you in many points raised therein.

In that leader you say, "It is an outstanding fact that very few manufacturing concerns with high reputations as private car makers have been able to make satisfactory heavy vehicles at the first attempt, while in more than one instance they have tried to obtain a footing in the newer market only to abandon the endeavour as being damaging to their prestige"; and a little further on you say "It is questionable whether a complete study of any one of four separate products of automobile engineering is not sufficient fully to occupy almost any one man, the four being—flying machines, touring car chasses, vans or light waggons, and heavy waggons or tractors."

Now, Sir, I am quite agreed with you that, with the designer, any one of the four studies is quite sufficient to occupy the whole of one man's time and brain for the time being, for if he tries to grapple with other questions of a totally different nature, it is almost certain that some important point will be overlooked which may cause the whole of his labours to result in failure. Turning to the question of difference between a private car and a motor waggon—

First, and most important of all, is their loads. It does not matter whether a car is carrying one person or 20 persons, weighing a ton, or a motor waggon carrying a ton of flour or coals. In the case of the car it is a *live* load, and the waggon has a *dead* load, which are directly opposite in their requirements as regards the nature of the construction of the conveyance, for in the first case a live load will adjust itself to the irregularities and conditions of the road, while a dead load is always the same and gives the greatest resistance possible to such irregularities.

The second important point the designer must consider is how he will be able best to minimise and absorb the difference between a live load and a dead load, for while the all-steel frame and numerous springs, live axles and pneumatic tyres, combined with well-upholstered seats, only serve in a very moderate way to minimise shock when carrying a live load, they become totally inadequate for carrying a dead load, and he has to turn his eyes and thoughts in other directions in search of other material more suitable to his needs for the construction of the motor waggon.

The third important point is from a financial point of view, where wear and tear is an important factor, the tyres playing the most important part, for, with a dead load, a rigidly constructed waggon on an uneven road is sure sooner or later to end in disaster. The wear and tear due to a waggon so constructed is sufficient to seriously handicap any business. Speed is a secondary consideration.

Now if we consider the three points raised in the foregoing, the first we cannot alter, but the second the designer will be able to deal with in the most efficient manner by the introduction of woods of enormous strength in the construction of his main frame, instead of steel, for there are now plenty of well-known woods (especially in the Colonies) which, taken on a weight-for-weight basis, for comparison of strength compare most favourably with the best brands of steel, and in some cases are superior to the best nickel steel, and they are (many of them) only in the least degree affected by the varying conditions of the weather, and in a few cases are better without being protected by paint, while they have the great and multiple advantages of being cheap in first cost, cheap to manufacture, requiring very few tools, at the same time providing a stronger frame, and one which absorbs readily disagreeable shocks and bumps.

H. MAPLETHORPE.

AN OPINON AND REQUEST.

Sir,—Taking into account that I am a regular reader of your very interesting paper since its beginning (in fact I buy two copies of your paper every month), will you kindly allow me to address two observations, or rather call your attention to two points which, I believe, might contribute or afford some improvement to your valuable paper?

I.—No doubt *The Automobile Engineer* is the best paper published yet out of the English and French papers on the subject. But I am sure it will certainly be of great interest to the automobile people if its Editor would give more room (or rather any, as none up to now has been allowed to it) to the interesting class of automobile vehicles called steam waggons, motor lorries or motor tractors—and certainly this matter would be more in its proper place in *The Automobile Engineer* than the notes or descriptions of aeroplanes, air ships, etc.

II.—I buy every month two copies of each issue, because I put aside all your notes (as I have done for some 10 years, for all the notes are interesting in *The Automobile Engineer*), and to classify each note or design in its proper place or batch, I have to put sometimes the *recto* in one place and the *verso* in another. This fact is common with all the publications or technical magazines and reviews I know. I believe it would be really an improvement if you would take the trouble or the care that each matter or chapter begins with the top of a page and is completed within the same page, so that the reader may easily tear off the page of special interest to him, without taking off half of another chapter.

L. FIALON.

ERECTING SHOP TROUBLES.

Sir,—I should like to know whether some space could be devoted to the consideration of workshop system, having special reference to the erecting shop, because a study of prevailing methods would lead the uninitiated to suppose that the majority of works either do not realize the real usage of an erecting department, or have no system at all in connection therewith.

Now it seems to me that the duty of an erecting shop should mainly consist of assembling and lining up those parts which have already been accurately machined to some jig in the machine shop, and that steps should be taken to discover any part which, by bad machining, or bad draughtsmanship, may prove a considerable hindrance or entail a corresponding amount of hand work, and to report thereon at once to the powers that be.

As things are at present the most curious and wonderful happenings may be observed at any time in most assembling shops. Mostly these are no doubt small matters, but it is these small matters which run up the erecting cost to such a very great extent. To take concrete instances of this curious phenomenon, there was a firm who had, and still have, a great reputation for excellence and careful construction, yet that firm saw fit to supply frames to their erecting shop to which dumb irons were rivetted, rendering the insertion of a spring eye impossible until the interior has been chipped out by hand, thus entailing a considerable waste of time on the part of one fitter and the gang's boy, who was required to hold a heavy metal block against the dumb iron until chipping was completed. Now, had this occurred with the first dozen or so cars put through nothing could have been said, since there is invariably a considerable rush on these first cars, but in this case all that season's cars had to be fitted in the same manner, although an attempt was made to introduce a hand miller for the job about the middle of the season, which failed because it was impossible to insert the milling cutter until a considerable area had been chipped out to take it. Everyone knew that such a waste was going on, but no serious attempt was made to alter things.

In another case a control bracket could not be fitted to the frame side until a very considerable amount of that bracket had been removed with a file, causing some comparison between milling and filing times. One would imagine that the weakening of a pressed steel frame by recessing the flange to take a float chamber would be the cause of instant trouble, yet such a thing has been known to continue throughout an entire season. Again, it was the practice in many shops to force a big end to free itself on the crank pin by vigorous use of a hammer, so that time may be saved, and the return of the bearing to the machine or to its original fitters obviated. Undoubtedly time was saved to the erector, but the number of connecting rods returned after test was sufficient indication of the value of such practice.

Such things as these argue a lack of detail appreciation, and form one of the reasons why most erecting shops are so inefficient. Moreover, it would appear that the average draughtsman knows little or nothing of what actually is taking place in the assembling shop and goes there for the purpose of admiring some new fitting or model rather than for the study of detail improvement.

The attention of a charge hand may be drawn to the various points which delay his gang, and under special circumstances he may report it to the shop foreman. In many cases however, the charge hand has a soul above such trivialities, and enough to occupy him in the time sheets and bonus cards of his men.

There remains the interference department, as it is generally called. Here again it is more usual for time to be spent in watching the output of the machine shop than in checking jig accuracy, or rendering the erecting shop more efficient. And consequently there seems no reason why there should be the slightest improvement in the latter department, unless the much-talked-of American invasion actually materializes, thereby necessitating an alteration in production methods.

If this should be so it might direct the attention of those responsible to the enormous difference between erecting as it is practised in the United States and the job bearing a similar name in this country—with a consequent improvement in our own methods.

PRACTICS.

CHAIN-DRIVEN GEARBOXES.

Sir,—I notice in Mr. J. R. Cautley's article on "The Purposes of Chain Driving," that he mentions the chain-driven gearbox of the London General Omnibus Co., Ltd., and states that the success of the same is to a large extent problematic, as the boxes have not been running long enough for an authoritative statement to be made.

I wish to state that we have now 300 of these chain-driven gearboxes in regular service, each doing their 115 miles per day, and the first 60 which were put into service have done an average of 30,000 miles each during the past 12 months.

This experience has proved to me that adjustment of the chains is unnecessary, as is also the stopping of the lay shaft to avoid wear of chains when on top speed, such wear being almost negligible.

Mr. Cautley mentions that these boxes are running well by reason of the fact that they are in the hands of skilled drivers, but I am afraid that this gentleman's knowledge of the gear changing skill of the London 'bus driver is not very great, for as a matter of fact they have managed to damage every other type of gearbox that we have had, with the exception of the one in question.

A great charm is that the chains act as a safety valve in the case of brutal drivers, as should the clutch be let in suddenly with the engine racing and a fully loaded vehicle, it would only break a chain, which could be repaired at the cost of 1s. or so, and no further damage would be done.

In my opinion, this is the most successful gearbox that has been produced for change speed gearing on motor vehicles.—Yours faithfully,

FRANCIS SEARLE.

THE CONSTRUCTION OF WHEELS.

An interesting lecture on the subject of wheel construction was given by Mr. H. L. Heathcote, B.Sc., before the Royal Society of Arts, on April 5th, dealing with the whole history of wheels as applied to vehicles, and describing in comparatively minute detail the processes of wheel construction used in the Rudge-Whitworth factory. The most instructive portion of the lecture was undoubtedly that in which Mr. Heathcote pointed out the difference between the ordinary wooden wheel and a wheel with tension spokes, from an elastic point of view. It was shown that in order to obtain a maximum amount of cushioning from a wheel itself, apart from the tyre, it was best to use a tension spoke wheel with a moderate spoke tension and with a flexible rim, so allowing the wheel to give slightly under shocks applied radially, precisely after the fashion of the pneumatic tyre filled with compressed air.

Wooden wheels, it was stated, owe much of their strength to the weight of their felloes, which was disadvantageous with a wheel which should preferably be made to absorb some shock.

Passing on to the consideration of Rudge-Whitworth wheels in particular, it was shown that the cushioning effect of the wheel was improved by dishing the outside spokes considerably and causing them to enter the rim near the edges instead of the centre, the idea being that the outside spokes support the weight elastically while the inner spokes take the greater part of the driving torque, there being fifty per cent. more spokes on the inside than on the outside of the wheel.

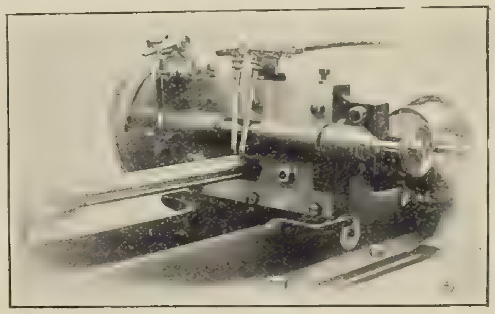
Finally the lecturer explained some of the tests to which the component parts of Rudge-Whitworth wheels are subjected, including the well-known impact pendulum, the annealing process for hub shells, and the methods of spoke tension testing by means of comparing the pitch of the note emitted by a spoke when struck with a standard tuning fork.

CASTELLATED SHAFTS.

Referring to the letter from Mr. Marcus C. Hunter, which appeared in last month's issue, a correspondent says:—

"Mr. Hunter gives an illustration of a type of sliding gear shaft and states that these are milled by using a special cutter. That this style of gear shaft has some good points is evident from the fact that it is in use by a number of the leading French automobile builders, and—as Mr. Hunter states—one of its principal points is that by grinding what was originally the hole in the gear to a true circle, the true running of the gear is assured, provided, of course, that the grooving of the shaft on which it fits is also accurately done.

Now to mill these shafts by using a concave face milling cutter is not a very easy job, owing to the difficulty of maintaining the exact curve on the cutter, with the radial sides, and to ensure that all the grooves are cut to exactly the same depth. It may therefore interest your readers to know that a special machine for cutting these



grooves is being put on the market here by Messrs. C. W. Burton, Griffiths and Co., who are the British agents for it. The advantages of the machine are several:—Firstly, the intermittent indexing of the shaft, by which the process is carried out in rotation on each groove until they all reach the required depth, that is, at every stroke of the tool the shaft is rotated to the next groove, and so on. This ensures that all the grooves are cut to the same depth, as the feed of the tools takes place only after the shaft has made a complete revolution. Secondly the use of what are practically planing tools of very simple form, which are used in pairs, working from each side of the shaft, which must have an equal number of grooves. Thirdly, the provision of a special apparatus by which the tools, when mounted in the machine, are ground to the exact diameter of the shaft when the grooves reach the required depth.

The illustration shows the tools in position on a shaft, and the time for grooving an average gearbox shaft is claimed to be about twenty minutes.

THE SOCIETY OF AUTOMOBILE ENGINEERS (U.S.A.)

The summer meeting of the S.A.E., which now has a membership roll of over eight hundred, will take place at Dayton, Ohio, on June 15th, 16th, and 17th, on the invitation of the Dayton Chamber of Commerce and the Wright Brothers, amongst others.

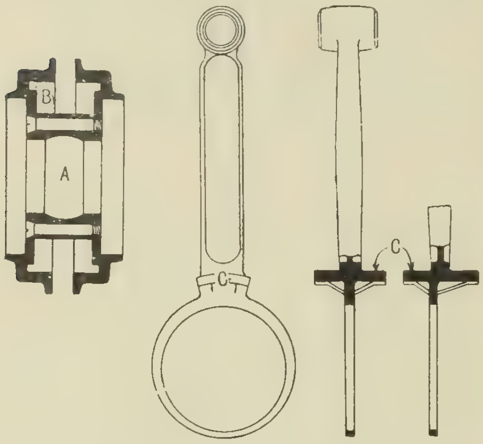
In addition to the papers to be read, various social events have been arranged, and there are several aviation fixtures.

RECENT AUTOMOBILE PATENTS.

By Eric W. Walford, F.C.I.P.A.

Connecting Rod System for Revolving Cylinder Engines.

In the Gnome engine one of the connecting rods takes a ring bearing on the crank pin, and the other rods are pivotted to this master rod, naturally at a slight

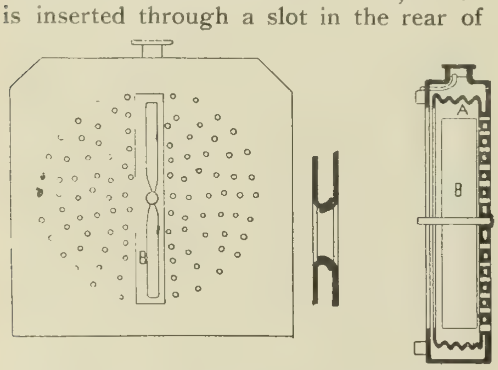


distance from the axis of the crank pin. Consequently these connecting rods are not always radial to the crank pin. In the present case the crank pin is provided with the bearing member marked A, and each of the rods terminates in a thin ring portion which surrounds the bearing portion A, lying in the space B. The ring portions are differently arranged on the different rods. For instance, in one rod the ring portion will be central, whilst in the others they will be more or less to one side, so that all the ring portions may be accommodated in the space B, and the connecting rods are maintained central. Each connecting rod is formed with shoulders at C, which engage the underside of the recess B, and take the radial stresses due to centrifugal action. It will be seen that the bearing portion A is made in two parts allowing the ring portions to be threaded into place.

Société Anonyme Rossel-Peugeot. No. 27,660/10.

A New Type of Radiator.

The radiator comprises one tank within another, the two being spaced so as to provide a jacket around a central air chamber A. In this lies a fan B, which is inserted through a slot in the rear of

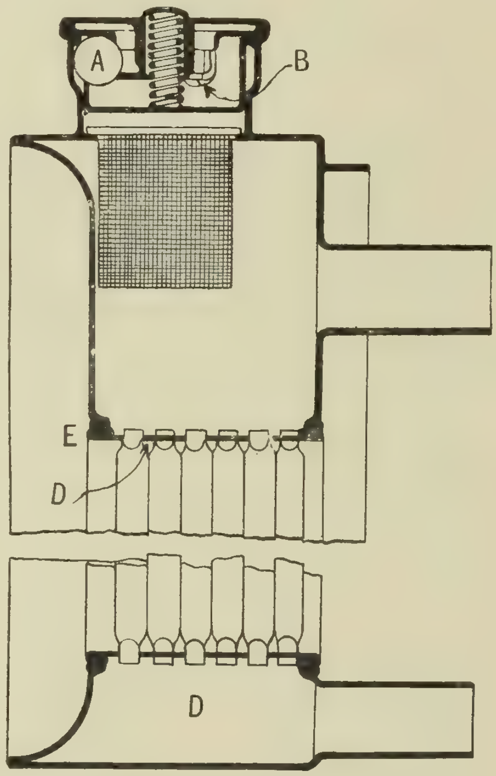


the radiator. The two tank plates are formed with holes, the metal knocked out of place to form one hole being attached to the corresponding metal of the hole in the other plate as shown in the detail view. These flanges are soldered together, forming a water-tight joint around the hole, and through these holes the air is drawn by the fan.

V. Luypearts. No. 16,153/10.

Another Radiator Construction.

The top and bottom tanks are formed without soldered joints by being built up from sheet and tube material with the joints acetylene-welded together. The same applies to the filler, which is provided with a special cap retained by the locking discs A, which are moved into their inoperative position by pressing down the central plate B, which is provided with a knob projecting through the filler. The tanks are provided with curved flanges in order to direct the air through the cooling tubes. The tubes are flat in their middle parts with round ends for attachment to the plates D, the flat parts being spaced by corrugated strip material, which stiffens the tubes and provides increased cooling surface. The cooling tubes when attached to their plates D can be moved in and out of the tanks quite easily. The plates and the adjacent parts of the tanks are provided with turned-over edges, and between

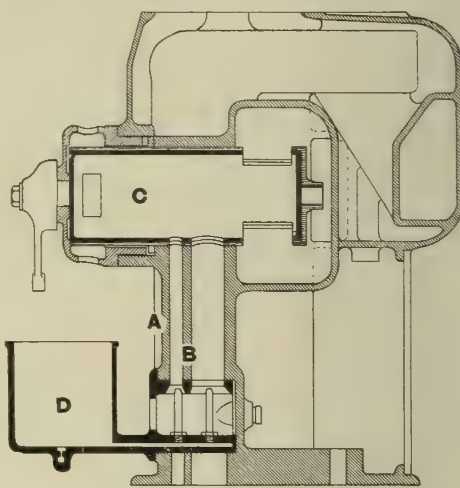


these lies a tapering strip or wire E, which is soldered in place. To remove the cooling system this wire is unsoldered and removed, allowing the plates and tubes to be withdrawn.

F. Lamplugh. No. 5,503/10.

A Carburettor Arrangement.

To prevent having to provide special means for heating the carburettor chambers, and to simplify the piping arrangements, the cylinder has the usual carburettor passages cast in it, and the float chamber which carries the jets is attached thereto. The central pair of cylinders in the case of a four cylinder engine are set at a suitable distance apart, which incidentally accommodates the central crank shaft bearing. At this point the cylinder casting is formed with a web A which has a rib B bored out to form the main carburettor passages. These communicate with the throttle barrel C which is located in the water jacketted part, whilst into the lower end of the carburettor passages project the jet nozzles which are carried by a hollow arm extending from the detachable float chamber D. In this man-

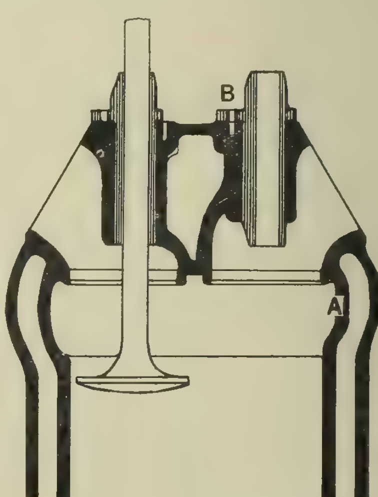


ner external pipes are dispensed with, and also special means for heating the carburettor.

No. 21,741/10. Fabrica Italiana Automobili Torino.

Valve Arrangement.

One of the chief difficulties with overhead valves arranged to open directly into the cylinder head is to provide a construction which can be used with cylinders of small bore having large valves. Usually the cylinder is formed with a recess at A to accommodate the large valve heads, but difficulty is experienced in putting these into place. This is overcome by making the valve guide eccentric and setting it out of its normal position, allowing the valve to be put into place as illustrated. When the valve head has reached the recess A the valve guide is turned around into its proper position and locked by inserting a set pin at B. The



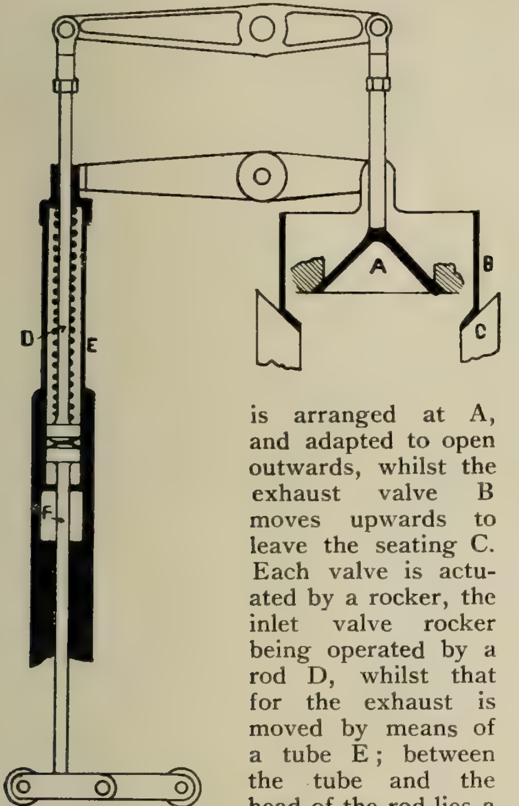
movement of the valve guide brings the valve into its proper axial position. In an alternative construction the valve guide is not put into place until the valve has been put into position. Thus the clearance between the valve stem and the housing of the valve guide is used to allow the valve to be put into place out of

its proper axial position and then set concentric with the housing when the guide is fixed.

No. 6,278/10. Daimler Motoren Gesellschaft.

Valve Mechanism.

The valves in this system are arranged concentrically, the drawing illustrating this diagrammatically. The inlet valve

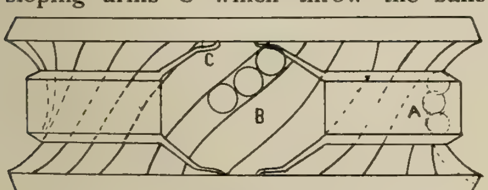


is arranged at A, and adapted to open outwards, whilst the exhaust valve B moves upwards to leave the seating C. Each valve is actuated by a rocker, the inlet valve rocker being operated by a rod D, whilst that for the exhaust is moved by means of a tube E; between the tube and the head of the rod lies a compression spring which forces the rockers in opposite directions and keeps their respective valves on their seats. The valves are operated by a rod F, which when pulled downward moves the sleeve E against the compression of the spring, and when forced upwards slides the rod D and opens the inlet valve, also against the compression of the spring. The rod F is actuated by a rocker centrally pivotted and engaged by cams at both ends, one cam being used for pushing the rod up, and the other for drawing it down.

R. Esnault-Pelterie. No. 12,830/10.

A Ball Bearing Worm Gear.

The worm itself is of the ordinary type, and the worm wheel is formed with grooves of a suitable shape to retain bearing balls of which, in the construction illustrated, there are three in each groove, which are prevented from dropping out by means of a retaining ring A which surrounds the wheel. A gap is provided at B for the worm, and at this point the ends of the retaining ring are formed with sloping arms C which throw the balls

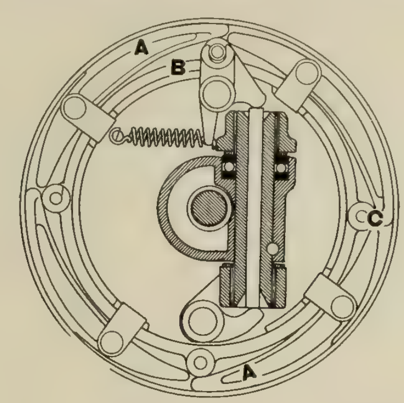


back into the retaining ring as soon as they are relieved from the pressure of the worm. It will be seen that the balls move laterally into and out of the retaining ring.

No. 9,725/10. W. Hillman.

An Elaborate Brake.

The brake drum is engaged by a number of blocks A pivotted to a fixed disc and the engagement is effected by moving in a contra clockwise direction a ring C which carries rollers D adapted to engage the braking ends of the blocks A. When moved in the opposite direction the ring engages the opposite ends of the blocks and tilts them positively out of engage-

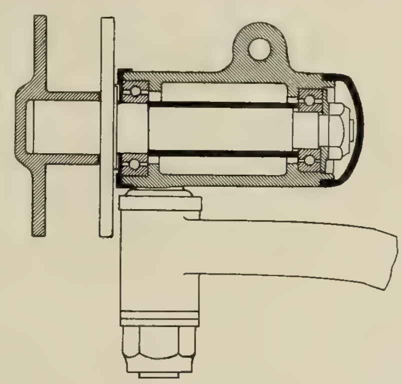


ment so that there can be no drag when the brake is off. The brake is actuated by a rocker C by means of a bell crank lever, one end of which is acted on by the spring shown and the other by a rod which passes through the steering pivot.

No. 19,760/10. K. I. Crossley and A. W. Reeves.

A Curious Steering Wheel Arrangement.

The spindle of the steering wheel is extended over the steering pivot and is mounted in bearings as shown. The object of the invention is to obviate any parts projecting outside the wheel plane, but it would seem that this is attained at



the expense of certain disadvantages, for the whole of the weight is carried at the bottom end of the steering pivot and a heavy load is put on the nut or other securing device, while further, the wheel bearings are at a considerable distance from their load.

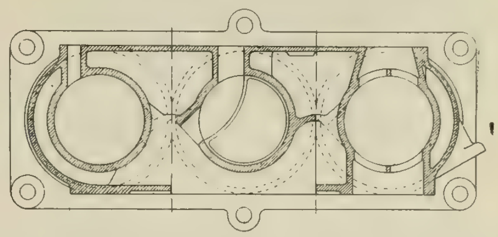
No. 23,339/10. M. J. Barreau.

A THREE-CYLINDER TWO-STROKE ENGINE.

Endeavours to design a satisfactory two cycle internal combustion engine have been extremely numerous and at the present, invention is directed principally towards the eliminaton of crank case compression with its customary attendant disadvantages, although it may certainly be allowed that a few engines compressing in the crank case are most satisfactory.

On the next page we illustrate an engine recently made by Mr. W. M. Appleton, in which the mixture is compressed by a piston pump thereby giving higher final compression and better scavenging than usual. The drawings show the first engine manufactured which has three cylinders, although this number is not essential to the principle. It will be seen that the piston is duplex, and the compression chamber, is, of course, the annular space round the trunk and above the lower portion of large diameter. The lower part of each piston, in fact, acts as a compressor, sucking mixture from the carburettor and compressing it into a transfer chamber, but instead of endeavouring to hold the compressed mixture in order to pass it to the working cylinder immediately above the "pump" at the right moment, the cranks are set so that the pump from one cylinder feeds the combustion chamber of a cylinder adjacent, and so on, the transfer being direct and the compressed gas remaining for but a very short period of time in the transfer passage.

The exhaust ports are uncovered by the descent of pistons in the ordinary way and the inlet ports are disclosed similarly, but the passage of gas from the carburettor to each pump is controlled from ports through the pistons themselves, as may



Section through cylinders and also the piston of the middle cylinder showing intake passage through the piston.

be seen in the plan sectional view. Necessarily everything depends upon the accurate proportioning and arrangement of the ports, but an extremely rough experimental engine has been shown to be capable of running at some 2,000 r.p.m., notwithstanding very unnecessarily heavy reciprocating parts, and somewhat cramped passages; this being with a 4 in. bore and a 5 in. stroke. At the same time, 35 h.p. has been obtained at 1,100 revolutions, and considerably better things are expected from an improved edition.

Unlike the majority of its kind, this particular two-cycle engine runs well, over a big range of speed, and should show a horse power curve not greatly inferior to that of a four cycle engine, if indeed it would be inferior at all. The principle would seem to be promising, the obvious disadvantage being the great weight of the pistons, but these need not be by any means so large as is the case with the experimental engine.

Really the most noticeable point about the experimental engine is the flexibility and smoothness of running at the slower speeds of revolution. When making only a few hundred revolutions per minute there was no misfiring or "skipping," and the same may be said, with equal truth, of the other end of the speed scale. In this first engine there were too, a few obvious constructional faults which could be removed easily in future designs. Thus the intake passages were too small, causing wire drawing at high speeds, and the inventor believes that the ports could be enlarged slightly with advantage.

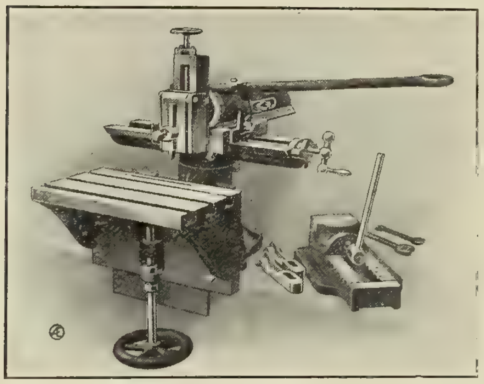
MISCELLANEOUS.

A HANDSHAPING MACHINE.

In the majority of small garages which are to be found in odd corners of the country, there is very seldom any facility for dealing with the occasional serious breakdowns which may occur within their province. However excellent they may be in other respects, most of these small garages are unprovided with any sort or description of motor power. With these difficulties in view, and with the idea of placing an efficient repairing plant within the reach of such small shops, Messrs. Nelson Bros. have brought out a useful form of hand shaping machine, which is capable of dealing with fairly large work.

The illustration shows the machine and its accessories as delivered to a customer and readv for erection in the shop.

A number of bolts secure the cast-iron base to the bench, on which it is designed to work, rigidity being obtained by the large size of the base and by a lip formed on the forward end of the iron casting. Attached to the base by a



A Small Hand Shaper.

sliding joint is the work table, which is provided with slots for the holding down and positioning bolts, and is operated by the large handwheel shown at the bottom of the illustration On the right can be seen a form of vice, which may be fitted when it is desired to operate upon round bar, or other material which can be more conveniently positioned thereby. Traversing is effected by means of the screw thread and crank gear visible on the right of the machine immediately below the operating arm.

Attached to the traverse screw is a catch by means of which an automatic traverse is obtained when the operating lever moves to the end of its stroke. A considerable leverage can be obtained from the operating lever in order that fairly heavy cuts may be used without undue exertion. Above the tool holder the feed screw can be seen suitably placed for easy operation when the machine is used by one man. All bearings have an ample surface, while the whole machine seems strongly built and capable of standing up under the rough treatment which it is only too likely to receive.

Although not a machine from which a very great amount can be expected, it is nevertheless one of the most useful tools which can be placed in a garage in which there is no power plant, and, by the exercise of a little thought and ingenuity, can be made to carry out quite unexpected jobs. There is also a possiblity that it might prove of use in a large garage where it would be uneconomical to start the power plant for the purposes of a single emergency job at an unexpected moment.

CATALOGUES RECEIVED, Etc.

ROLLER BEARINGS.—A new list of their manufactures in this department has been issued by the Auto-Machinery Co., Ltd.

MOULDING MACHINE.—A new moulding machine is dealt with fully in a booklet obtainable from the Adaptable Moulding Machine Co.

CHAIN DRIVING.—Chains for the transmission of power in shops and for other purposes, are described in a pamphlet which reaches us from the Westinghouse Brake Co., Ltd.

CHAINS.—The Coventry Chain Co., Ltd., have issued a new list of chains for all classes of work especially for automobile purposes. In it applications for camshaft driving are described.

HALLITE JOINTING is now being handled by Hall and Hall, who have purchased the business from Hallite Douglas, Ltd.

TESTING MACHINES.—Some very large tensile and compression testing machines have been installed by the Electrical Testing Laboratories of New York.

THE HURST ELECTRIC MANUFACTURING CO., Belfast, are now the principal agents in Ireland of the Electric Ordnance and Accessories Co., Ltd.

CHAIN DRIVING.—By a printer's error the name of the author of the above-named article, which appeared in our last issue, was given erroneously as Cautey instead of Cautley.

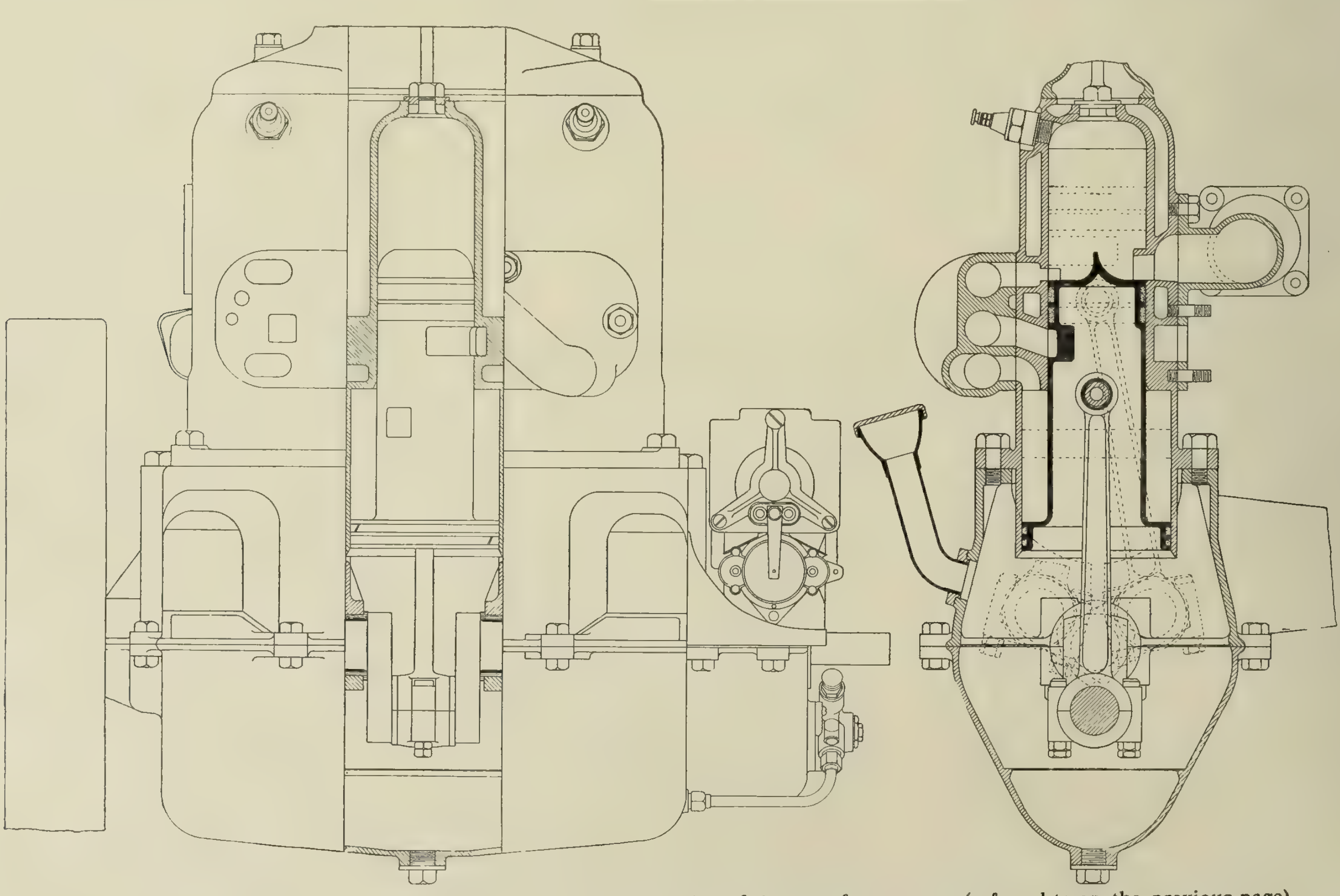
THE MOSS GEAR CO., LTD., is the title of a new concern whcih has just been started by Walter Duckitt, late of the Austin Motor Co., Ltd., and Charles Taylor, of Birmingham. The new company will specialise in the manufacture of cut gears for the automobile trade, and also will manufacture cam shafts and spare parts.

CARBURETTORS.—As an advertisement of the Solex automatic carburettors, S. Wolf and Co. have issued a large sectional diagram intended to be placed in some noticeable position on the wall of a garage. Every part of the carburettor is shown in section, and the drawing would be useful should an owner or his mechanic desire to dismantle this particular accessory.

STEEL WHEELS.—The Atlas Resilient Road Wheel Co., Ltd., are publishing a booklet descriptive of their steel wheels for motor vehicles. In it are described several different types, including a new extra light wheel made to assist motor omnibus constructors in conforming to the Scotland Yard regulations. Another apparently very strong wheel is a steel type made for steam and other heavy wagons with a good tyre-carrying rim outside the steel rim, permitting tyres to be renewed by ordinary methods.

THE SKEFKO BALL BEARING is now being handled in this country by the Unbreakable Pulley and Mill Gearing Company, Limited. This bearing has already been described in *"The Automobile Engineer,"* but the makers have recently published an interesting table showing their effect of speed upon the load capacity. This is in accordance with the formula $P = \frac{1}{5} Z K D^2$, where P equals the safe load in lbs., D the diameter of the balls in $\frac{1}{8}$in. units, and K a coefficient obtainable from the table below :—

Revs. a min...		50	300	600	1,200	2,500	5,000	10,000
Co-efficient K	44	26·4	22	17·6	13·64	10·56	7·7	5·06



The Appleton Two Stroke Engine, showing the valve ports and a section of the transfer passages (referred to on the previous page).

THE

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CONTRIBUTIONS.

Articles of a technical nature relating to the design or construction of automobiles for land, air, or water, will be carefully considered by the editor. Matter must be clearly written or typed on one side of the paper only, and a stamped addressed envelope must be enclosed for return. No responsibility can be accepted for the safety of contributions although every reasonable care will be taken.

Correspondence on interesting subjects, which come within the scope of this journal is cordially invited. All letters must contain the name and address of the writer, but not necessarily for publication. If a reply is desired by post enquiries must be accompanied by a stamped addressed envelope.

CONTENTS.

RACING AND ITS EFFECT UPON DESIGN.

WHETHER racing exercises a beneficial or an evil effect upon the design of touring automobiles has been a frequent subject of discussion for several years. It has, we believe, never been disputed that in the very early days of motoring almost every improvement was traceable directly to racing. As soon as the touring car ceased to resemble its racing prototype in outward appearance there arose critics saying that the usefulness of racing was dead, and amongst a certain section of automobilists at the present day it seems to be an accepted thing that there is nothing whatever in common between the ordinary touring car and the racing vehicles to be seen on the Brooklands track, or in the great road races on the Continent. The usefuless of racing has been perhaps the more obscure by reason of the fact that competitive speed work has long been regarded purely and simply as a matter of advertising. Certain firms have abstained from racing, and even from reliability trials as well, for the reason that "they did not need the advertisement." The buying public have been taught by many writers that the success of the racing cars made by any particular firm was no criterion of the quality of the touring vehicles made by the same firm, and it has been stated over and over again that the publicity gained by successful racing is not productive of sufficient orders to make it worth while for motor manufacturing concerns to enter for competitive speed events.

It seems almost universally to be neglected that racing has an indirect usefulness which probably greatly exceeds its immediate and traceable effect upon the returns of sales departments. It is safe to say that more can be learnt concerning materials of consruction from half a dozen racing cars than from five hundred touring vehicles, and the same holds good for many features of design. It takes but little reflection clearly to see that the majority of really good touring cars are those made by firms who have now, or have in the past, possessed high fame because of their racing successes. It is, of course, easy to argue that cause and effect have been confused in the preceding sentence; that the reason the best racing reputations were possessed by the leading makers was that they, having a large touring car trade, had more to spend on the building of racing machines; but no one who has studied automobile design for even so short a space of time as three years past could fail to appreciate the direct effect of even "freak" racing machines on standard practice.

Most noticeable, of course, is the long stroke engine. While in this country taxation has doubtless helped towards the production of the present dominant type, the same influence has not been at work in France, and it is safe to say that long stroke engines developed more rapidly in the last-named country than here, the starting point being the successes of long stroke engines in the Grand Prix des Voiturettes of 1909. At first, when long stroke engines were fitted to touring cars, many objections could be urged against them. Usually, they were noisy, harsh running and difficult to control, that is to say, although they were excellent racing engines, they were not good from a touring car point of view. Now, however, most of these difficulties may be said to have been overcome quite successfully, with the result that the modern touring car has a smaller, more efficient and more economical engine than its predecessors of but a few years back. That this would have been so had it not been for racing, is extremely difficult to believe. No doubt the long stroke small bore engine would in time have proved its usefulness, but had there not been reason for the concentration of the attention of designers throughout the automobile industry simultaneously to this one type, its development must have taken very much longer than it has done.

Returning again to the particular race that has just been mentioned, the advertising value of this event was very small, it resulted in increasing the fame of—at the most—three French manufacturers, and if it is looked upon in this way only it might be said that all the other entrants had wasted money. However, taking the broadest possible view and regarding the matter from the point of view of the prosperity of the industry as a whole, there is no doubt that the introduction of the powerful nominal 15 h.p. cars of to-day has resulted in an enormous increase in the number of motor-car owners. A really satisfactory vehicle has been put within the reach of an enormous number of individuals, the expense of these cars being small both as regards first cost and cost of upkeep. This has resulted in a commercial activity little short of a "boom" with this particular type of car. Now it has already been pointed out that the development of the 15 h.p. (that is to say, the 80 mm. x 120 mm.) type of car is traceable directly to the immense interest in longer stroke engines aroused by the 1909 Grand Prix des Voiturettes. This race has therefore resulted directly in the production of a new type of car, and indirectly to the opening of a new market for cars; also to the spending of many thousands of pounds by private individuals to the advantage of the automobile industry.

Turning to a less wide issue, and considering the case of a single manufacturer, in this country he can obtain a certain reputation by successfully racing cars on the Brooklands track. To do so, means an expenditure of a large sum of money. Such vehicles as the 16 h.p. standard class racers now are, cost many hundreds of pounds to build, and even more to alter and maintain on the track, but this expenditure ought not to be charged against the advertising account of the firm *in toto*. Even if ill-luck prevents the winning of a single race throughout the season, it is certain that the endeavour to win teaches the designing staff of a firm so much which will tend towards the betterment of the firm's cars as a whole that it is well worth the money. Whence, for example, have arisen the present excellent lubrication systems which are common in modern touring cars? Why is it that a 40 b.h.p. car weighs but little more than half what it did three years ago? Why is it that over-heating is a trouble entirely of the past? Why is it that cars, taken altogether, are easier to steer and better sprung than they were?—In each case there is no doubt that racing has had an enormous influence for improvement.

Of course, development from year to year, the knowledge gained from the car of one season and put into the car of the next, counts for a great deal, but not for much when it is borne fully in mind that many things pass unnoticed during ordinary usage on the road, while standing out with great prominence on the track.

Just to touch on quite a few points in engine design : the very light pistons of the present day, the well-balanced parts and the size of the valves, owe their origin almost entirely to racing, because the need for care with regard to these particulars is not noticeable except when it is desired to obtain the utmost possible power from an engine for a comparatively long period of time. It must not be forgotten that the engine of a racing car is running under practically the same conditions as the engine of a touring car on the second speed on a level road with everything opened up fully. The most ignorant of drivers would scarcely expect his engine to hang together were he to treat it thus, and obviously therefore, the racing engine which has got to stand similar rough handling must be made with the most extreme accuracy and care. Another point which may be mentioned is that very much concerning springing has been learnt at Brooklands. Not only is the track surface comparatively rough, but at high speeds the effect of small undulations is very marked, wherefore it has been found necessary to design special springs for light, fast racing cars, or to fit various types of shock absorbers, and by such experimenting certain things have been ascertained which, even if they were known before, were not appreciated at anything like their proper value.

It would be possible to continue to magnify instances at very considerable length, but no useful object would be served by doing so, as the purpose of this article is not so much to defend the usefulness of racing, as to point it out to those who have not yet realised the value of speed competition to the designer, and, through the latter, indirectly for the betterment of the financial position of his employers.

WORM AND BEVEL GEARING COMPARED.

During the past year there has been considerable argument as to the relative advantages and disadvantages of bevel gearing as a final drive for touring car work. Mechanical engineers generally, have always had a distrust of the worm because, in the days when the theory of its action was not understood and all worm gears were very badly made, it gained its reputation for inherent inefficiency. Thus interest in worm gears as a practical means of transmitting power was not considered until the coming of the electric passenger lift, and other heavy machinery which it was desired to drive by a very much geared-down electric motor. Still it is a good many years ago since it was discovered that worms of a certain type could be used with success and economy, and it has been claimed frequently that some of the lift hoists, in particular, have shown an efficiency of nearly ninety per cent.

However this may be, it is an undoubted fact that the public demand for silent-running cars has brought the worm gear to its own. Of course, even now there is a large number of manufacturers in this country who have not yet had courage enough to give up the bevel, and scarcely anything besides the bevel axle is to be found on the Continent or in America, and this may, we think, be taken as an invitation to the British automobile engineer to lead the way for the rest of the world. It has often been said that to adopt the new form of final drive is an admission of weakness, in that it is a confession of the inability of the maker to produce bevel gears which will run quietly, but a study of cars of many different sources of origin, extending over a good many years, shows beyond all doubt that the attaining of a quiet bevel drive is very much a matter of accident. There are certain makers who have been extremely lucky in this respect; certain others who have been equally unlucky, but there appears to be no single manufacturer who has discovered any means whereby he can ensure absolute equality in this matter of sound between one gear and another, although they may be made by precisely similar methods and with the most scrupulous care.

A bevel pinion and crown wheel are slightly cheaper to manufacture than a worm and worm wheel, but, when the necessary plant for producing the latter form of drive has been laid down, it is questionable whether the bevel scores very much, even on mere works cost. Afterwards, when it comes to assembling and testing, the majority of bevel axles require some adjustment, and, if the adjustment is that of depth of engagement, to perform it usually necessitates opening up the axle and changing packing washers—trying first one and then another until a happy result is obtained—a most expensive process. With a worm, unless there is something radically wrong with it which should have been perceived in the workshop inspection, once it is assembled it will need no more attention and no adjustment, while a properly proportioned worm and wheel have over and over again proved capable of outlasting almost any other form of gearing.

If it is acknowledged that it is possible to obtain satisfactory efficiency with a worm gear, and further, that the worm gear can be produced at a cost no greater than that of the bevel gear (including the testing costs) and that the two forms are at least equally durable, then it is obvious that the natural quietness of the worm is sufficiently an advantage to make it the better of the two. These assumptions are all matters of proved fact, but they do not take into account one other thing of very considerable importance, this being that while there are many men who know much concerning the best methods of manufacture for bevel gearing, there are very few who can either make or design a satisfactory worm gear. Really the number of firms making back axle worm drives for automobile work is extremely small, but few individual manufacturers undertaking the task, while one or two who have been bold enough to attempt to make for themselves, have been disappointed by reason of the poor efficiency obtained. A bevel gear which is badly made, or rather which is not exceptionally well made, may be satisfactory on every point except that it will be noisy. On the other hand, the obvious fault in an inaccurately made worm transmission will be not noise, but loss of efficiency. There is good reason to believe that for all classes of both light and heavy road vehicle work the bevel will eventually disappear, and no doubt most motor manufacturers will themselves produce the substitute. It appears certain, however, that all early attempts to manufacture worm drives will be disappointing, because there are certain best methods which only considerable experience can show, and it is thought worth while to point out this perhaps rather obvious fact, because in certain instances it has certainly not been appreciated. If worm gear making is approached too light-heartedly, there is danger that cars may appear on the roads in quantities with inefficient axle gearing, and should this occur it is more than likely that the evolution of one of the best forms of gearing will receive another, and equally undeserved, set-back.

So far it has been assumed that back axle construction will continue along the lines of present day practice, but there is ample room for speculation here, because there is no doubt that "live" axles are disadvantageous by reason of their weight. Except for very heavy work the old form of chain transmission is probably gone never to return, but it is possible that some application of a single chain will eventually prove to be the best for use with very light, low-powered chassis. On the other hand, combined back axles and gearboxes are occasionally advocated, although European experiments have not so far proved anything to the advantage of this construction. Again the present method of obtaining changes of speed ratio will almost certainly give way sooner or later, before some better mechanism, and such an alteration might reasonably be expected to exercise a certain influence on the whole design of the car.

Notwithstanding such visionary considerations, however, the worm driven "live" back axle is likely to be standard practice all the world over before long, and it is, therefore, worthy of a little special study.

THE SPEED OF RECIPROCATING ENGINES.

A consideration of the limitations with additional reference to horse power formulæ.

By James Langmuir Napier.

(*Concluded from page 341*).

THE distribution of pressure on the piston and the inertia of the reciprocating parts may conveniently be considered together, so far as their effect on speed is concerned. In this connection, the first point to be studied is the form of the indicator diagram, and it is sufficiently accurate for the present purpose to assume that the actual diagram will follow closely the form of the theoretical diagram, after a suitable allowance has been made for the inevitable initial cooling of the exploded charge.

In Fig. VI., p is the atmospheric pressure, N is the compression pressure, and M is the initial pressure after explosion; a represents numerically the compression volume, and b the total volume

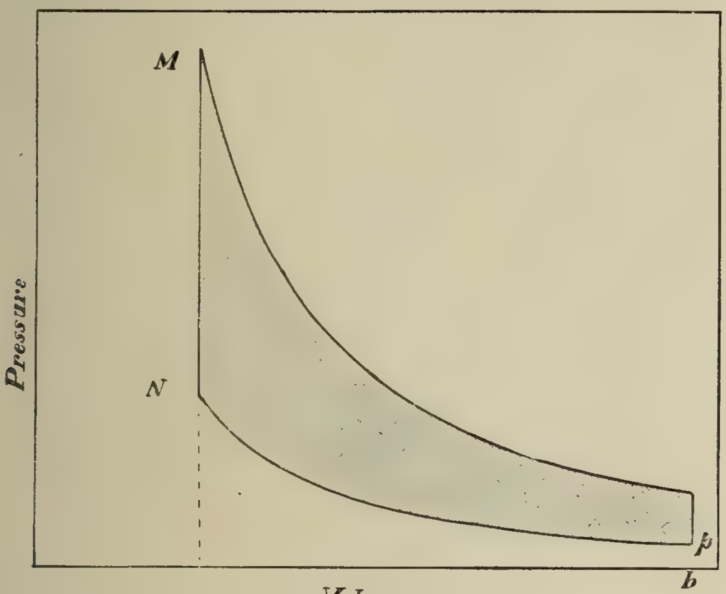


Fig. VI.

after explosion; $(b—a)$ is proportional to the length of stroke. It is convenient to call the ratio $\frac{M}{N}$, R; and the ratio $\frac{b}{a}$, n; the latter ratio representing what used to be known as the number of expansions.

If we take the law of adiabatic expansion to be $PV^{1.4}$ = constant, P and V being the pressure and volume of the gas at any instant, then we have

$$pb^{1.4} = Na^{1.4} = \text{constant,}$$

$$\text{and } N = p\left(\frac{b}{a}\right)^{1.4} = pn^{1.4}$$

and, since the curve starting at M has the same character, the effective pressure shown by the diagram at any volume V will be $(M — N)\ V^{-1.4}$, which may be written

$$p\,(R — 1)\left(\frac{n}{V}\right)^{1.4} \quad \dots \quad \dots \quad \dots \quad (1)$$

For ordinary purposes, a diagram such as Fig. VI. represents fairly enough the work done in a cylinder, but it misrepresents the distribution of pressure in point of time. In considering the case of the ordinary four-cylinder, four-cycle, internal combustion engine, and in analysing the effect of each working stroke, the diagram must be drawn as in Fig VII. The full explosion pressure, without deduction on account of compression, takes effect on the piston at the beginning of the stroke, and the back pressure due to compression in another cylinder takes effect at the end of the stroke, where it may or may not (in the diagram Fig. VII. it does) cause negative pressure.

Let $(b—a) = 2r$. Then $b = 2r + a = na$

$$2r = a\,(n — 1)$$

$$\frac{2r}{n—1} = a$$

$$\text{and } \frac{2nr}{n—1} = b$$

The effective pressure shown by the diagram Fig. VII. at any volume V will be

$$M_1\,V^{-1.4} — N_1\,(c — V)^{-1.4} \quad \dots \quad \dots \quad (2)$$

where $N_1 = pb^{1.4} = p\left(\frac{2nr}{n—1}\right)^{1.4}$; $M_1 = RN_1$ and

$$c = a + b = \frac{2r\,(n + 1)}{n — 1}$$

At any distance s from the beginning of the stroke

$$V = \frac{2r}{n — 1} + s \text{ and } c — V = \frac{2nr}{n—1} — s$$

and if θ be the angle through which the crank has moved form the dead centre at the beginning of the stroke, then $s = r\,(1 — \cos\theta)$, and

$$V = \frac{2r}{n—1} + r\,(1 — \cos\theta) \text{ and } c — V = \frac{2nr}{n—1} — r\,(1 — \cos\theta)$$

Substituting these values in (2), we get

Pressure per sq. in. =

$$\frac{pR\,(2n)^{1.4}}{\{2 + (n—1)\,(1—\cos\theta)\}^{1.4}} — \frac{p\,(2n)^{1.4}}{\{2n—(n—1)\,(1 \quad \cos\theta)\}^{1.4}} \dots \quad (3)$$

The general expression for the area of the diagram up to the point in the stroke where the total volume is V, is

$$C — 2.5\left\{\frac{M_1}{\ \ ^{.4}} + \frac{N_1}{(c—V)^{.4}}\right\}$$

which is the integral of (2). Substituting the values of V and $(c—V)$, and integrating between the limits a and s, we get the following expression for the work done per square inch of piston area while the crank moves through $\theta°$ from the dead centre:

$$W = \left\{2.5\ pr\ \frac{(2n)^{1.4}}{n—1}\right\}\left\{\frac{R}{2^{.4}} — \frac{R}{\{2 + (n—1)\,(1—\cos\theta)\}^{.4}} — \right.$$

$$\left.\frac{1}{\{2n—(n \quad 1)\,(1 \quad \cos\theta)\}^{.4}} + \frac{1}{(2n)^{.4}}\right\} \quad \dots \quad \dots \quad (4)$$

When θ is 180° –that is, at the end of the stroke—this becomes

$$\left\{2.5\ pr\ \frac{(2n)^{1.4}}{n—1}\right\}\left\{\frac{(R—1)\,(n^{4}—1)}{(2n)^{4}}\right\}$$

$$= (2.5)\ pr\,\frac{2n}{n—1}\,(R—1)\,(n^{4}—1)$$

and, dividing by $2r$, we get

$$\text{Mean pressure per sq. in.} = (2.5)\ p\,(R—1)\,\frac{n\,(n^{.4}—1)}{n—1} \dots (5)$$

Not to stray too far from my subject, I shall only glance here at the effect of compression on mean pressure as indicated in equation (5). Fig. VIII. shows this graphically, and it will be noted that within ordinary limits neither the mean pressure nor the heat efficiency in a cylinder is affected to any enormous extent.

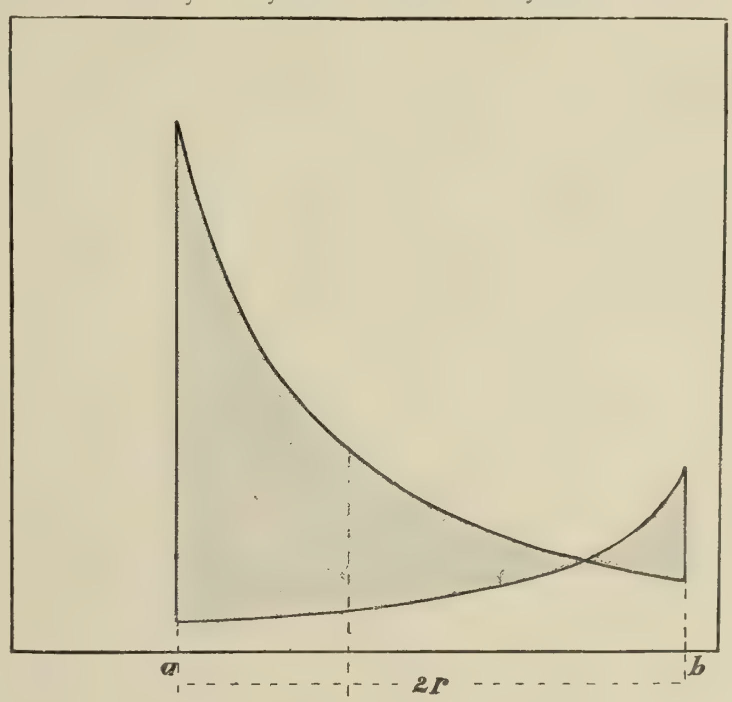


Fig. VII.

These curves also assume that no external cooling takes place, and when it is remembered that explosion temperature increases with compression, and that loss of heat by external cooling is proportional to difference of temperature, it will be apparent that a reduction must be expected even in the moderate increases

shown. It is probable that, if loss of heat be taken into consideration, a point of maximum useful compression would be reached. Further investigation of this subject is outside of my present limits. It is sufficient to indicate here that it is not surprising that difference of compression ratio should be found practically to have small influence on the expectation of horse-power.

The formulæ given for pressure and power calculated from such a diagram as Fig. VII. are too complicated for extensive

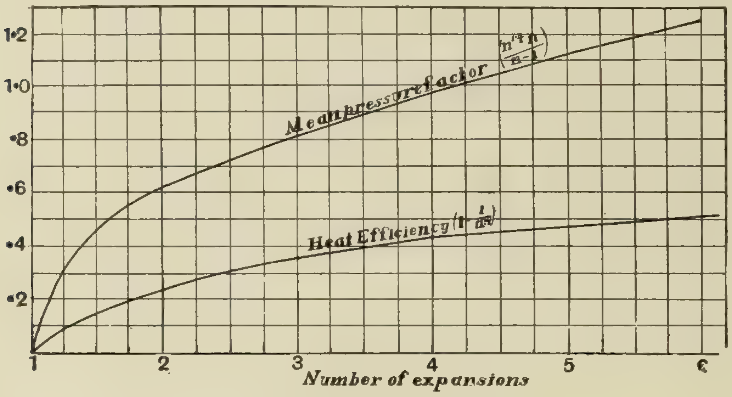


Fig. VIII.

use. I have used one of them in the construction of Fig. IX., which shows the pressure on the piston and the tangential force on the crank pin due to the piston of Fig. VII. throughout half a revolution of the crank. I also show a curve of the vertical inertia forces (so far as they are in excess of the pressure on the piston) due to reciprocating masses. The engine is supposed to have four cylinders, 4in. dia. × 6in. stroke. R is 3 and n is 4. The total reciprocating mass is taken as unity (equal to about 8 lbs. per cylinder), and its effect is assumed to be due to a uniform velocity of 1,200 revolutions per minute. As I understand Mr. Burl's graphic illustration of his formula for the limit of engine speed, this engine has already exceeded the limit, and yet the speed assumed cannot seriously be considered excessive. The weak point of Mr. Burl's formula, as applied by him, is that it takes no account of the number of cylinders and cranks nor of their arrangement. In an ordinary four-cylinder engine with cylinders and cranks in one plane, the reciprocating mass to be accelerated at the end of each stroke is the mass of four pistons and their attachments, while the explosion takes place in only one cylinder.

Assuming that equations (3) and (4) are too elaborate for the calculation of many points in a curve, it is permissible to adopt the graphic method and to ascertain the necessary quantities by actual measurement from such a diagram as Fig. VII. Another course—which is the one I have adopted—is to imagine an approximate diagram of simple form such as is shown in Fig. X.

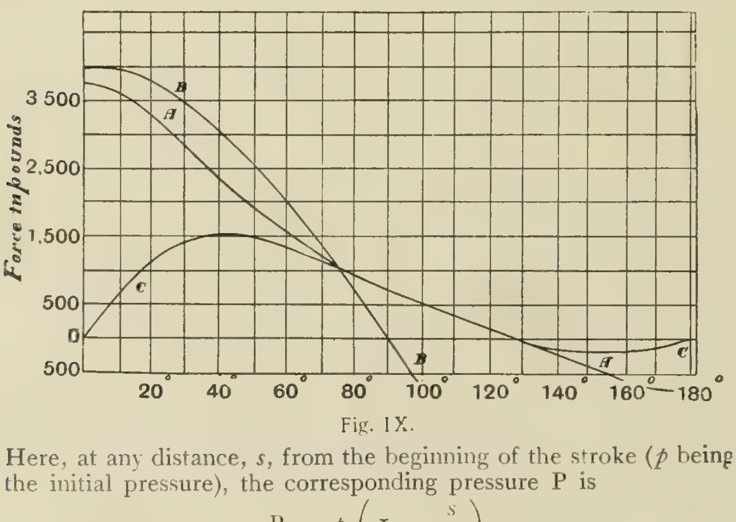


Fig. IX.

Here, at any distance, s, from the beginning of the stroke (p being the initial pressure), the corresponding pressure P is

$$P = p\left(1 - \frac{s}{2r}\right)$$

which, if we put $r\ (1 - \cos\theta)$ for s, becomes

$$P = p\left(\frac{1 + \cos\theta}{2}\right) \quad \ldots \quad \ldots \quad \ldots \quad (6)$$

and the work done from the beginning of the stroke up to the same point is

$$W = pr\left(\frac{3 - 2\cos\theta - \cos^2\theta}{4}\right) \quad \ldots \quad \ldots \quad (7)$$

In Equations (6) and (7) I consider p to represent pressure × area, and the following arbitrary assumptions are made in the diagrams in which these equations are used:

$p = 1980.$
$r = 0.25.$
$2r$ = stroke = 0.5.
M = total reciprocating mass = $\frac{4}{3}$.
W = 495 foot-pounds per stroke.

The statement that the mass of reciprocating parts does not materially affect speed is not intended to apply to engines of the type of certain steam pumps which are independent of a crank and flywheel. The conditions which follow are not capable of exact reproduction in practice, but for purposes of comparison we may imagine an engine, with an indicator diagram as shown in Fig. X., working against a constant load equal to the mean pressure. The piston is then subject to an accelerating force equal to the difference between the pressure at any point and the mean pressure.

At any distance s from the beginning of the stroke the accelerating force is $p\left(1 - \frac{s}{2r}\right) - \frac{p}{2} = \frac{p}{2}\left(1 - \frac{s}{r}\right)$ and the acceleration is $\frac{p}{2M}\left(1 - \frac{s}{r}\right)$, M being the reciprocating mass.

If s be space; v, veolcity; a, acceleration; and t, time, we should then have

$$s = r\left\{1 - \cos\left(t\sqrt{\frac{p}{2rM}}\right)\right\}$$

$$v = \frac{ds}{dt} = \sqrt{\frac{p}{2rM}}\ r\sin\left(t\sqrt{\frac{p}{2rM}}\right)$$

$$a = \frac{dv}{dt} = \frac{p}{2M}\cos\left(t\sqrt{\frac{p}{2rM}}\right)$$

which value of a satisfies the equation $a = \frac{p}{2M}\left(1 - \frac{s}{r}\right)$, and it is obvious that the motion of the piston with respect to

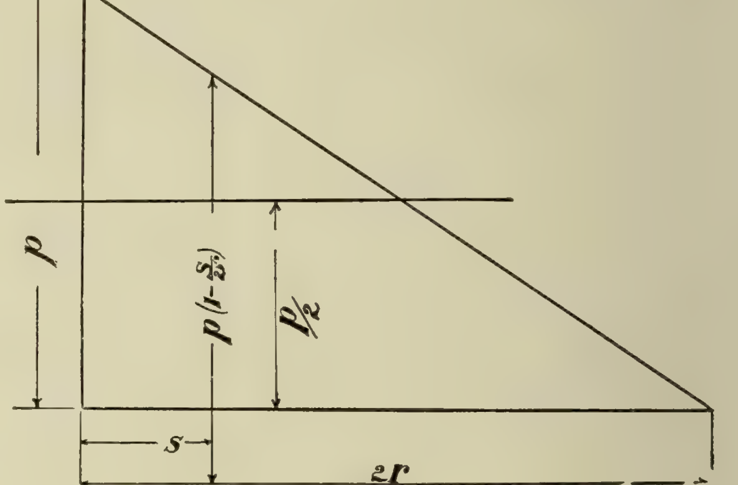


Fig. X.

time is a simple harmonic motion, similar to the motion of a weight suspended by a spring vibrating vertically.

At the end of the stroke $s = 2r$, therefore,

$$2r = r\left\{1 - \cos\left(t\sqrt{\frac{p}{2rM}}\right)\right\}$$

$$2 = 1 - \cos\left(t\sqrt{\frac{p}{2rM}}\right)$$

$$-1 = \cos\left(t\sqrt{\frac{p}{2rM}}\right)$$

The angle whose cosine is (—1) is π; therefore the time occupied by one stroke would be

$$t = \frac{\pi}{\sqrt{\frac{p}{2rM}}} \quad \ldots \quad \ldots \quad \ldots \quad \ldots \quad (8)$$

all the terms of which are constant for a given engine.

The speed of this imaginary engine would be strictly conditional by the mass of the reciprocating parts, and the addition of a crank and flywheel to the engine would be entirely superfluous; for if the crank moved through an angle θ while the piston moved through the space s, then $s = r\ (1 - \cos\theta) = r\left\{1 - \cos\left(t\sqrt{\frac{p}{2rM}}\right)\right\}$, and therefore $\theta = t\sqrt{\frac{p}{2rM}}$, a constant function of t. The velocity of the flywheel (consistently with the simple harmonic motion of the piston) would be uniform,

and its mass would therefore be a matter of indifference.

Incidentally it may be remarked that if the cylinder of this imaginary engine, instead of being firmly fixed, were itself supported by a spring, the condition of forced vibration would be set up and the mass of the cylinder itself and the strength of its supporting spring would then become questions of moment. As will be seen later, a condition analogous to this exists in the ordinary motor car.

So far I have assumed that resistance is independent of speed, which is not the case in practice. The resistance $\frac{p}{2}$ might be reached at a higher or lower speed than that corresponding to Equation (8), with the result that the flywheel comes into play.

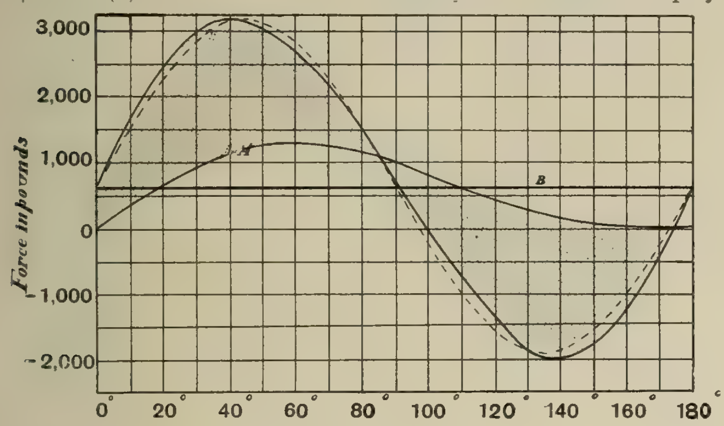


Fig. XI.

Its speed is no longer uniform, and its mass becomes of importance. Equation (8) gives us, in fact, a measure of the speed at which the transference of energy to and from the flywheel is least; and for those who insist on looking at the matter from that point of view it supplies a measure of "pleasant horsepower." If the time of one stroke in seconds be $\frac{\pi}{\sqrt{\frac{p}{2rM}}}$, then the piston speed in feet per minute is

$$\frac{60 \times 2r\sqrt{\frac{p}{2rM}}}{\pi}$$

$$= \frac{120}{\pi}\sqrt{\frac{p}{2}\,\frac{r}{M}}$$

which, with a different constant, is Mr. Burl's formula for maximum permissible piston speed. It will be noted, from the path by which I have reached Mr. Burl's result, that the formula ignores the mass of the flywheel and assumes a form of resistance which is not the common one. I shall now show the effect of assuming a flywheel of definite mass and a resistance consisting of a constant turning moment. I assume, as in the last case, the indicator diagram of Fig. X., four single-acting cylinders giving one impulse per stroke, four cranks in the same plane, and a flywheel of mass N supposed to be concentrated in the path of the crank pins. I neglect the obliquity of the connecting rod.

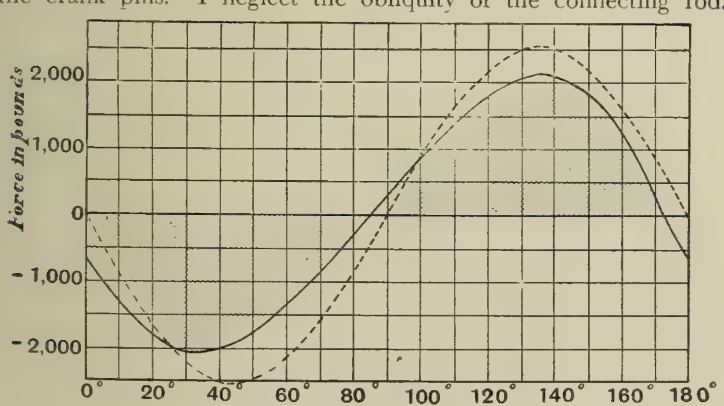


Fig. XII.

The load on the engine is supposed to consist of a turning moment sensibly constant within the limits of speed encountered. The tangential force at the radius of the crank pin due to the load is therefore $\frac{p}{\pi}$ = 632 lbs. The velocities are calculated from an assumed constant lineal velocity of crank pin at the beginning of the stroke of 33·57 feet per second, corresponding to a mean velocity of about 1,200 revolutions per minute when the crank radius is three inches. The constants used are therefore

p = area of one cylinder × initial pressure per sq. in. = 1980.
r = length of crank = 0.25ft.
M = total reciprocating mass = $\frac{4}{3}$.
N = equivalent revolving mass at radius r = 6.
C = lineal velocity of crank pin at beginning of stroke = 33·57.
$\frac{p}{\pi}$ = tangential resistance at radius r = 632.

The following are variable:
θ = angle passed through by crank from beginning of stroke.
S = lineal distance of crank pin from beginning of stroke.
s = lineal distance of piston from beginning of stroke.
V = lineal velocity of crank pin.
v = lineal velocity of piston.
A = acceleration of crankpin.
a = acceleration of piston.
t = time in seconds corresponding to angle θ.

Certain relations obtain between these variables:

$S = \theta r.$	$s = r\,(1 - \cos\theta).$
$V = r\dfrac{d\theta}{dt}$	$v = \dfrac{ds}{dt} = r\sin\theta\dfrac{d\theta}{dt} = V\sin\theta.$
$A = r\dfrac{d^2\theta}{dt^2}$	$a = \dfrac{d^2s}{dt^2} = r\cos\theta\left(\dfrac{d\theta}{dt}\right)^2 + r\sin\theta\dfrac{d^2\theta}{dt^2}$
	$= \dfrac{V^2}{r}\cos\theta + A\sin\theta.$

$$\text{Also } \frac{d}{d\theta}V^2 = 2V\frac{dV}{d\theta}$$
$$= 2r\frac{d\theta}{dt}\frac{dV}{d\theta}$$
$$= 2r\frac{dV}{dt}$$
$$= 2Ar$$

When the crank is at any angle θ, two forces act on the piston:

P = the pressure of the gas = $p\left(\frac{1+\cos\theta}{2}\right)$ from Eq. (6).

— Q = the force accelerating M = —Ma.

The tangential component of these forces is (P—Q) sin θ, which

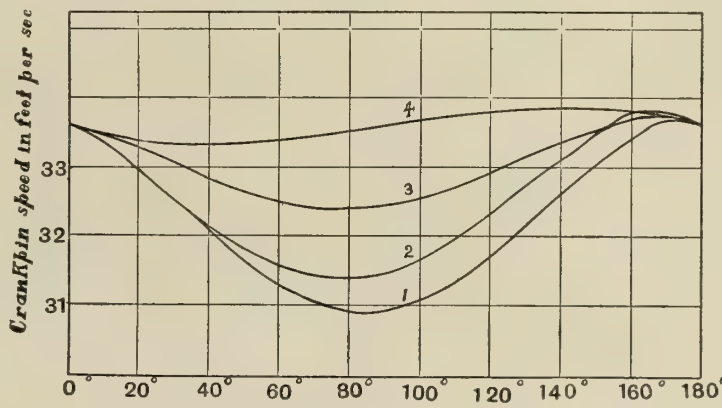


Fig. XIII.

is opposed by the force accelerating the flywheel and by the constant resistance. Consequently we have

$$(P - Q)\sin\theta = AN + \frac{p}{\pi}$$

Inserting the values of P and Q, this becomes

$$p\sin\theta\left(\frac{1+\cos\theta}{2}\right) - M\sin\theta\left(\frac{V^2}{r}\cos\theta + A\sin\theta\right) = AN + \frac{p}{\pi}$$

$$p\sin\theta\left(\frac{1+\cos\theta}{2}\right) - \frac{MV^2}{r}\sin\theta\cos\theta - \frac{p}{\pi} = A\,(N + M\sin^2\theta)$$

and

$$A = \frac{\frac{p\sin\theta}{2} + \frac{p\sin\theta\cos\theta}{2} - \frac{MV^2}{r}\sin\theta\cos\theta - \frac{p}{\pi}}{N + M\sin^2\theta} \qquad \ldots(9)$$

The work done by the expansion of the gas behind the piston while the crank moves through angle θ is, as given in Eq. (7), $pr\left(\frac{3 - 2\cos\theta - \cos^2\theta}{4}\right)$.

The work done in opposing the constant resistance is $\frac{pr\,\theta}{\pi}$

The work done accelerating M is $\frac{Mv^2}{2} = \frac{MV^2\sin^2\theta}{2}$

The work done accelerating the flywheel is $\frac{NC^2}{2} \quad \frac{NV^2}{2}$

We have therefore:

$$\frac{MV^2 \sin^2 \theta}{2} + \frac{pr\theta}{\pi} = \frac{NC^2}{2} - \frac{NV^2}{2} + pr\left(\frac{3-2\cos\theta-\cos^2\theta}{4}\right)$$

$$V^2\left(\frac{N+M\sin^2\theta}{2}\right) = \frac{NC^2}{2} - \frac{pr\theta}{\pi} + pr\left(\frac{3-2\cos\theta-\cos^2\theta}{4}\right)$$

and

$$V^2 = \frac{NC^2 - 2pr\frac{\theta}{\pi} + \frac{pr}{2}(3-2\cos\theta-\cos^2\theta)}{N + M\sin^2\theta} \quad \ldots \quad \ldots \quad (10)$$

the differential coefficient of which, with respect to θ, will be found to be $2r$ times the value of A, as given in Equation (9).

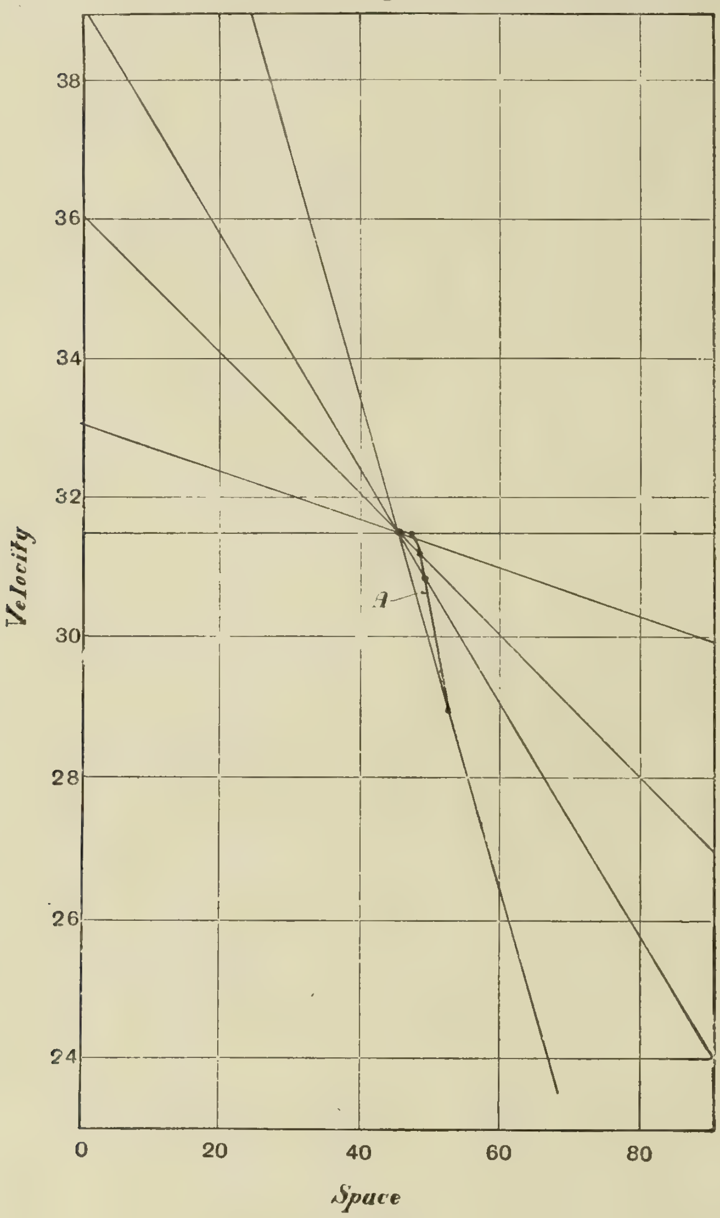


Fig. XIV.

In Fig. XI., the line A shows the tangential force exerted on the crank pin due to pressure on the piston. This may be compared with the corresponding line in Fig. IX. The line B shows the constant tangential resistance due to the load, and the ordinates within the shaded areas represent the unbalanced tangential force AN, as calculated from Equation (9), at any angle of crank. In Fig. XII. these forces are shown referred to a straight line, the forces above the line being positive—that is, tending to increase the speed of revolution and *vice versa*.

Figs. XI. and XII. are typical of all four-cylinder engines with cranks in one plane. To an extent depending in varying quantities on the reciprocating mass, the moment of inertia of the flywheel, and the load, the engine is driven by the flywheel at the beginning of the stroke. In the case under consideration, the engine is driven by the flywheel until the crank has advanced about 87°.

The magnitude of the unbalanced tangential forces, due to the reciprocating mass M, is indicated by ordinates measured above and below the line B (Fig. XI.). It will be noted, however, that an increased load, by raising the line A (Fig. XI.) proportionally throughout, tends to obliterate the unbalanced force due to M. Conversely, a light load increases the effect of those unbalanced forces. In Fig. XII. the dotted lines indicate approximately the extent the unbalanced forces would attain if the engine were running without load at the same speed as in the loaded conditions to which the shaded areas apply. The vibration due to this cause may be observed in any four-cylinder car with the engine running light, but probably it is not generally recognised that the vibration is principally one about the axis of the crankshaft, tending to rock the car on the road springs.

In the six-cylinder engine with cranks at 120°, and in the eight-cylinder engine with cranks at right angles, the unbalanced tangential forces due to the mass of reciprocating parts exist only in their secondary remnants due to the shortness of the connecting rods. Such engines fitted with connecting rods of infinite length or their equivalent, and neglecting such practical considerations as play in bearings and twisting of crankshafts, can be arranged so as to be theoretically in perfect balance—longitudinally, radially, and tangentially—at any speed and whatever the mass of their reciprocating parts, so long as the engines are running light. Running under load, however, the influence of uneven turning moment, as indicated by curve A (Fig. XI.) or C (Fig. IX.), would become marked, and conditions of speed and load are conceivable in which the perfectly balanced six-cylinder engine might appear inferior to the relatively unbalanced four.

The force AN, tending to accelerate the flywheel in one direction and the car in another, is shown by the ordinates of the shaded areas in Figs. XI. and XII., and the extent of these areas indicates the amount of energy transferred to and from the flywheel, but these diagrams fail to indicate adequately the variation of velocity of the flywheel due to its finite mass. It is, in fact, essential to the construction of the diagrams that, if the velocity at the first point of no acceleration be assumed, the area of the shaded portions is unaltered, whether that velocity be constant throughout the stroke, or whatever diminution of velocity may have happened since the stroke began. Such variation of velocity is indicated only by alteration in the position of the peaks of the diagram, and that not to a very noticeable extent. Thus in Fig. XI. the dotted line represents the position which the shaded portions of the diagram would assume on the supposition that the velocity of revolution is constant at the lower limit of speed reached. Constant speed assumes that N is infinite, the difference between which and N = 6 is not well marked on the diagram. The shaded area to the left of the point of no acceleration is manifestly equal to the area enclosed by the dotted line. That this should be so is clear from consideration of the physical facts involved as well as from the form of the expression used in Equation (9).

From that equation

$$AN = \frac{p\sin\theta}{2} + \frac{p\sin\theta\cos\theta}{2} - \frac{MV^2}{r}\sin\theta\cos\theta - MA\sin^2\theta - \frac{p}{\pi}$$

The work done in accelerating the flywheel, represented by the shaded area in Fig. XI., is between zero and θ.

$$r\int AN\,d\theta,$$

which is $pr\left(\frac{3-2\cos\theta-\cos^2\theta}{4}\right) - \frac{pr\,\theta}{\pi} - \int MV^2\sin\theta\cos\theta\,d\theta$

$$- \int MA\sin^2\theta\,r\,d\theta$$

In this expression, since θ is by hypothesis the angle at which A is O, the term involving A vanishes; and since when A is O, V must be constant, the term involving V may be integrated on that basis, and therefore *when A is nothing*

$$r\int AN\,d\theta = pr\left(\frac{3-2\cos\theta-\cos^2\theta}{4}\right) - \frac{pr\,\theta}{\pi} - \frac{MV^2}{2}\sin^2\theta$$

whether we consider V as a variable function of θ, or as a constant having the value that the variable would reach when A = O.

The reason for selecting the first point at which A is O, and at which V is a minimum, is that we are thus able to compare individual areas of the same sign. The statement made is equally true algebraically of the second point at which A is O and V is a maximum, with the difference that the comparison would then be made between the algebraic sums of two sets of areas of contrary signs, which under the circumstances is not desired.

In order to exhibit more clearly the effect of mass on speed, the ordinates of Fig. 13 have been calculated from Eq. (10). Fig. 13, curve No. 1, shows the variations in crank-pin velocity that would occur during half a revolution of the crankshaft of an engine such as we have been assuming. Curve No. 2, which shows an apparently smoother-running engine, assumes that the stroke has been doubled—that is, increased to twelve inches. Note that curve No. 2 assumes automatically that the moment of inertia of the flywheel has at the same time been quadrupled, since the N of Eq. (10) assumes the mass of the flywheel to be concentrated at the radius of the crank pin. Since the velocity of crank pin is the basis of the diagram, it follows that in curve No. 2 the period of one stroke has been doubled. Although,

therefore, the variation of crank-pin velocity has been reduced with an increased stroke, the reduction is really due to the increased mass of the flywheel; and whether such an engine would really run more smoothly than the engine of curve No. 1 would depend on the moment of inertia of the car about the axis of the crankshaft and its natural period of vibration on the road springs. Curve No. 3 (Fig. XIII.) assumes that M, the reciprocating mass, has been halved, and curve No. 4 that it has been reduced to one-sixth of its original amount. It will be noted that curve No. 4 has reached the lower limit of variation of velocity. If the reciprocating masses could be further reduced, or, what comes to the same thing, if they were better balanced, the variation of flywheel velocity would be not reduced, but increased, and the resulting vibration would possibly by of a more unpleasant character.

So far as we have seen it does not appear that, within the limits imposed by the strength of materials, mass whether reciprocating or revolving, where both are employed, has any important influence on speed; nor does it appear that of the two, reciprocating mass is the more important.

It is certainly the case that it will be found in practice that any mechanism involving reciprocating and revolving parts may be run at a speed at which slack bearings, bad lubrication, want of balance, or some unsuspected weakness of material may cause unpleasant manifestations, but it does not follow that a similar machine cannot be designed without such imperfections. The true limits of engine speed are more likely to be found in the considerations outlined in the first part of this article, the lower limit of speed being fixed by external cooling of the cylinder, and the higher by the difficulty of getting the charge in and out of the cylinder.

Unbalanced mass and any other reason for variation in the speed of revolution as indicated in Fig. XIII., have a certain positive influence on speed which may be briefly described:

$$V = \frac{dS}{dt}\,;\ \frac{1}{V} = \frac{dt}{dS}\ \therefore\ t = \int \frac{1}{V}\,dS$$

when V is a function of S, and the average speed of revolution is, of course, $\frac{S}{t}$, S being the whole distance and t the whole time.

I have not been able to find any convenient formula for t as derived from V in Eq. 10. We may, however, consider the curves of Fig. XIII. to consist of straight lines for approximate calculation. We may suppose, for instance, that curve No. 1, starting at 33.5ft. per sec. at 0°, proceeds straight to 31ft. per sec. at 90° and returns straight to 33.5ft. at 180°.

Fig. XIV. shows four such straight lines, all having the same mean velocity referred to space, but varying respectively between the limits of 33 and 30, 36 and 27, 39 and 24, and 46.5 and 16.5. For the four lines we get

$V =$	$33-\frac{S}{30}$	$36-\frac{S}{10}$	$39-\frac{S}{6}$	$46.5-\frac{S}{3}$
$\int\frac{1}{V}\,dS =$	$C-30\log\left(33-\frac{S}{30}\right)$	$C-10\log\left(36-\frac{S}{10}\right)$	$C-6\log\left(39-\frac{S}{6}\right)$	$C-3\log\left(46.5-\frac{S}{3}\right)$
$t =$	2.860	2.878	2.912	3.108
$\frac{S}{t} =$	31.47	31.27	30.90	28.96

The points of true mean velocity referred to time, S/t, are shown on Fig. XIV. by dots connected by a thick line A. It will be noted that the divergence from the apparent mean velocity, 31.5, does not become considerable until the variation of velocity is much greater than need occur in practice. The load has been supposed to be that due to a constant velocity of 31.5, but if, as is probably the case in most circumstances, the load, or any part of it, increases in proportion to the square of the velocity, the variation would then have an additional effect by reason of the mean load increasing with an increase of variation in velocity. The practical importance of these points will be measured by the degree of refinement aimed at.

THE THOMAS TRANSMISSION.

A Combined Electrical and Mechanical Change Speed Mechanism.

It is now rather more than a year since the Thomas transmission became known, and it is only recently that any extensive tests have been carried out with it. Seeing, however, that a converted lorry carrying a very heavy load has recently completed a 2,000-mile R.A.C. trial in a satisfactory manner, it appears that the undoubted theoretical advantages of the system are not offset by serious practical disadvantages. The fundamental idea of the transmission is very ingenious, but the operation is not very easy to explain. It may, however, best be considered by first neglecting entirely the electrical operation of the gear. Inside the flywheel of the engine there is an epicyclic gear, duplex planet pinions being carried round by the flywheel. These planets mesh with a pair of sun wheels, one of which is attached to a shaft running right through the whole apparatus to the propeller shaft, while the other sun wheel is on a sleeve external to the shaft. Obviously, then, if by some means the two sun wheels are clutched together we obtain a direct drive from engine to back axle. On the other hand, if the wheel on the sleeve is held stationary, the large planets run round on it, and the sun wheel, on the driving shaft will revolve at a reduced speed. Thus from this epicyclic train it is possible to

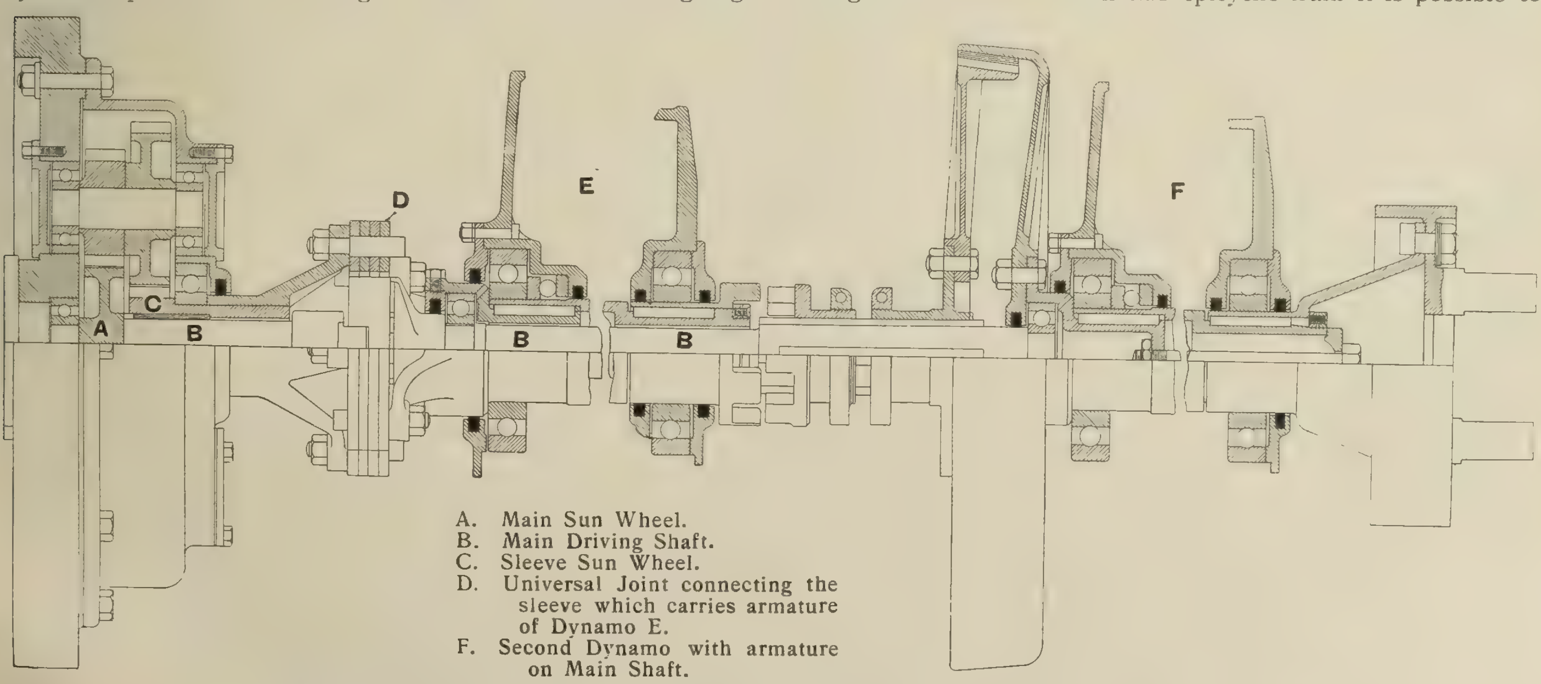


Fig. I. Diagramatic Section of Thomas Transmission with all dynamo details omitted for the sake of clearness.

obtain two mechanical speed ratios.

Now, let it be assumed that instead of locking the wheel on the sleeve, this is restrained by some braking action, then the sleeve will slip and, according to the degree of force applied, so different speed ratios will be obtainable varying from the direct drive at one end to the definite reduction when the sun wheel is locked. Similarly, if instead of being braked the smaller sun wheel was actually revolved by some external means, other gear ratios would be obtained until such a

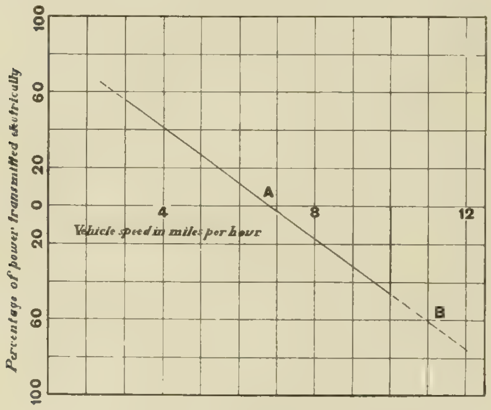


Fig. II. Diagram showing the two proportions of power transmitted electrically.

speed was reached that there was direct transmission. In practice the ratio obtained when the sleeve wheel is locked comes about the middle of the whole range of speeds obtainable from the transmission.

The real essence of the invention lies in the method whereby the power required to restrain or accelerate this sleeve sun wheel is obtained without waste, or rather with very much less waste than it could be by mechanical means. This is done by attaching to the same sleeve as the sun wheel the armature of a dynamo of which the field magnets are fixed. Supposing, therefore, that the engine is driving the flywheel and the armature is quite free to revolve, then obviously, no actual driving will take place, but when a current is taken from the dynamo this at once proceeds to absorb power from the sleeve sun wheel, thereby slowing it down, and, as soon as it slows down, the main driving shaft with its sun wheel will commence to revolve. The greater the current taken from the dynamo the greater the retarding action on the sun wheel, and the faster the driving shaft will revolve.

For the time being (neglecting the other side of the locked position of the sleeved sun wheel), the current derived from the dynamo, while the latter is being used for the purpose of braking the sleeve wheel, is led to another similar dynamo also with its armature on the driving shaft, but situated nearer to the propeller shaft end of the gear. The current derived from the first dynamo is then utilised to drive the second machine as a motor, so the main driving shaft receives some torque from the engine through the epicyclic gear, and additional torque from the armature of the second dynamo. The greater the current taken, that is to say, the stronger the braking action on the sleeve sun wheel, the more powerful is the torque exerted by the second dynamo, so, according to the particular speed at which the various parts are running, a certain portion of the power is always being transmitted mechanically, and another portion electrically.

Proceeding to the next stage, that is, when the sleeve sun wheel is not merely locked, but is actually driven, this is done by using the rearmost machine as a dynamo, and the front one as the motor, when the operations are simply reversed, electrical power being taken from the driving shaft at the rear end and transmitted forwards to the sleeve sun wheel.

In Fig. I. the general mechanical details are shown, and it will be noticed that there is a dog clutch capable of locking the sun wheel sleeve to the main shaft to give the top speed, while there is a friction clutch behind this, and the latter is controlled in the usual way. The curves in Figs. II. and III. are explained fully by the inscriptions beneath them, and the diagrams of connections also explain the electric circuits whereby the different speeds are obtained. The only additional explanation needed being perhaps for the last two of these and concerning the battery, which is used for starting purposes, and is, of course, charged up when running on the top speed or downhill. It is not proposed to give any detailed description of the switch gear whereby the control is obtained, but it may be remarked that the number of connections are small, there being only four leads from each dynamo. The controller consists of a drum, mounted on a spindle which is rotated by the controller lever through a quadrant and spur wheel. At the periphery of the drum projections are fitted which, on the rotation of the drum, can be made to press outwards the ends of special fingers which are pivoted at the other ends. On the moving end of each finger is mounted a block contact, which meshes with a brush contact and sparking tip. When the contacts are engaged the brush spreads and the block is so fitted to the finger that it can adjust itself to give perfect contact. The block, brush and sparking tip are all independent units and can be replaced in a few seconds if necessary. The sparking tip takes all the sparking when the break occurs and, as it is an independent unit, can be replaced very quickly. The whole controller is oil immersed, which is in every way beneficial, because it reduces any possibility of sparking to a minimum,

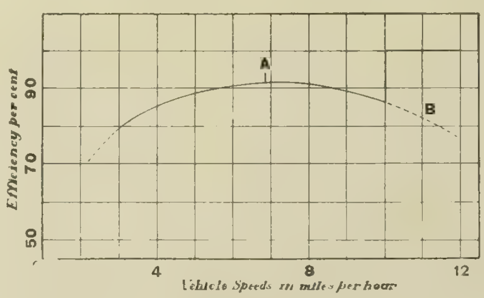


Fig. III. Calculated efficiency curve at constant engine speed; A - B usual running speeds.

increases the insulation of the whole controller, keeps the resistances from getting too hot and lubricates all the moving parts. The drum and the whole of the working parts of the controller are bolted to the lid of the tank which contains the oil, and the lid is hinged so that the whole controller can be swung out with it and thoroughly inspected in a few seconds.

It is wonderful how nearly constant it is possible to keep the speed of the engine on a long hill with a varying gradient. With the test lorry for example, on a really steep hill the range used is principally from No. 3 to No. 6 speed, and the handling of the lever is no more difficult than the opening and closing of a throttle or the variation of the ignition point. The instant the engine shows the slightest signs of slowing the lever is moved down one notch, and as soon as the engine begins to increase speed up goes the gear again so easily that the handling becomes practically automatic.

Needless to say, a good many advantages are claimed for this gear, particularly for lorry and omnibus work, but the drawbacks in both these cases are the cost and the weight, although neither of these are so much above that of an ordinary gearbox as might be imagined. The transmission system, however, seems to be eminently suitable for that particular class of locomotion which the internal combustion engine has hitherto been unable to exploit on account of transmission difficulties. This is, of course, light railway work. On rails, weight is a matter of very small importance, the great trouble being the need for an enormous reserve of power for

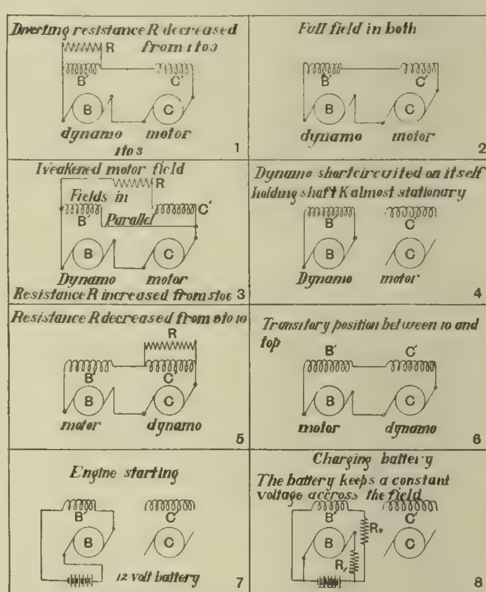


Fig. III.—Control Connections.

1. For speeds 1, 2, & 3. 2. For speed 4. 3. For speeds 5 & 6. 4. For speed 7. 5. For speeds 8, 9, & 10. 6. Intermediate between 10 and top. 7. Battery engine starting. 8. Charging battery.

starting purposes. With such a car as the Thomas it would be quite easy to start even a heavy train, and the control would not be dissimilar to that of a car with electric power. Heavy gradients could be tackled and, for light railway work in particular, the internal combustion engine has tremendous advantages over the usual steam locomotive. That this has already been realised is shown by the fact that the Leyland Co. are at present building a railway car fitted with one of their own engines and a Thomas transmission. We hope to be able to describe this machine in detail in the near future, and, in conclusion, would say that while the Thomas transmission does not appear to be likely to displace the gearbox for most types of motor vehicles at present in common use, it might quite possibly be found to be a means for simplifying the handling of very heavy wagons or tractors, or for railway work.

AUTOMATIC AIR VALVES FOR CARBURETTORS.

A Description of Eight Different Typical Designs.

By R. Waring Brown.

THIS article may prove the more interesting to the designer in view of the great amount of discussion which is taking place at the present time on the subject of carburettors, and the attempts of many experts to state a definite law connecting quantities of air and petrol. The divergence of their opinions, however, leaves us in as great a quandary as ever. The intention of the writer is to outline the principle and design of the auxiliary air valve, and to draw attention to the necessity for improvement to this detail of carburation if its use is to be continued. Broadly speaking there are two kinds of carburettors, which may be termed the "constant-vacuum" and "variable vacuum" types. Undoubtedly at the moment the "constant vacuum" type is the most popular. After lengthy experiments the writer is of the opinion that the ideal carburettor is that constructed to maintain a constant depression in the carburettor, and for this reason that necessary adjunct to this type, the auxiliary air valve, is brought forward as being worthy of greater consideration on the part of the designer. As the hand-operated air valve can only approximate to its work the automatic air valve alone will be considered. The term "constant vacuum" must be taken in a broad sense, as of course, it is apparent that there must be a slight variation in pressure above and below normal to operate the air regulator.

The air regulator, or auxiliary air valve, frequently mis-called an extra air valve, is actuated by the engine suction, and the necessity for this is fairly obvious on consideration. The mixture does not enter the engine steadily, but in a state of oscillation owing to the uneven suction of the engine. These oscillations have a diversifying effect on the petrol and air entering the carburettor; the air is prevented from following up the rapid movements of the pistons at high speeds of the engine by its elasticity, in fact it has been proved by actual experiment that the mixture taken in is sometimes as much as 50 per cent. less than the piston displacement owing to the inertia and elasticity of the air. Petrol is non-elastic and comparatively heavy, so each engine pulsation adds to its momentum until it issues from the jet in a steady stream. The air however, tends to decrease proportionately as the engine speed increases, resulting in too rich a mixture unless some arrangement is made for regulating it. The generally accepted method is to provide an additional and adjustable air opening to admit air between the throttle and choke tube, controlled automatically by a spring-loaded valve. The exact purpose of the valve is to vary and regulate the resistance to the air entering the carburettor by increasing or decreasing the area of the intake, thus governing the pressure forcing petrol from the jet; as the variation of air opening will increase or decrease the difference in pressure between the inside and outside of the carburettor.

It may be as well to note the exact requirements of an automatic air valve before passing to illustrations of the various prevalent designs and these are:—

1. The valve must maintain a constant negative pressure in the carburettor, and consequently must be capable of admitting the greatest quantity of air required at the highest possible engine speed.
2. It must also close absolutely tight at the lowest speed of the engine.
3. The force necessary to shut the valve against the engine suction should be practically constant.
4. Friction in the valve and mechanism must be avoided as much as possible.
5. The action of the valve should be damped so as to be unaffected by the engine pulsations.
6. The entrance of air through the valve should be silenced.

The negative pressure should be maintained at about 5lbs., equal to 14 inches of water, and corresponding to a velocity of 240 feet per second, though, of course, this would vary somewhat owing to temperature variation of the air. Experiments prove that with a pressure less than that stated there is loss due to dribbling

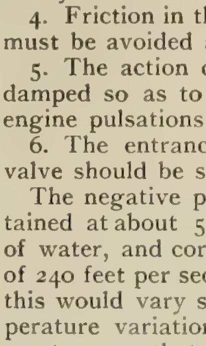


Fig. I.

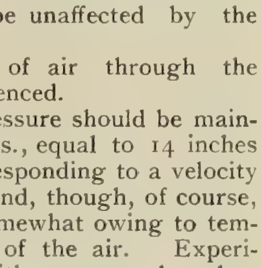


Fig. II.

at the jet and through the petrol not mixing thoroughly with the air. The average engine running at high speeds requires to take in charges at the rate of about 2,500 per minute, resulting in mixture oscillations of 40 per second, and the air taken in decreases as the engine speed increases, being from 50 per cent. to 75 per cent. of the piston displacement at the highest speed, it will be seen therefore, that the valve has a great deal more to contend with than is at first noticeable, and it will easily be seen in what measure the aforementioned points are fulfilled in the following examples of automatic air valves.

Fig. I. shows a type of what is probably the most common of air regulating devices which, though not satisfying many of the conditions laid down above, is used on many well-known carburettors. A is a piston (actuated by the engine suction) with slots cut in it corresponding with slots in the main body B, and the tension of the spring can be adjusted by the nut N. This is an actual design as fitted to one of the leading makes of cars, and it will be noted that the air ports are of special formation. There is a great difference of opinion as to the utility of these port formations, but the writer believes that, if their total area is sufficient, the shape cannot influence the quantity of air passing in, as the effective area of the ports, however they are formed, will be alike at similar engine speeds, thus only affecting the travel of the piston according to the length of the ports. It therefore follows that the wider the ports the less will be the movement to give the greatest opening. The experiments which have been carried out by the writer in connection with carburettors for aerial purposes confirms the accuracy of this reasoning in every way. One great disadvantage of the Fig. I type is its liability to stick, the main characteristic of the design being its simplicity.

Fig. II is a pattern which has found great favour in the United States. It will be seen that the valve is of the mushroom type, which has proved very satisfactory in practice, as the friction is small and the valve is proof against sticking. Also any slight wear can be taken up by grinding in, thus always keeping an air-tight joint. The particular feature of the design shown is the cone-shaped spring holding the valve A against the engine suction. When the valve first opens it compresses the large diameter coils of the spring S, and as it opens still further it gradually compresses the smaller diameter coils. It is claimed that tapering the spring thus makes it possible to offer a uniform resistance for a progressional valve opening, the spring resistance being directly proportional to the engine suction. As has before been mentioned these spring-loaded, piston and mushroom valves are objected to on the score of allowing too rich a mixture at low speeds, but as the valve simply vibrates on its seating at slow speeds, and offers a greater resistance to the entrance of air than it would at high speeds, its action is simply equivalent to increasing the spring tension at low speeds.

Fig. III is another type of piston valve, the novelty of which lies in the elimination of spring control by the substitution of a weighted slipper S, connected to the piston by links. In this case the movement is controllable by adjusting the sliding stop B. It will be realized that the controlling effect of this slipper or counterbalance over the piston is practically con-

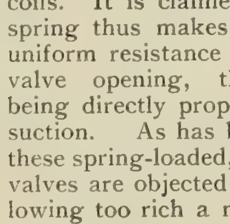


Fig. III.

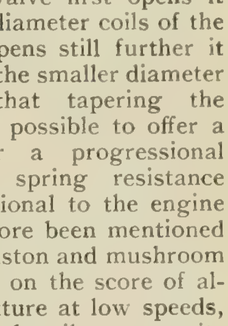


Fig. IV.

stant, unlike that of a spring which varies with the degree of tension and compression. This dead weight has the disadvantage of being unadjustable, whereby the negative pressure necessary to open

the valve cannot be governed, as in the case with an adjustable spring. Another disadvantage, common to all piston valves, is that after a period the piston wears, allowing air to leak past and the negative pressure to become small, so causing unsatisfactory running at slow speeds.

None of the preceding examples allow for damping the action of the valve, but in this respect Fig. IV is interesting. The valve proper is of the piston type, the damping device consisting of the chamber C containing glycerine or other suitable liquid. There is a plate sliding on a portion of the body, and actuated by means of the valve spindle E through the medium of the adjustable nut H controlling the spring tension. This plate has a number of 3-16 in. holes drilled in it, and another plate, on its upper surface, is fixed. When this is revolved it uncovers as many holes as required. It will be seen that any movement of the piston brings the first plate downwards, so forcing the oil through the holes. Owing to its viscosity the passage of the fluid is relatively slow, thus damping any violent oscillation. The spring S is very light, its strength being just sufficient to return the valve to its shut position, and, though not fulfilling theoretically many of the conditions laid down, this arrangement works very satisfactorily.

Fig. V is a valve of peculiar construction. It consists of a valve casing K upon the lower portion of which is the seating for the valve A, while at the top portion

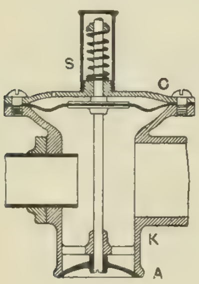


Fig. V.

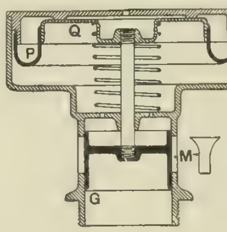


Fig. VI.

of the body is a leather diaphragm held in position by the cover plate C. The peculiarity of this valve is that the suction of the engine acts upon the diaphragm which is connected to the valve spindle. Being arranged thus, only a portion of the diaphragm is available for actually lifting the valve off its seat, since the pressure must act equally on the valve and diaphragm, and it is only because of the greater area of the diaphragm that any movement takes place. Once the valve is off its seating however, this neutralising effect practically disappears. A small hole is drilled in the cover through which air is drawn by a downward movement of the diaphragm, this actng as a dashpot. The valve is fitted with an adjustable spring at S, which is very accessible, and it has many points in its favour.

Fig. VI. is another type of valve which was first introduced by Krebs in 1902, in a paper before the Académie des Sciences, Paris. The valve G is vertical, and its stem is surmounted by a large disc Q, to which is attached the indiarubber diaphragm P, which is also permanently fixed to the cover plate, the valve ports being opened and closed by the suctional pull of the engine. There is a vent above the rubber diaphragm which allows air to enter slowly when any downward movement takes place, thus acting as a damper on the air controlling mechanism. The writer has conducted many experiments with this type of valve, and the only troubles were the hissing set up by the entrance of air and the piston, which stuck occasionally owing to grit entering the air ports.

Fig. VII. is another type of valve much used by Continental firms. The valve V is actuated by suction through the hollow spindle, which creates a vacuum in the case C, thereby causing the piston P to rise against the spring S, taking with it the cone-shaped valve. Damping effect is obtained by the entrance of air on the under side of the piston, as by coming in and passing out very slowly it deters any sudden movement of the piston. This is a very excellent type, as the large chamber B acts as a silencer to the incoming air and the valve, being very thin and cone-shaped, seats itself in, so reducing friction to a minimum.

The last illustration, Fig. VIII., is of a complicated though interesting type, which is now fitted to the new six-cylinder Renault engines. A. is the valve, above the stem of which is a balance-weight B, attached to the spring. The quantity of air admitted is in proportion to the valve lift, which is proportional to the engine suction, and the first movements of the valve are made with little difficulty, for only the inertia and weight of the valve need to be overcome. As soon as the valve stem comes into contact with the balance-weight however, the opening is more difficult, for the valve has to lift the weight and overcome the spring resistance, the weight, spring and valve being designed to give the correct quantity of air at all engine speeds. Immediately below the valve is a cylindrical chamber C, to which petrol is admitted to the same height as in the jet chamber, communication being by way of the passage V. In the chamber C is a piston, mounted on the same spindle as the valve. The piston is made slightly smaller than the chamber, and consequently a free vertical movement is allowed, though violent oscillations are prevented by the density of the petrol in which the piston moves. The chamber L is only to prevent petrol entering the air passage. To facilitate starting the balance weight is mounted on a quick pitch screw, the threaded cap connected to a hand lever, forming part of the control gear by which it may be lowered, thus closing the valve tight.

In reviewing the different designs shown the great differences in opinion can be detected throughout. All the valves shown

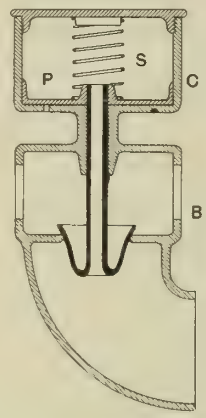


Fig. VII.

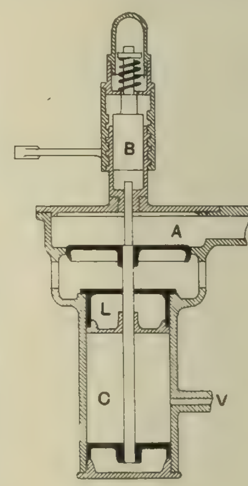


Fig. VIII.

operate independently of any construction controlling the fuel, and this is another point where opinions differ, as some designers believe in making the automatic air valve coupled to a fuel valve in such a manner that the relative openings are in a constant proportion, as witness the Polyrhoë, Scott-Robinson, and others. Still many of the types of valve illustrated could be utilised in this manner if it was considered desirable. After giving due consideration to the subject it is evident that there is much room for improvement in the design and construction of the automatic air valve. The success of the valve will always depend on its simplicity, especially in regard to adjustment and number of parts, and, while particular attention should be devoted to balancing the valve and reducing friction, the writer is of the firm opinion that developments in the direction of the air valve, either operating independently or in direct connection with the petrol admission, will do much to obtain that finality in carburettor design, of which we are all so earnestly in quest.

THE 40 HORSE POWER VELIE CHASSIS.

An American design with some ingenious and novel points.

THE Velie Motor Vehicle Co., of Moline, Ill., U.S.A., confine themselves to the manufacture of one chassis model of a nominal forty horse power, to which they fit any type of body. This chassis is equipped with a four-cylinder four stroke engine of $4\frac{1}{2}$ in. bore and $5\frac{1}{4}$ in. stroke, with a 75 lbs. gauge compression, and develops its rated power at 1,000 r.p.m. The cylinders are off-set $\frac{3}{8}$ in. and are cast in pairs with all the valves on the left hand side. In Figs. I. and II. it will be noticed that the valves are amply water jacketed, and a rough casting with four branches is used for the exhaust, while the intake ports of each pairs of cylinders are combined. All the valves are 2 ins. in diameter, have a $\frac{3}{8}$ in. lift and are provided with large radii at the head, being made of nickel steel electrically welded to machine steel stems, the head and stem being ground accurately to size. They are provided with removable cast iron guides of ample length, but unfortunately to remove these guides is far from easy as they are pressed in position.

Valve grinding is sometimes difficult when the valves are all on one side as it may be an awkward matter to remove and replace the valve springs because of the limited amount of room, but in this engine a toothed key is used for holding the valve spring which may thereby be removed and replaced with extreme ease.

Turning to the tappets and tappet gear,

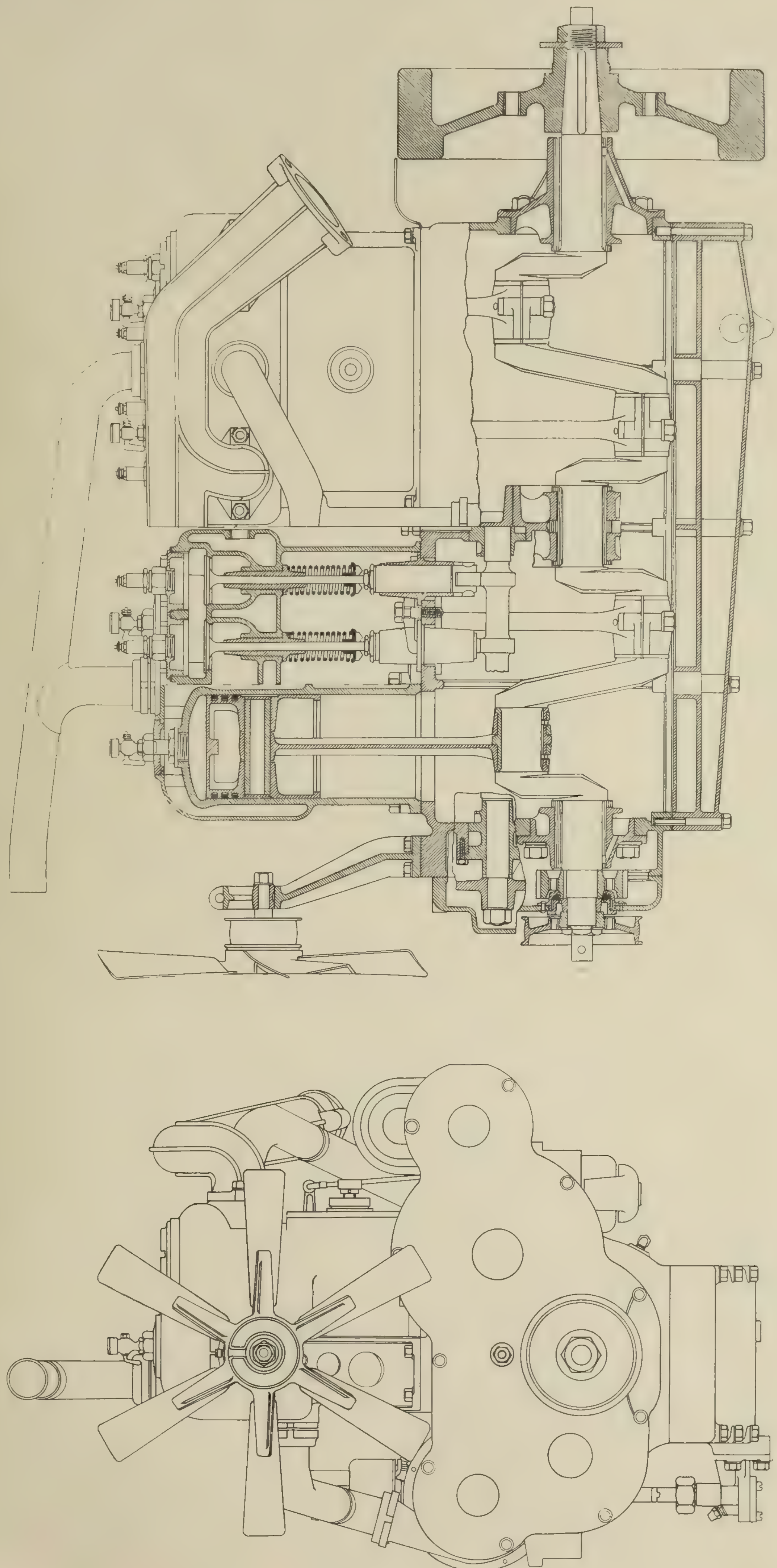


Fig. I.

THE 40 H.P. VELIE ENGINE.

Front end elevation and side sectional elevation.

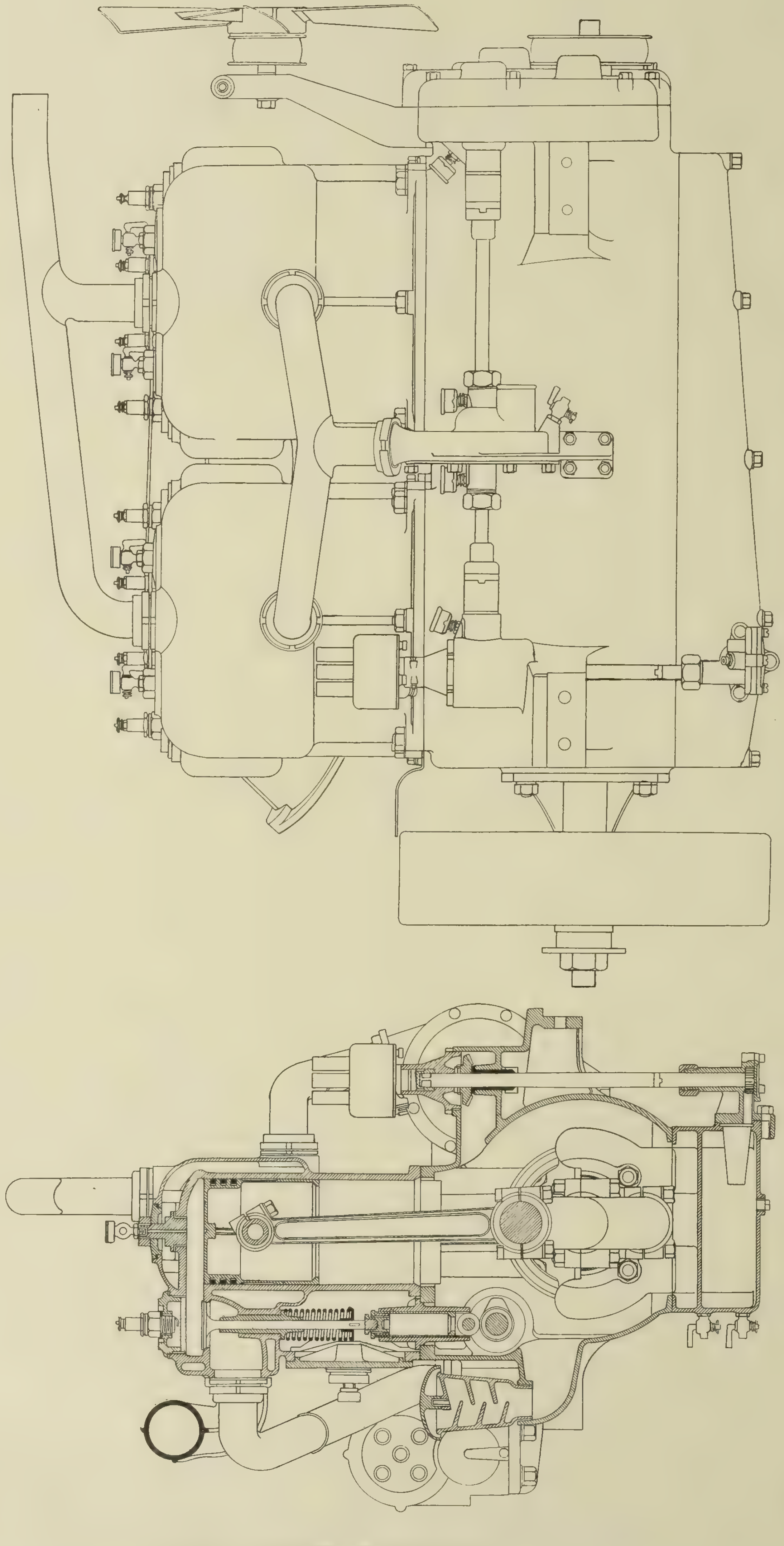


Fig. II.

THE 40 H.P. VELIE ENGINE.

Section through rear cylinder and off-side elevation.

we find quite a unique construction, which is clearly depicted in the vertical section Fig. II. A hardened and ground steel roller is used and the tappet proper is made of nickel steel tube of a large diameter having the forked end, which carries the roller, rivetted to it. This large diameter of tappet gives a good bearing and at the same time a light construction. Near the top of the tappet several small holes are drilled corresponding to a groove in the bearing or bronze guide, so providing means for returning the oil (pumped by the tappet) back to the crank case. This tappet is also provided with two adjusting screws and a fibre plug to deaden noise, while the valves are also enclosed further to assist in minimising sound. It will also be noticed that the fibre plug is set into the adjusting screw, to keep it from softening and splitting when oil soaked.

The camshaft is a one-piece forging of 40 ton carbon steel 1 in. in diameter, having a $1\frac{1}{2}$ in. tappet roller, and is provided with three long bronze bearings. A cast iron gear is keyed to the shaft, meshing with a steel pinion on the crank shaft and also with steel pinions driving the magneto and water pump, the magneto being placed on the left hand side on the front crank-case arm. All the timing gears are drilled to receive puller bolts to permit their easy removal, and a steel thrust washer is placed between the camshaft gear and the front bearing.

The cylinders are designed for cooling by pump circulation and are provided with expansion plugs in the head, to compensate for the unequal expansion and contraction of the metal. Three rings are used for each piston, and three oil grooves, these being relieved at the gudgeon pin boss, while the pin is clamped tight in the upper end of the split connecting rod and given a bearing in the boss of the piston, the avowed object of employing this construction being to obtain a longer bearing surface for the pin, which is $1\frac{1}{8}$ in. in diameter, is hollow and is ground to size. It will be noticed that part of the clamping bolt in the connecting rod passes through the pin to locate it endwise and prevent it from turning.

The connecting rods are drop forged from 40 ton carbon steel and are of I beam section, the crank pin end being provided with split die-cast nickel babbit bearings. Three packing slips are placed between rod and rod cap to provide a means of adjustment for wear. A dowel pin is placed in the cap to assist in locating it on the rod and also in holding the packing in place, as it can readily be understood that it would be a difficult matter to replace packing without a means for locating it. The crankshaft is also forged from forty ton carbon steel, double heat treated and accurately ground to size, having three bearings $1\frac{3}{4}$ ins. in diameter, the total length of bearing surface being 12 ins., while the crank pins are also $1\frac{3}{4}$ ins. in diameter and $2\frac{3}{4}$ ins. long. A pulley for driving the fan is keyed to the front end of the crankshaft while just behind it, and rivetted to the half time case, is a neat stuffing box to prevent oil leaking out at this end, while the flywheel is keyed on the usual taper at the rear end and is also drilled to receive puller bolts to remove it from the shaft. It is to be regretted that no provision has been made for a stuffing box or thrower-rings at the rear end of the crankshaft, though the usual provision is made for the return of oil at this point. The front and rear bearings are solid while the centre bearing is split, and all are bolted to the barrel type of crankcase. These bearing bushings are also die cast nickel babbit and no special provision is made for taking up wear in the front and rear bearings. The magneto is driven through the usual form of Oldham coupling, being held to its base by four cap screws and kept away from the heat of the exhaust; it is also placed high enough to keep water from reaching it. The water pump is placed on the right hand side, and is of the centrifugal type, having its shaft extending clear through and using an Oldham coupling at each end, the rear end serving as a drive for the distributer and oil pump (through a set of bevel gears at the top of the vertical shaft carrying the Atwater-Kent Unisparker). These gears are housed in the rear crankcase arm and receive ample lubrication from the splash in the crankcase. This makes a very simple and accessible drive while, by removing two bolts and the upper bearing of the vertical shaft, the entire drive and pump may be removed. Stuffing boxes are provided to guard against oil leaking through the drive and pump.

It would have been a good feature to place some kind of valve between the pump and the outlet from its case, so that the oil would not have to be drained from the case when removing the pump. However, as the Oldham coupling is provided in the drive, the lower part of the crankcase may be dropped with the pump attached so partially eliminating the undesirable feature. A large strainer is placed in the oil outlet, and oil is pumped from the crankcase to a sight feed on the dashboard, whence it flows back to the crankcase, where the cylinders and all the bearings are lubricated by a constant level splash system, the bearings being provided with pockets to catch the oil and feed it as required. Two pet cocks are used for determining the high and low level in the sump which, as stated above, is integral with lower part of the crankcase, having a capacity of eight quarts of oil. Stand pipes are provided for maintaining a level and an overflow for the return of oil to the sump, which slopes towards the rear to prevent oil from flowing away from the pump when climbing steep hills. It will also be seen that proper provision is made for the return of oil from all bearings to the crankcase. By dividing the crankcase in this manner a very rigid construction is formed for supporting the main bearings and cylinders while, by removing the ten bolts holding the lower half, it is possible to inspect all bearings.

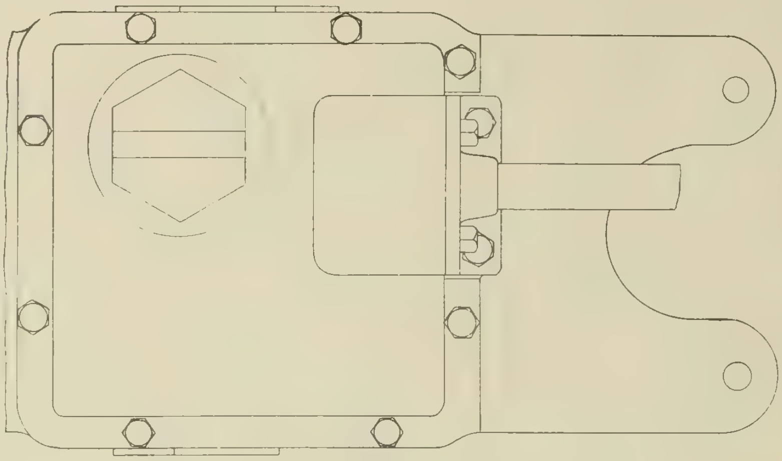
Eight sparking plugs are placed in the valve pocket covers, which are drop forgings and so designed as to prevent heat from accumulating in them, and though placing a sparking plug in the path of the exhaust gasses is not a very good practice, it is not a serious drawback, because the ignition system is dual, there being a Splitdorf magneto as well as the high tension accumulator system. Mounted upon the half-time case is a fan bracket, having an eccentric fan belt adjustment which may be locked with a bolt; while the fan is mounted upon ball bearings, has six aluminium blades and is provided with an oil cup of ample dimensions.

The entire cooling system has a capacity of five gallons of water, and a honeycomb radiator is used mounted upon an angle bracket on the front cross member of the frame, while two studs of $\frac{1}{2}$ in. diameter fastened to the bottom of the radiator at about 5 ins. on each side of the centre, hold the radiator in position. This construction relieves the radiator of any stresses which may occur owing to one of the road wheels mounting an obstacle, while, instead of using holes in the angle bracket to receive the studs, this bracket is slotted, so providing an easy means for removing the radiator without disturbing any other unit. This radiator also has a stay rod running to the dash, to brace it and to maintain an equal distance for the bonnet, while the bonnet ledge on the radiator is laced with leather to prevent rattle, as also is the ledge on the dashboard and that upon the frame.

A Kingston carburettor is used provided with a hot air pipe attached to the exhaust pipe.

Passing on to the clutch we find evidence of neat design. It is of the dry plate type, having three plates and thirty-two cork inserts, the centre or driving plate being made of hard bronze and carrying the cork inserts, while the driven plates are of gray iron. For securing the driving plates to the flywheel an unusual method is adopted. The driving pins are set into the flywheel and locked by means of gout screws, and they have also a groove turned in them to receive steel balls, which are placed in the boss as cast on the driving plate, a small spring being placed behind each ball to hold it in position. This provides an efficient means of holding the plate, while, at the same time, the latter can easily be removed with the remaining parts of the clutch. F. and S. ball bearings are used for the clutch spigot, and the holding spring is placed on the outside where the tension can easily be adjusted. The clutch thrust is self-contained, and disengagement is made through four toggle joints, provided with set screws and lock nuts which are very accessible. A neat clutch brake is provided—clearly shown in Fig. III., which depicts the entire clutch, and the universal joint between gearbox and clutch is divided horizontally so that the clutch may be dismounted without disturbing the engine or gearbox.

The gearbox is carried upon a subframe, as is the engine, lining up being accomplished with brass packing pieces. Three speeds are given, and the box is made by the Brown and Lipe Gear Co. It is shown in Fig. IV., the gear ratios and number of teeth being given thereunder. Gears and shafts are of chrome nickel steel, which is heat treated, all gears being held to their shafts by rivetted chrome-nickel steel pins instead of the usual key, while the shaft carrying the sliding gears is square, and hollow to provide a long bearing for the spigot of the driving shaft. The thrust of the sliding gears is taken by a hardened steel washer and four balls at the end of the driving shaft, while it is transmitted through this shaft to the Timken roller bearing upon which the shafts are mounted. These bearings are provided with adjustment from the outside of the



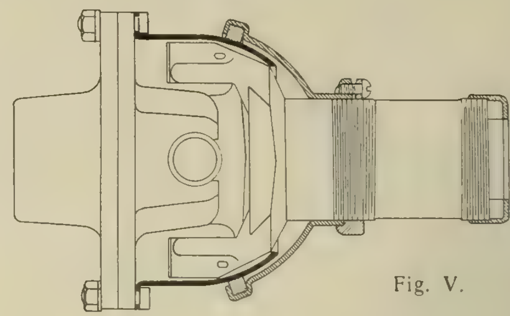


Fig. V.

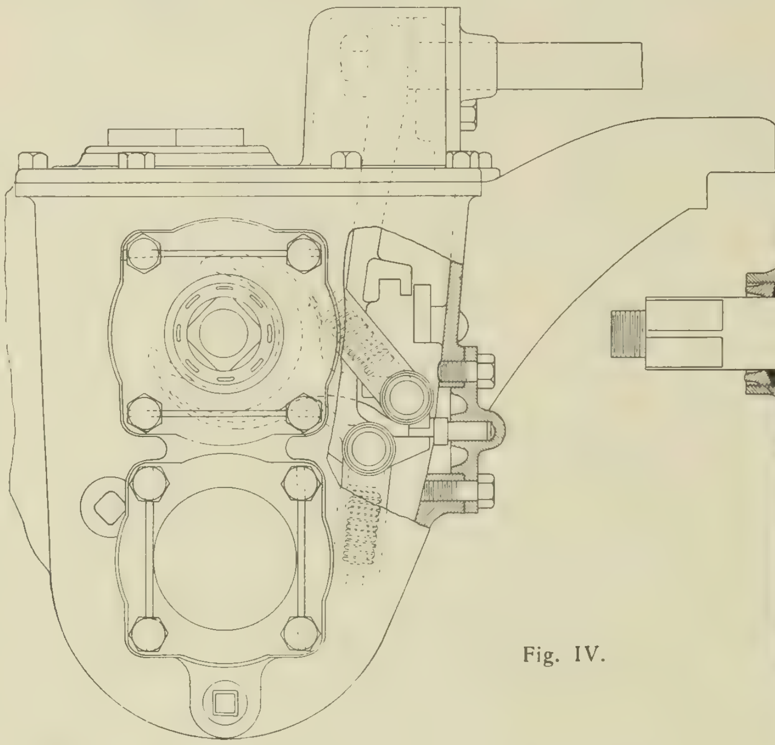
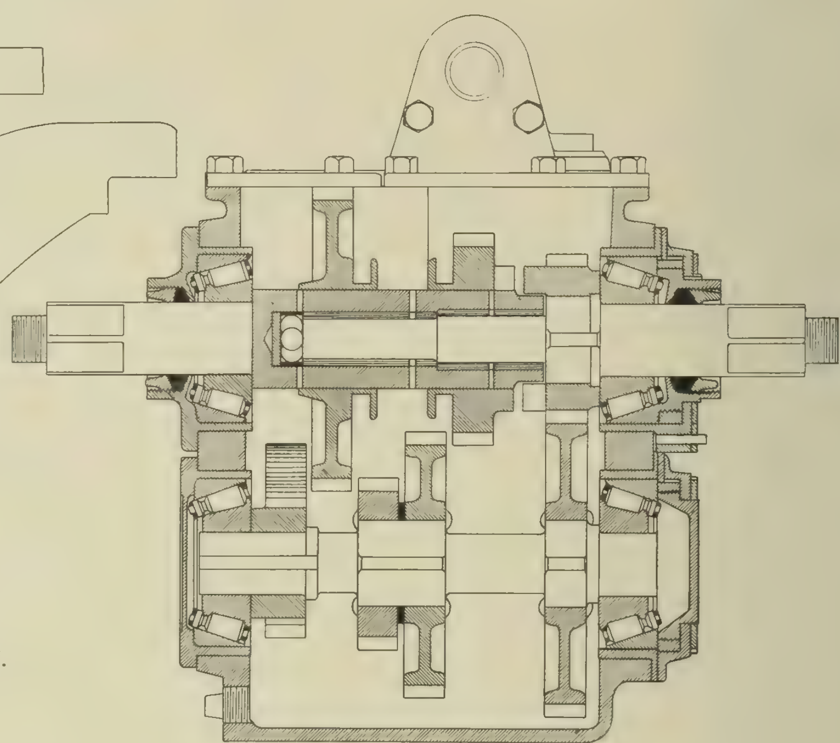


Fig. IV.



TRANSMISSION OF THE 40 H.P. VELIE CHASSIS.

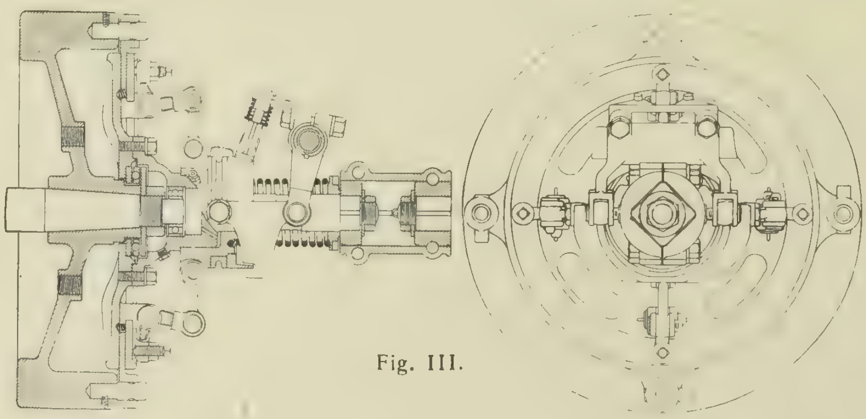


Fig. III.

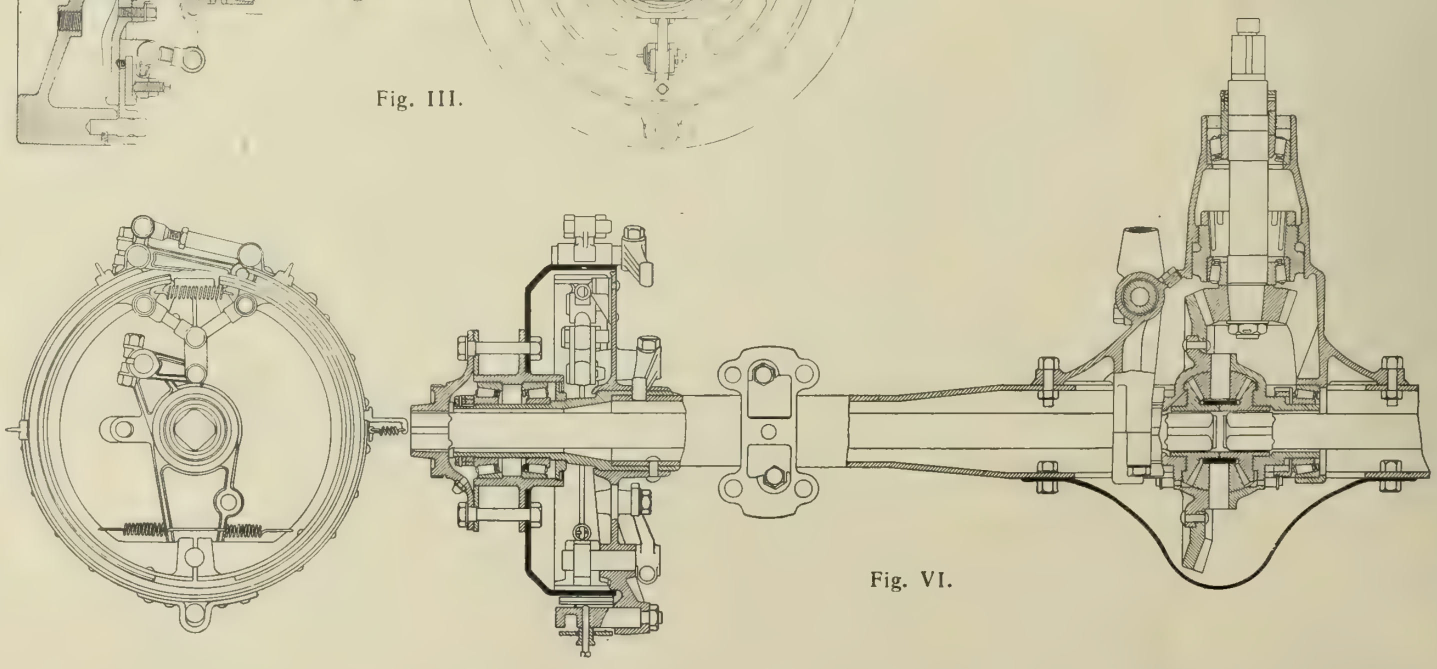


Fig. VI.

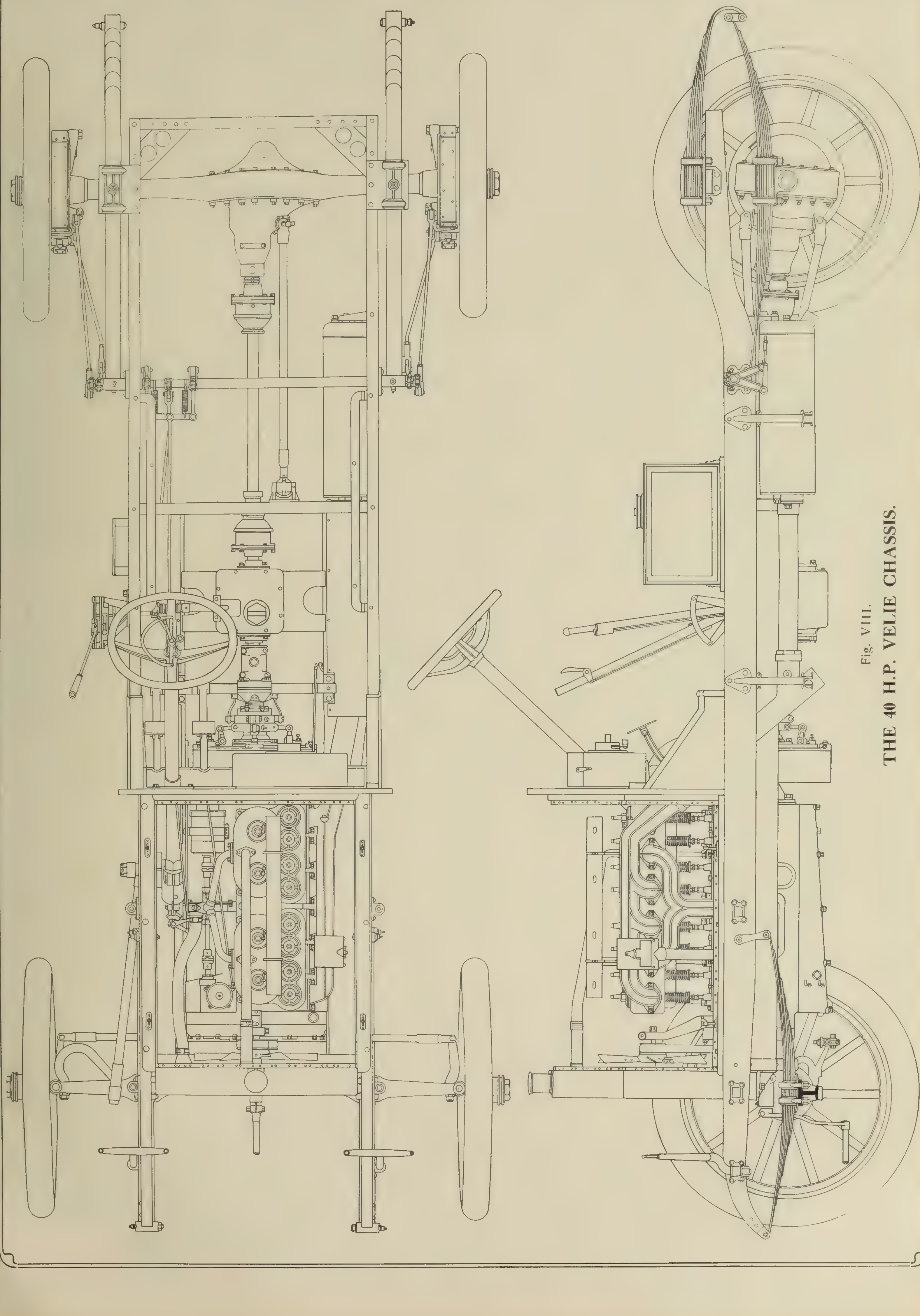


Fig. VIII.
THE 40 H.P. VELIE CHASSIS.

case at the forward end and are wisely carried in brass housings. Thus, as a whole, the gearbox is very short, so insuring great strength for the shaft, while it is guaranteed to transmit 60 h.p. The change gear mechanism is clearly shown in the back view in Fig. IV. : there is an internal gate and the striking forks slide upon stationary shafts, their sleeves being provided with the usual type of plunger and spring. The selection lever has a guide plate and pin to ensure its position for any speed, and works in slots cut in the striking forks themselves. Outside control is through a single slot gate and lever.

From the gearbox the drive is through Spicer universal joints (Fig. V.) and a $1\frac{1}{2}$ in. nickel steel shaft, these joints being provided with steel covers permitting them to be packed with grease. With the car loaded to its normal capacity straight line drive is obtained to the rear axle shown in Fig. VI., which is made by the Timken Roller Bearing Axle Co. The outer flanges of the hub and driving clutch are made integral, thus avoiding any play or rattle between the hub and hub-clutch. The roller bearings readily carry the thrust and radial loads of the wheels, and are of ample size, two being placed in each wheel and one at each end of the differential box and pinion shaft. There are four pinions in the bevel type differential, which is so arranged that the whole driving unit can be removed through the large cover, which is of pressed steel. The housing proper and the tubes are a one piece stamping, light and of great strength, while a casting bolted to this housing carries the driving pinion with its shaft and the differental with its bearings. A double V-shaped torque rod is fitted, pivoting in spring buffers carried in a bracket on the cross member behind the gearbox, and running parallel with the driving shaft. Two sets of brakes are provided, both on the hubs, the service brake being contracting and the emergency brake expanding, both operating on the same 12 in. pressed steel drums, and both being lined with Raybestos. These brakes and their operating mechanism are clearly shown in the end view in Fig. VI.

The drive from the rear axle is taken through the rear springs, and, although this is a radical departure from the present tendencies in design, the starting of the car is rather smooth. The brake rods all pivot from the same point, making the brake action good with the axle in any position.

Fig. VII. shows the front axle, which is also made by the Timken Company and provided with their roller bearings. It is a drop forging of I beam section with integral spring pads, having the cross rod behind it and the steering knuckle lever above it to protect both from striking obstacles in the road, a feature of this construction being that the front axle is the lowest point of the entire chassis and serves as a protection for the remainder of the mechanism, an important point where very bad roads are common.

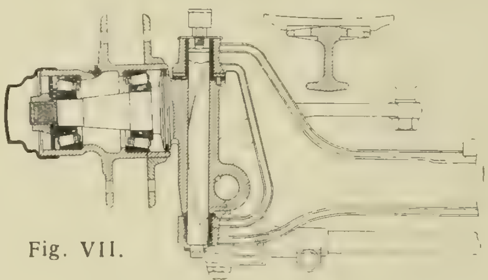


Fig. VII.

The steering gear is of the worm and sector type provided with large ball thrust bearings, and has a 17 in. hard rubber wheel with corrugated finger grips moulded on an aluminium spider. The control sector is placed above the wheel, the control being transferred by bevel gears to a vertical control shaft on the housing. The gear ratio is 2 to 1, and the column is a $1\frac{3}{4}$ in. tube, mounted at an angle of 47 degrees with the main frame. It is supported from the sub-frame where it is thoroughly protected from dirt.

Passing on to the chassis Fig. VIII., it can be seen that careful study has been given to the arrangement of the various units in an accessible manner. The frame is of channel section 4 in. x $1\frac{1}{2}$ in. and mounted upon semi-elliptic springs in front 38 ins. long and 2 ins. wide, while the rear is mounted upon three-quarter elliptic springs 46 ins. long and 2 ins. wide. A sub-frame of channel section 4 ins. by $1\frac{1}{2}$ in., carries the engine, gearbox, steering column, radiator and pedal shaft, relieving the stresses which arise in the side members due to their angularity with the road. Small permanent mud aprons are used between side frame and main frame, while the main mud apron is carried upon spring buffers to permit its easy removal and replacement and to prevent rattle. It is also lined with leather, and this principle is extended by lining the entire frame with leather where the body rests, to prevent squeaks and other noises which usually arise at this point.

The petrol tank is mounted directly upon flexible brackets, also lined with leather and rivetted to the side frame members, while the clamps for holding the tank are lined with leather too. An unusual amount of rattle is found in most tanks, and to overcome this feature heavy ribs have been placed inside of the tank and aluminium coated sheet steel is also used instead of copper, which greatly stiffens the tank, in which a reserve compartment and valve is an integral part. Fuel feed is by gravity, but a hand air pump is applied for pressure when climbing steep hills, with a low level in the tank. Oil and grease cups are placed at every point requiring lubrication.

Drop forgings and pressed steel are used through the entire construction, which no doubt is responsible for the remarkable lightness of this high powered car, the weight of the complete five-passenger touring vehicle being about 2,900 lbs., when the body is of the normal open type.

THE MACHINING OF ALUMINIUM CASTINGS.

Giving some Useful Hints in connection with Various Operations.

By D. Walters.

The machining and cutting of aluminium is a comparatively modern operation, having developed with the advent of the automobile trade, in which industry the metal is used extensively for engine cases, gearbox cases, covers for the magneto and camshaft drives, and various other small items such as circulating pump bodies, etc. An engineer of the older school, engaged in the heavier or various textile machine making branches of the trade, rarely if ever, has occasion to machine aluminium, and, indeed, regards it somewhat as a novelty or curiosity when he sees any of that metal being operated upon.

Although aluminium is used in large quantities in most automobile manufacturing concerns, its high price is greatly against its adoption on a still larger scale, and several of the firms who now make a popular priced car are using cast-iron engine and also gearbox cases in some instances, although the latter material is considerably heavier than aluminium, and it would seem reasonable to assume that pressed steel would prove a better substitute where cheapness is a matter of prime importance. More particularly so perhaps, when it is remembered that cheap cars are most often produced in large quantities.

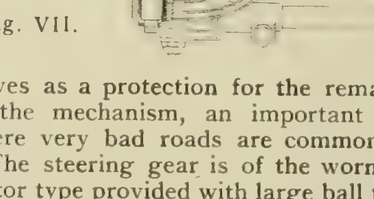


Fig. I.

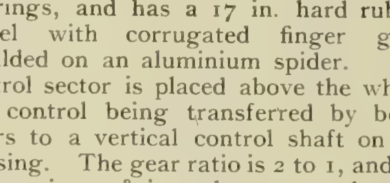


Fig. II.

In practically all progressive firms aluminium is faced up on milling machines, both vertical and horizontal, to the entire exclusion of their rival, the planing machine, which not long ago had a fair proportion of this work assigned to its department. Milling machines are pre-eminently adapted for the economical machining of such metals as aluminium, on account of their very wide range of feeds and speeds, which the planer or shaper cannot be said to possess.

Practically all the large flat surfaces on aluminium castings are milled up with inserted milling cutters in large cast-iron heads, these being cheaper to make than solid steel cutters of equal size. Fig. I. shows a cutter of a type familiar to most machine shop officials and operators; it is very good on cast iron, but this type of cutter does not give good results on aluminium. The flat cutters or blades do not give sufficient clearance for the chips which clog under a rough surface, although the cutter travels so fast that the chips fly in all directions. A

cutting speed of between 500 feet and 600 feet per minute is quite common with a feed of 12in. to 14ins., and with turpentine as a lubricant a very excellent finish can be obtained at this speed and feed,

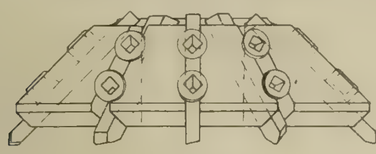


Fig. III.

resulting in a surface polished like glass, provided suitable cutters are used. The writer has seen very excellent results obtained with the types shown in Figs. II. and III. in the accompanying sketches, whilst, for very narrow or confined spaces, the type shown in Fig. IV. is very suitable, being simply a steel shank turned to fit the milling machine spindle with a fly cutter inserted at the bottom on an angle as shown, and secured by a set screw or wedge. The type of cutter shown in Fig. II., is of round section steel, being secured in position by grub screws of fairly large diameter, having a sunk square hole in each to insert a suitable spanner for tightening. It will be appreciated that it is very dangerous to have any projecting screws when running at the high rate of speed common for these cutters. The cutters should be placed radially with the front rake ground

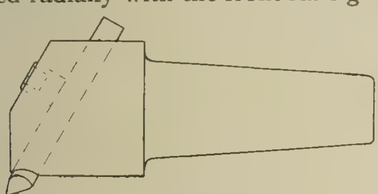


Fig. IV.

off; they should have a wide angle from the vertical so as to give the cutters a good chance to throw the chips clear of the following cutting edges, and thus prevent clogging, which produces a very rough finish on the work. The writer has found this an important item in making cutters for this class of work, and suggests an angle of 45 degrees at the least. The cutters only just cut at the extreme points, and do not have a broad scraping surface like the type shown in Fig. I. They give a clear, sharp cut, which is the secret of obtaining satisfactory results when working the material in question.

The spacing of the cutters is a subject on which experts disagree and, whilst

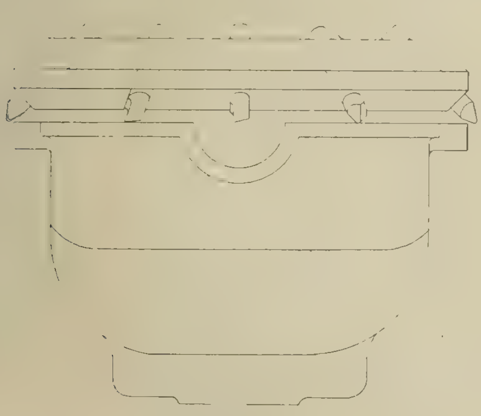


Fig. V.

some firms space the cutting edges not more than 2ins. or 3ins. apart, the writer personally prefers them very much wider, say 5ins. or 6ins., which gives excellent results. If the cutters are kept sharp, with the cutting edges spaced this far apart, the chips have certainly a much better chance of clearing themselves, and thus prevent crowding under the succeeding edges.

Fig. III. shows a type of cutter using square section blades, and instead of a hole being drilled through at an angle to receive the cutters as in Fig. II., a number of gashes are milled in the width of the cutter it is intended to use, and two large hollow grub screws similar to those used in Fig. II. secure each cutter in

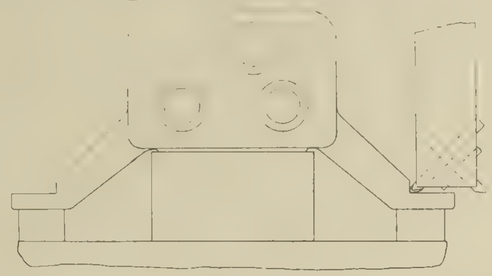


Fig. VI.

position. Of course the two holes should be drilled and tapped out before the cutter slots are milled, the latter being milled across the centre of the holes, and the screws held by about $\frac{1}{4}$in. at each side of the cutter in the material which is not milled away, the screws, of course, being much larger than the section of the cutters. All the above-described cutters screw on to a spindle or shank which fits in the milling machine spindle.

The small cutter, shown in Fig. IV., is very suitable when working along narrow surfaces, or for re-entrant angles in a confined space. It is turned solid with the shank, and only has one cutting edge on it. Fig. V. shows a large cutter operating across the joint of the bottom half of an engine case casting. In

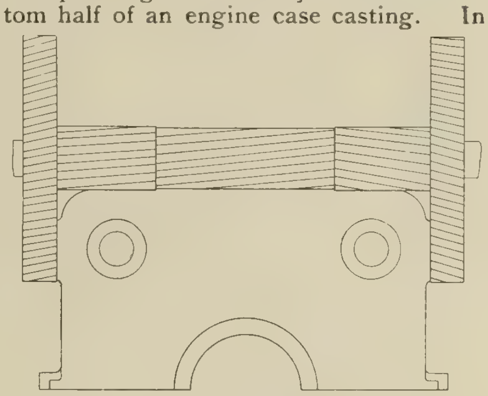


Fig. VII.

some instances the whole face is covered, as shown, or, if the diameter is insufficient, one side is machined on the forword feed of the table, and the opposite side on the backward or return feed, both methods being quite satisfactory and giving equally good results.

Fig. VI. shows a small type of cutter operating on the feet of a gearbox, the latter being packed up on suitable packing strips, and clamped down securely on to the milling machine table. The cutter is fed across the feet as shown, the two inserted blades being held in position by grub screws being placed in various positions. A very high spindle speed is required for this type of cutter and, as speed variators are now universal, this usually can be effected without changing any belts.

Fig. VII. shows the top half of an engine case being operated upon in an entirely different manner. Whereas the before-mentioned operations have been performed on a vertical spindle machine, the one here described is done on a hori-

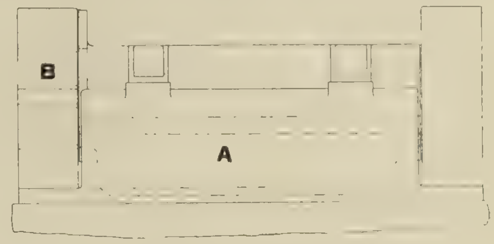


Fig. VIII.

zontal spindle type of slab miller, with gang cutters for face and side milling at the same operation and, whilst the same degree of finish is perhaps not obtained across the top face, the economy effected is very great, as surfaces can be completed at the same time by simply ganging cutters of suitable size. Those cutters operating across the top are not of the inserted type, but solid steel body cutters, whilst the two outside cutters which mill the camshaft inspection cover faces, are sometimes solid and sometimes of the inserted blade type. The writer knows cases where these operations were formerly done on planing machines at a labour cost of about 10s. each, whereas by the gang method of milling the same operation now costs about 1s. 3d. Such is the economy of milling as compared with former and unsuitable methods.

When boring, drilling or facing aluminium, much faster feeds and speeds can be employed than with most other metals, just the same as in milling, and, where jigs and suitably designed fixtures are used, very great economy can be effected in labour costs and a higher standard of accuracy obtained in the finished product—an essential feature in automobile construction. Fig. VIII. shows an engine case casting placed in position in the jig on a boring machine table for boring and facing the crankshaft and camshaft bearings. The jig or fixture is located accurately on the boring machine table by a central tongue fitting in a slot, as shown in the end view, Fig. IX. The case, which has been previously milled and drilled at the joint for the securing bolts, is assembled and the four feet dropped on the planed surfaces to receive them, as shown on the top of the

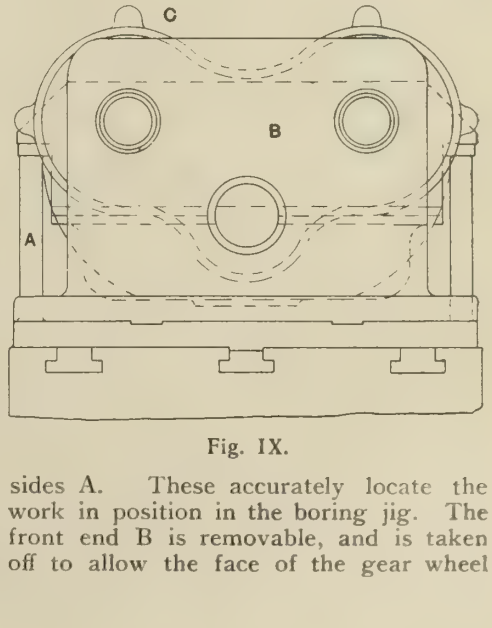


Fig. IX.

sides A. These accurately locate the work in position in the boring jig. The front end B is removable, and is taken off to allow the face of the gear wheel

guard C, which is cast solid with the case, to be faced up. This operation is accomplished by an automatic or hand cross feed, and a cutter, as shown in Figs. II. and III., inserted in the machine spindle in place of the boring bar. The work is fed across the face of the cutter, and the finish obtained is excellent. It is, in reality, a horizontal milling operation performed on a boring machine. The front end B is now secured in position by the two tongues, shown in Fig. IX., and set screws. The boring bar is then inserted through the hard steel bushes, accurately spaced for the correct gear centres, and the holes bored with a single or double ended boring cutter of the ordinary type which calls for no mention.

The facing of the various crank and camshaft bearing supports is best effected by an inserted cutter facing head, as may be observed in Fig. X. It is bored out to fit the boring bar and slipped on, being driven by a short bar in a cotter hole, of which there will be several in each bar for various sized cutters. The slot milled across the back, and shown in Fig. X., drops on the cotter and the latter drives the facing cutter in this satisfactory manner. At D the four cutting edges are shown, the cutters being secured by the wedges E. They are placed radially and held in position by the set screws shown, which pass through the wedges and into the body of the cast iron cutter head, thus forcing the wedges up against the cutting blades and holding them. This type of cutter is far superior to the ordinary flat, straight cutter simply passing through the bor-

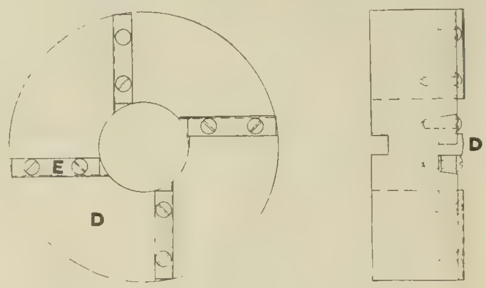


Fig. X.

ing bar, and makes a much better job in much less time. Of course, right and left handed cutters are required to face up the opposite sides of the bearing faces.

When drilling aluminium, a very high rate of speed should be attained, and when using twist drills care should be taken, when breaking through the holes, that the bottom edges are not broken or forced off the castings. Straight fluted drills are often used for this purpose, as there is no tendency on their part to feed through the hole when the point is forced through at the same speed as the pitch or lead of the spiral flute.

Aluminium is very difficult to tap by a machine tapping attachment for, owing to the soft nature of the material, it clogs on the taps and quickly strips the threads out of the holes. Especially does this apply when trying to use taper taps, for, instead of tapping the holes out, they ream them out into a taper hole; therefore a plug or bottoming tap is the best to use when trying to tap a hole out on a drilling machine. This will curl a clear chip off the metal if the tap is sharp, and some firms turn a few threads off the bottom of the tap and drill a small hole just in front of the commencement of the thread, into which the chip is supposed to curl and drop out at the bottom through a hole drilled up the centre to meet the cross hole. In this case usually the tap has no flutes whatever milled up it, so that there is only one cutting edge. Of course blind holes cannot be tapped to the bottom by this method. Machine tapping on aluminium is not attempted on any large scale, as far as the writer's experience goes, that operation generally being performed by hand by the assemblers or erectors as the case may be.

THE CONSTRUCTION OF AEROPLANE FUSELAGES.

A Brief Review on Current Practice.

By W. G. Aston.

THERE appears to be a certain amount of misconception with regard to exactly what is meant by the "fuselage" of an aeroplane, and this probably results from the borrowing

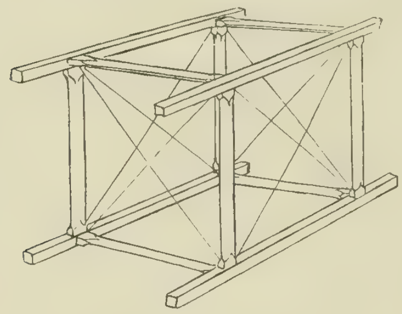


Fig. I.

of this uncomely word from the French, but for the purpose of the present article it will be considered to imply that part of the framework of the machine which, in a monoplane, supports the engine and pilot, and connects the main planes with the subsidiary controlling planes. In a biplane exactly what constitutes the fuselages is less easy to define, indeed many authorities suggest that this type of machine, if of the accepted Farman type, properly speaking, has no fuselage at all. For present purposes it will be regarded as the member which supports the tail from the main cell of the machine.

It will be as well, before dealing with specific details, to consider the principal desiderata of these fuselages, which may be classed under the following headings:—

1. Lightness.—Apart from the natural requirements of an aeroplane, which enforce the greatest possible reduction in weight in all its component members, there is a special reason why efforts should be made to reduce the weight of the fuselages to a minimum; namely that, in the usual type of machine, with the main planes in front and the tail behind, the use of a considerable length of fuselage, in order to accentuate the natural longitudinal stability of the machine, throws a considerable load upon the tail, and, in some cases, necessitates the use of a lifting tail in a machine which would best be served by a non-lifting tail, the function of which is purely directive. The only remedy in this case, if a non-lifting tail is insisted upon, is to balance matters by extending the fuselage considerably forward of the main planes, and mounting the engine at its extremity, as in the case of the Antoinette, which, in the opinion of some designers, has certain disadvantages.

2. Strength. — The actual strength which a fuselage requires to have for flying purposes is so slight that if the

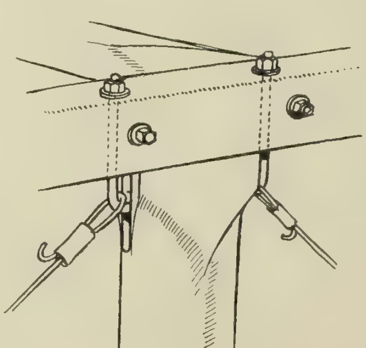


Fig. II.

machine could only remain in the air, it could safely be made with the most flimsy material. Unfortunately however, the aeroplane has to be a land vehicle as well as an aerial one, and it is in running along the ground before rising, and afterwards in alighting, that the fuselage receives the most severe shocks. Closely

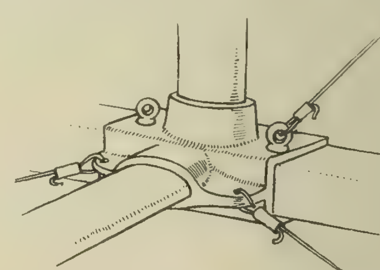


Fig. III.

connected with the question of strength is the matter of rigidity. This quality is just as important, and subject to the same modifications, as was pointed out in connection with supporting surfaces. The matter resolves itself into a pure compromise. On the one hand there is a perfectly rigid framework, which is liable to fracture under sudden shock, and, on the other hand, there is a more or less flexible affair, which by means of its latent "give," can absorb a certain amount of shock without damage. It is fortunate that in the aeroplane this small degree of flexibility is no disadvantage at all. On the other hand, there is every reason to believe that when a non-lifting tail is employed it may even be a decided advantage. This point however, seems to have escaped a number of designers, who, if they had realised it, undoubtedly could have reduced the weight of their machines quite appreciably. On the other hand this flexibility must not

prevail throughout the entire fuselage, as it is, of course, important that the main weight should be fixed, and the engine bedded down to a structure as rigid as possible.

3. Minimum Head Resistance.—If we may judge from the examples exhibited at the recent Olympia Show, it is evident that designers are beginning to realise what a large proportion of the total head resistance of a machine is accounted for by the fuselage. To take the most common form of monoplane fuselage as an example, namely, the Bleriot type, illustrated in Fig. I., this consists of four longitudinal members, united by a series of wooden cross pieces and braced by numerous diagonal wire stays. The amount of head resistance produced by this construction, even when the fuselage is meeting the wind "end on," is very considerable, and may be considered to be practically equal to the resistance of all the members of the construction taken separately, and added together. Experiments made by Dr. Stanton, of the

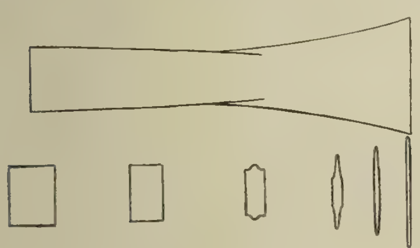


Fig. IV.

National Physical Laboratory, have shown that when one body is placed in front of another, and submitted to a current of air, the front one will exercise a shielding effect upon the rear one, which depends upon the size of the bodies, and the relative distance between them. In the case of the cross pieces of fuselages these are nearly always placed so many diameters apart that all shielding effect is entirely lost, and this result is accentuated by the fact that as a rule, cross pieces are made roughly of streamline section. In order to avoid this disadvantage, a number of constructors cover their fuselages with fabric or thin wood, but unfortunately this practice introduces serious disadvantages which have to be taken into careful consideration. Several examples at the Olympia Show indicated that this consideration had been omitted. It is unquestionably of little value to take a Bleriot type fuselage, and merely convert it into a long box by the addition of panels of fabric. To do so in fact, is frequently to convert a moderately safe machine into a decidedly dangerous one owing to the large

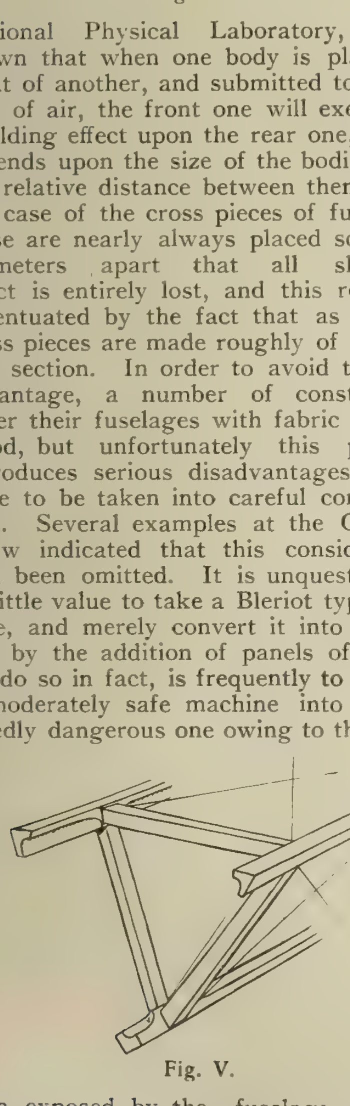


Fig. V.

area exposed by the fuselage to side winds, and to the resistance which it offers to quick turning movements, in which circumstances it behaves in the same manner as a large fixed rudder.

It is clear that to avoid these difficulties the section of a covered-in fuselage must be such that, when it passes through the air broad-side on in either direction, it must offer a minimum of resistance. If this side resistance be high the machine is liable to become absolutely uncontrollable in a side wind of any appreciable strength. Close by the centre of gravity of the machine the section of the fuselage

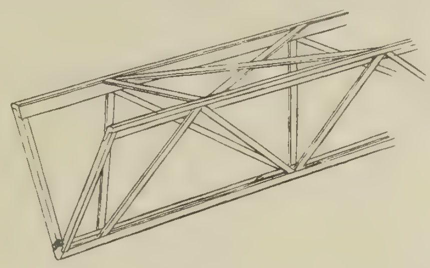


Fig. VI.

is, comparatively speaking, of small importance, but towards the tail, where the angular speed during a turn is fairly high, and where also the moment is considerable, it is essential to have a section of very low head resistance. This has been carried out very well in several modern machines, notably the Bleriot two-seater and Handley Page monoplane, sections of the fuselages of which are given in Figures IV. and IX.

Some of the more important constructions will now be considered in detail. The construction of the ordinary Bleriot fuselage is shown in Fig. I., which illustrates a small section of the total framework. There are four square sectioned longitudinal members, composed of either ash or pine, and united by a series of cross pieces mortised into them. Rigidity is obtained by providing a diagonal wire stay between the opposite corners of the panels formed by the sides of each cell. The fuselage is of maximum

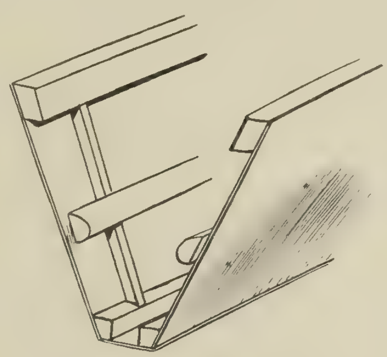


Fig. VII.

section in front, and tapers towards the tail in all dimensions. The method of attaching the cross pieces to the longitudinals is indicated in Fig. II. Owing to the system of cross bracing with tension wires, the wooden cross pieces have only to take a compressional stress, and they are therefore not positively secured to the longitudinals, but their tongues are merely inserted into the mortised grooves already mentioned. The stay wires are attached, as shown, to a broad U-shaped bolt, passing through the longitudinal with its vertical ends, and having its middle part accommodated in a recess formed into the cross pieces as shown in the figure. Of course, the vertical cross pieces are slightly staggered so as to prevent the parts fouling one another. This construction is both simple to assemble and extremely rigid, but it has a decided disadvantage in that it requires the longitudinals to be pierced considerably, with the result that they have to be made of fairly stout section so as to preserve adequate strength. It will also be seen that at each joint there are four holes, two in one direction and two in another.

The Beney method (Fig. III.), of building a fuselage of the Bleriot type is interesting, as showing a means by which this cross piercing can be avoided. In this case the cross pieces are attached to the longitudinals by means of cast aluminium sockets, the horizontal bracing wires being attached to special ears, cast on to the sockets as shown, whilst the vertical wires are taken to eyebolts which pass through the longitudinals in a vertical direction, and serve to keep the socket casting securely in position. In this case it will be observed that the longitudinal is weakened in the plane in which it has the least stress to withstand, but, on the other hand, the weight of construction is added to by the considerable number of aluminium sockets required.

Hitherto the Bleriot type fuselage has been easily the most popular for monoplanes, the various constructions dif-

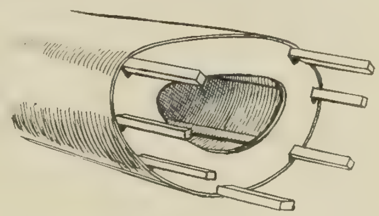


Fig. VIII.

fering from one another only slightly in the manner of attachment of the cross pieces.

The latest Bleriot monoplane intended for high speed work has a covered-in fuselage built up in the same way as the above, but instead of tapering to the tail it is spread out as shown in Fig. IV., so that, when viewed from above, it has somewhat the appearance of the side elevation of a fish. This is an excellent arrangement from almost every point of view, except that, after some little use, the fabric forming the covering is certain to get somewhat slack with the result that it will bag between its supports, and in consequence enhance the head resistance. At the same time it must be pointed out that a fuselage of this sort, in which the "trunk," as it were, spreads out into a tail, is not suitable for machines in which the tail has any lifting to do, and for this reason the type of Bleriot machine which still has a lifting tail, is furnished with the older type of open fuselage.

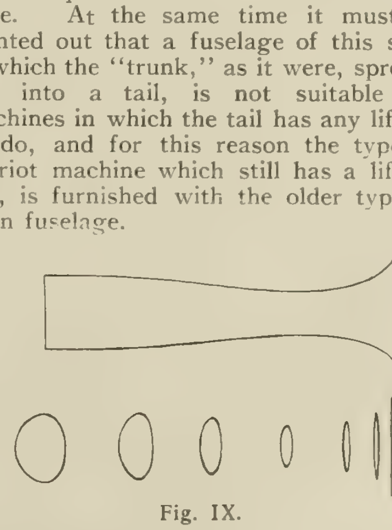


Fig. IX.

A good type of fuselage is that used on the Antoinette, a similar construction being employed on the Bristol monoplane, which, as illustrated in Fig. V.

This is a somewhat similar arrangement to the Bleriot, but is triangular in section instead of square. The resulting construction appears to be equally strong, and is, of course, a good deal lighter, as it employs considerably fewer parts. There is no difficulty in making the bottom member strong enough to perform

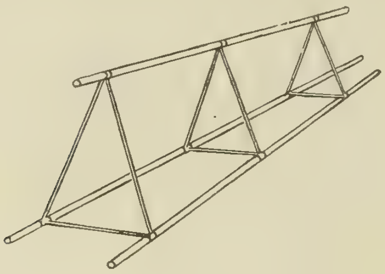


Fig. X.

the work accomplished by the two members which take its place in the Bleriot, for in the ordinary course of events the bottom member is purely in tension. At the same time a triangular fuselage of this type offers no very serious disadvantage if simply covered in with fabric, as its section is, comparatively speaking, such as to promote no excessive resistance to side winds.

In the Bristol the longitudinal members are considerably hollowed out between the joints of the cross pieces as shown, the latter being mortised in the usual manner, while the tie wires are supported by plain eye bolts. In the Blackburn monoplane a similar section of fuselage is used, but in this case the use of tie wires is entirely dispensed with. Fig. VI. indicates the construction. There are first of all three longitudinals of deep section, the wooden cross pieces uniting them being mounted in such a manner that the fuselage as a whole "gives"

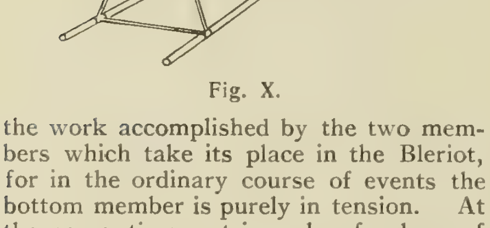


Fig. XI.

slightly under a stress. This is a very desirable feature, and is accomplished in this particular instance in a somewhat ingenious manner, though, at the same time, it is rather doubtful if it can adequately be carried out without adding rather appreciably to the weight of the construction. It will be seen from Fig. VI. that in no case does a complete series of cross pieces at right angles to the longitudinals occur at any one point. Where there are vertical cross pieces uniting these longitudinals, a horizontal piece at

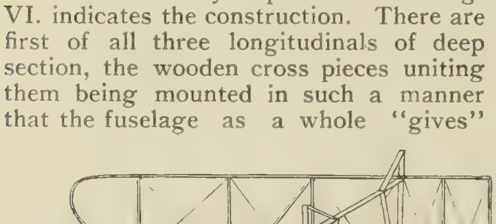


Fig. XII.

the top is absent, and *vice versa.* A direct cross piece is always subtended by two oblique ones. The result of this arrangement is that, at various points, each of the longitudinal members has a considerable length unsupported, and, when submitted to a compressional stress, this length is free slightly to bulge outwards, and so provide a measure of elasticity. At the same time no sacrifice is made in the strength of the fuselage or in its capacity to resist twisting.

This fuselage, like the Antoinette, is completely covered in, at the front by thin wooden sheathing and over the greater part of its length in the rear by fabric.

In the Martin Handasyde (Fig. VII.), the fuselage follows somewhat the lines of the Hanriot, and is constructed entirely of wood. Instead of being brought to a point, the apex of the triangle is cut off so as to form a narrow floor which is swelled out considerably in front so as to provide a comfortable cockpit for the pilot. The arrangement of longitudinals and cross pieces is shown in the figure. Much can be said for this method of construction in point of strength, cheapness of construction, and low head resistance, but it is scarcely as good a form as that of the Handley Page monoplane, which

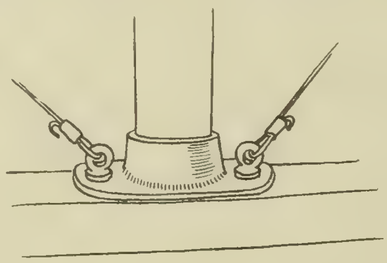


Fig. XIII.

is illustrated as to its construction in Fig. VIII., and as to its sectional grading in Fig. IX. In this instance efforts have been made to obtain a fuselage of true icthyoid form throughout, and the result is that it offers probably less head resistance than any other, while in fact, the resistance of the fuselage when turning is reduced to a minimum. At the same time the total area offered to cross winds by the rear portion of the fuselage, that is, behind the centre gravity of the machine, is practically balanced by the area in front; consequently a side wind should have no tendency to destroy the machine's sense of direction. Referring

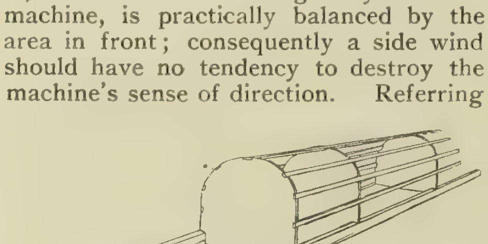


Fig. XIV.

to Fig. VIII., there are, as shown, seven longitudinals, with a series of forniers secured at intervals along them, and to these is tacked a thin three-ply covering. The resulting construction is, considering its strength, remarkably light, as of course, the longitudinal members do not have to be made of very great strength.

The Santos Dumont monoplane has a fuselage built up as shown in Fig. X., entirely of steel tubes, the cross pieces having an oval section and the joints being merely pegged and brazed. The section is the reverse of the triangular forms already mentioned, as in this case the triangle stands on its base, the result is that the single longitudinal forming the apex has to stand a considerable amount of compressional stress, whilst the tension is taken by the two longitudinals forming the ends of the base. This would probably not be an advisable method of construction in fuselage of considerable length, but in the Santos

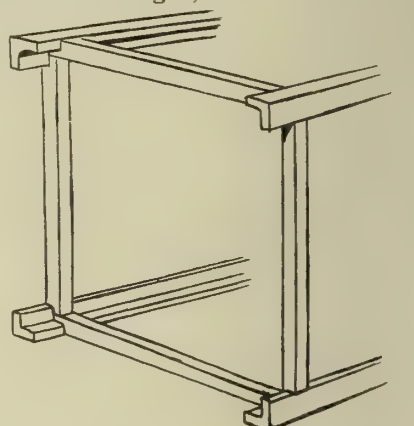


Fig. XV.

Dumont this member is comparatively short, and is submitted to very little stress.

The Valkyrie monoplane, a sketch of which is given in Fig. XI., is of unique construction, the fuselage serving, in addition to its ordinary function, the purpose of forming a long skid. The arrangement seems to work extremely well, but there are certain disadvantages which might however, be avoided, namely, that in the event of the machine coming down to the ground sideways (which is quite likely to happen when it is flying slightly across the wind) there is a strong chance of the fuselage being carried completely away and, as the wings are braced to it, this occurrence would involve damage to a considerable extent. At the same time

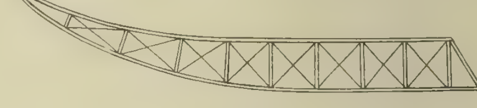


Fig. XVI.

it cannot be denied that by making the fuselage serve a double purpose a very considerable amount of weight is saved, but, on the other hand, the fact that the cross pieces cannot be covered in any way makes the head resistance of the fuselage, as a fuselage, rather high. The main horizontal cross pieces uniting the two halves serve to carry a cradle, on which are mounted the pilot's seat and the engine, additional cross pieces being formed in front and behind by the front and main planes respectively. In addition to the ordinary wire cross bracing a wooden diagonal compression member is inserted into each half of the construction.

The usual type of fuselage used in biplanes is illustrated in Fig. XII., which is a sketch of that used on the Howard Wright Farman-type machine. The general construction is similar to that of the Valkyrie fuselage described above. The main planes are furnished at the points where the longitudinal members of the fuselage join them with ribs made much stronger than at any other part of the plane, and as these ribs carry at their forward ends the elevator outrigger booms, they are, in effect, part and parcel

of the fuselage. The manner of attachment of the vertical cross pieces to the longitudinals is shown in Fig. XIII., the joints being effected with a simple cast aluminium or welded steel socket, which is generally secured by eye bolts, to which are attached the ends of the diagonal bracing wires. This type of fuselage with unimportant modifications is employed on the majority of biplanes, but recently there has been a decided movement with the constructors of this kind of machine to adopt a fuselage similar to that of a monoplane. A notable instance of this is the Breguet, the fuselage of which is shown in Fig. XIV. This consists of two long channel steel girders, which take all the stress of the machine, the superstructure being provided for secondary purposes, namely, that of making, as it were, the pilot and passengers of stream-line form, the former sitting with his back against the first of the vertical panels. The cross pieces of the fuselage are also of channel steel, the gauge being throughout about 14. The superstructure is entirely covered in with fabric, and being composed of very thin wood, adds very little to the total weight of the construction.

In a new type of Bristol biplane, Fig. XV., the fuselage is similar to that of the Bleriot, and is notable for the rather neat way in which the cross pieces are attached to the channelled longitudinals. These attachments require the latter to be pierced only by a couple of small pins at each junction. The framework is braced up with diagonal wires in the usual manner.

The Sanders biplane is of the Wright type, and has a fuselage which also serves the purpose of landing skids. This consists of two skate-like members, each of which is built up as shown in Fig. XVI., and which carry at the rear the main plane and in front the elevator. This type of fuselage was originally made by Short Bros., and is characterised by the fact that stout steel strip is used for bracing purposes instead of the wires. It is, however, not difficult to see that no advantage can accure from this practice, as the head resistance of the strip must be decidedly greater than that of wire, and at the same time it is an extremely difficult matter to provide any means of adjusting the tension of these strips, while their weight also is somewhat excessive.

THE USE OF PRESSED STEEL IN AUTOMOBILE CONSTRUCTION.

Being a Paper read before the Institution of Automobile Engineers

By L. A. Legros, M.I.C.E. M.I.M.E.

By the term "pressed steel," engineers generally understand that the material has been worked into shape from plate or tube by means of stamping and drawing processes performed either while the material is hot or, in some cases, cold. The process is therefore one which usually changes the form of surface rather than the thickness of the material operated upon, and therein it differs essentially from the other stamping processes of drop forging and coining, in which the material is forced to take a shape quite different both in surface and cross-section from that which it previously presented.

Pressing in its simplest aspect consists in giving to a piece of material originally flat, a shape into which it can be directly developed, such as that of a portion of a cylindrical or conical surface; of this the common pen nib of cylindrical shape or a simple channel section cross frame may be taken as examples. The next step in the

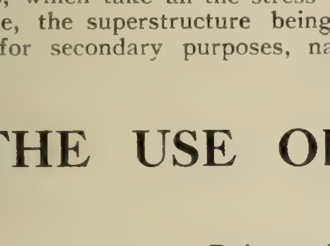


Fig. I.

evolution of stamping processes may be assumed to be that in which a plane surface is stamped by means of a convex punch and a plain hollow die into a convex form like that of an evaporating dish. This operation causes a change in the state of the material different from that caused by ordinary bending, insomuch as the resulting object produced has much greater stiffness than the blank from which it was formed.

Following on this operation the punch may be made to enter the die and carry a portion of the plate down with it into the parallel hole in the

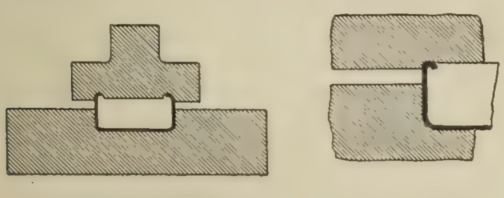


Fig. II.

simple form of die which has just been considered. In this case a short cylindrical section will be formed connected with, or tangential to, the spherical end of the object so pressed.

An example of this on a large scale is to be found in a dished boiler end-plate, though in this case the effect is generally - ' by forging or by continuously flanging round the plate in the hydraulic flanging press. Any attempt to press deep cup-shaped articles by the method just described will, however, result in failure, because of the formation of wrinkles due to buckling of the plate upon itself in its endeavour to accommodate the larger periphery of the blank to the smaller one of the die. Until a method of counteracting this wrinkling and preventing it from taking place had been devised, the press was

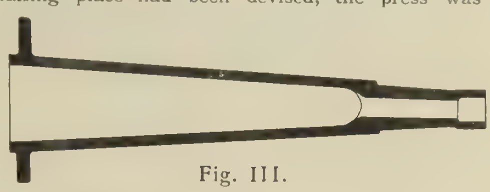


Fig. III.

very limited in its scope. So far as the author has been able to ascertain, no practical means of so doing was found much before 1860, if as early, and the honour of its invention cannot be definitely assigned to any particular person. Briefly, the method consists in the use of a hollow ram with an internal plunger (Fig. I.); the outer portion or ram bears upon the plate, and the inner portion or plunger draws the material down into the die while it is constrained against wrinkling by the end surface of the hollow ram which bears upon its upper surface and the top of the die which supports its under surface. This method is employed for a very large number of stamped articles in which the depth amounts to from one-

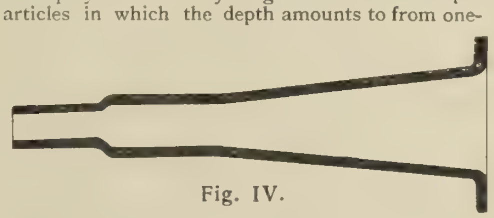


Fig. IV.

fourth to two-thirds of the diameter of the article.

The pressing of articles by this method is, however, obviously limited by certain conditions. If the tensile strength of the material is not sufficiently great to ensure the surrounding material being drawn down between the plunger and the die, the centre will be pulled out of the blank; or, if the edge of the punch be insufficiently rounded and the hole in the die be not large enough in diameter, the action will be one of merely punching a hole through the plate. Not only must the material have sufficient tensile strength and ductility to enable it to be drawn between the plunger and the die, but it must also have a sufficient margin of tensile strength to overcome the friction between the plunger and the die; thus, in attempting to press an article from a thin sheet, the condition may occur that as a single sheet it will fail in consequence of the addition of the friction on each of the surfaces bearing on the plunger and on the die respectively, whereas, if two or three sheets are treated simultaneously, the pressing can be successfully performed since the amount of the frictional load is in this case the same, but it is divided over the larger number of articles treated. Such a method is, however, only applicable to articles of which the exact size is of small importance, such as washing basins, cooking utensils, etc.

Curling.

When an article of cup-shaped form has been pressed, if the pressing be continued with a plunger formed with a groove of semi-circular section so placed that the edge of the article meets the commencement of the curve, the edge will be gradually forced over on itself until it forms a curled rim of semi-circular section, and if the pressing be continued, the rim will also be further pressed round and produce the same effect as if it had been bent round a wire (Fig. II.). This curling will only take place on articles of cylindrical, elliptical or somewhat simi-

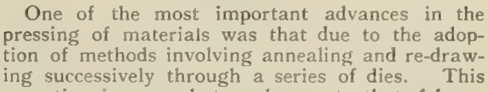


Fig. V.

lar forms because a plane plate would not be constrained by tensional or compressive stresses to follow the curve of the groove in the plunger. The curling may be effected either to the outside or to the inside of the object, the stresses in the material in the former case being tensile, and in the latter, compressive.

Drawing.

One of the most important advances in the pressing of materials was that due to the adoption of methods involving annealing and re-drawing successively through a series of dies. This operation is somewhat analogous to that of drawing wire. In the process of drawing wire, the

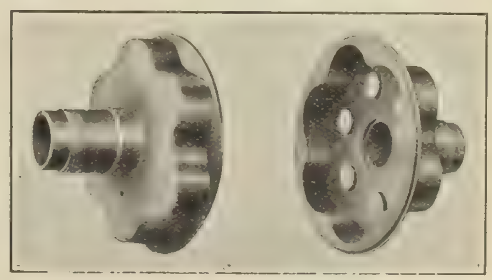


Fig. VI.

wire may, in some cases, pass through as many as six consecutive holes or drawing plates at one drawing. In the case of pressed articles, it is seldom possible to perform more than two or three operations of re-drawing without annealing. Amongst articles affording examples of re-

drawing may be cited ordinary cartridge cases, and the steel bottles for containing compressed gases.

Embossing.

Another process occasionally employed is that which may be termed embossing, in which a plate is placed between a male and female die and stamped or pressed to fit between them. This process permits of great rigidity being given

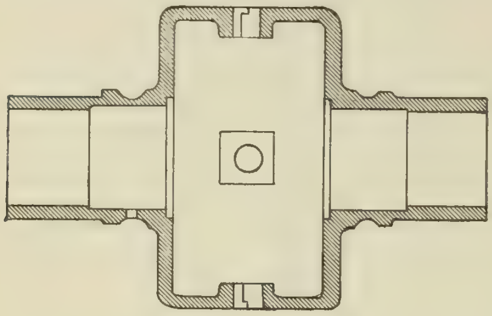


Fig. VII.

to articles which would otherwise be extremely weak from the absence of depth in their cross-section. An example of such embossing is to be found in the raised ring stamped in the lids and bases of the cans containing preserved food.

From the preceding brief survey of the methods employed in the production of pressed work, it will be noticed that it is easier to find examples in very light work and very heavy work than it is to find them in work of intermediate thickness such as is common in many parts of the automobile, but none the less the advent of the pressed steel frame for motor cars followed very soon after the freeing of the roads of this country to that vehicle. This was largely due to the fact that the pressing of steel frames for railway rolling stock had already received considerable attention.

The introduction of the pressed steel automobile frame by Messrs. Arbel in France was followed by that of a pressed steel wheel in which two dished discs were used in place of spokes. This wheel did not, however, appear to meet with general favour, possibly on account of the difficulty of obtaining access to the valves for pumping up the tyres. In more recent years the desirability of lightening the reciprocating parts of the engine has led to the adoption of the pressed steel piston, which, first used on racing cars, has since become very general on standard types of high speed engines. The influence of rating formulæ, and of a taxation based upon a formula which takes no account of piston speed, is likely still further to increase the employment of such pistons.

A combination of pressing and drawing is commonly employed for parts of the back axles, since the chain drive has been displaced from popular favour by the bevel or worm drive, both of which usually employ hollow axles constructed with central driving-gear cases. This casing which encloses the bevel or worm gears is also frequently formed by pressing it in two portions secured to each other and to the tubular flanged ends of the outer axle or of the axle casing.

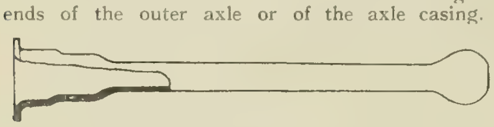


Fig. VIII.

The brake drums of the back wheels and their cover plates are now frequently formed by pressing, as are also a number of small details. The processes of pressing and drawing in general are by no means so largely employed in automobile construction at present as might be anticipated from their numerous applications to articles of everyday use. A notable example of the use of light pressed steelwork is to be found in the pressed steel panels used by some firms producing cars with a standard pattern of body; but the number of makers who have adopted this method of producing strong, cheap and durable body work is at present extremely limited. In this paper the author proposes to give examples of some of the best existing practice in pressed work for the various parts named, and to indicate briefly the direction in which he thinks the use of pressed steel may be further introduced into the construction of the automobile vehicle.

In consequence of the change of form to which the material is subjected by pressing in dies, it follows that the shape of the blank from which the pressed steel work is produced must be different in many cases from that obtained by developing the surface of the work into a plane. This is more particularly the case where the object pressed is of irregular outline and has varying radii of curvature. In such cases allowance must be made for the lengthening or shortening of the raised portion of the plate under the stresses introduced. The theory of these changes has not been very thoroughly investigated, and the determination of the correct form for the punch and die for cutting out the blank from which the stamping is to be produced are therefore largely questions of experience. It is obvious that accuracy in the form of the die will save considerable work in trimming and finishing the pressed work, and further, that once the form of the blank has been decided, it is not possible to make alterations of shape without constructing either a new punch or a new die as the case may be, to suit the alterations in the other member of the pair.

Chassis Parts.—Frames.

The most important instance of pressed work in automobile construction is the steel frame, of which a large number of designs are in general use. In the simplest form, the steel frame consists of two side members connected by four or five cross-members. With this form of construction the engine is usually carried direct on the main frame. Another form of frame is that in which a supplementary frame extends over the front portion of the car carrying the engine and clutch gear. This frame is supported at its ends on cross-members connecting the two main side frames. Owing to the fact that in nearly every case the front of the frame is narrowed to allow for sufficient locking of the front wheels, it will be seen that a large number of dies are required for the production of even a single type of chassis. The main frames are right and left-handed; the

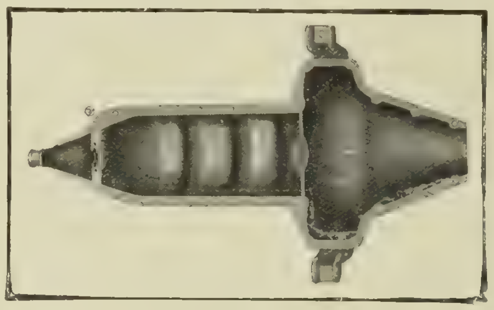


Fig. IX.

same applies to the supplementary frames if these are bent unsymmetrically either to their length or depth, or to both. The cross-members require separate dies for those connecting the narrowed front of the frame and for those for the wider back portion, and the cross-members connecting the back portion of the frame require to be different according to the position they occupy in the decreasing depth usual in the back of the main frames. In many cases also the cross-members are dipped to give the requisite clearance for parts of the driving gear.

Not only do the outlines and plans of the various forms of frames differ very widely amongst themselves, but the thickness of plate employed for them varies from ⅛ in. to ¼ in. (3 to 6 mm.) for ordinary cars, and from 3/16 to ⅜ in. (5 to 10 mm.) for commercial cars and lorries. It will therefore be seen that although the tools for pressing frames may be built up of several parts secured to a bed, it is necessary for the manufacturer to be provided with a very large number of die parts in order to cope with the varying demands for this class of work, and usually some special dies will have to be made to suit the individual requirements of any particular case. In the construction of frames it is very usual to make the cross-members of slightly less thickness than the main frames.

Back Axle Casing.

Next in importance to the frame may be taken the portions of the back axle casing, which with the displacement of the chain drive by the bevel drive in the case of touring cars, has led to the necessity for obtaining a back axle casing of light construction with a view to reducing the unsprung weight to as low a limit as possible.

The commonest component of this class is the half axle casing of tubular form usually of increasing diameter towards the centre of the vehicle, and terminating in a flange with or without a spigot end (Figs. III. and IV.) for registering in the central portion of the casing. These tubes, with the central casing, form the actual axle on which the load is carried, they are machined where they make the joint with the central casing, and also on the outer external end for receiving the spring pads or spring bearings and the ball bearings of the hubs. The bell-mouthing at the centre-line of the car gives a greater base for support on the central casing to which they are generally secured by bolts, although in some cases rivets are used. In the case of riveting it should be remembered that although the rivet is an excellent form of attach-

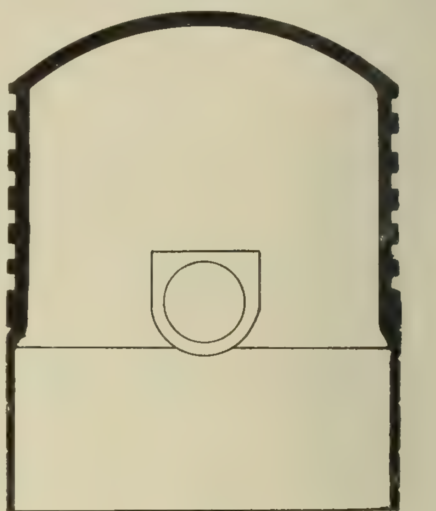


Fig. X.

ment where it is subjected to shearing stresses alone, yet it is not to be trusted when subjected to tension, especially where it is of an intermittent kind accompanied by shock. For this reason some forms of pressed steel composite back axle casings in which the axle tubes are connected by riveting to the central portion, appear likely to prove unsatisfactory if run over rough roads and thus subjected to shock. In the author's opinion it would be far preferable to avoid the use of rivets for connecting any of these parts of the back axle together.

The differential outer casing is now frequently made of pressed steel, although it is of a form which requires expensive tools. By carefully considering the design, however, it is usually possible to make one set of dies available for both the right and left-hand sides of this casing. In some cases a plug-hole is required for the purpose of filling with oil, and this can either be so arranged that it is present on both sides, or it may be formed subsequently, in which case it is only formed in the one half. In a few instances an internal bracket is carried for supporting the end of the propeller shaft, this bracket being also of pressed steel and carried from one side of the casing. A casing of this type requires to be faced not only to make joints with the conical axle tubes, but also to make joints between its two halves, and, further, the cylindrical flanged end must be faced to make an oil-tight joint with the propeller-shaft casing.

In some few cases pressed steel has been used

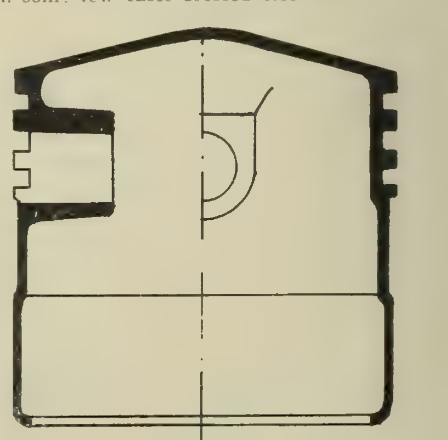


Fig. XI.

for differential boxes, particularly those in which the spur pinion type of differential is used. An example of these is given in Fig. VI., while a mitre-gear casing is given in Fig. VII.

Propeller shaft casings may be divided into two main classes, those in which the cardan-joint end is parallel (Fig. V.), and those in which it is flanged or formed of spherical shape for carriage in an appropriate bearing (Fig. VIII.). In the former case the propeller-shaft casing does not act as a radius rod, and the propeller-shaft

usually slides in the cardan-joint. In the latter case the propeller-shaft casing acts as the radius rod, the thrust when the engine is driving, or the pull when using the brakes, being transmitted to the chassis through the spherical end of the casing.

In the United States specialisation in the manufacture of automobile parts is carried so far that some firms only manufacture certain details. Typical of these is the Collins back axle, many of the principal components of which are formed of pressed steel. The axle is an example of the combination of the gearbox with the back axle bevel and differential gear; unlike most of the European patterns, it is pressed in two halves extending over the complete length from wheel to wheel, and these halves are welded together electrically from hub to hub along the section formed by a vertical plane passing through the longitudinal axis of the axle. An inspection door is provided on the back half of the axle, secured to it by studs and nuts, and a similar mode of attachment is employed for the propeller-shaft casing on the front of the axle. The position selected for the weld is that in which the lowest stresses may be expected, and from the constructional point of view, the design of such a casing appears to be preferable to that of a casing in which flanged tubes are secured to the central portion by rivets. The Collins axle is also adapted for chain drives by usnig the same casing and many other standard parts, but with the difference that chain sprockets are fitted at

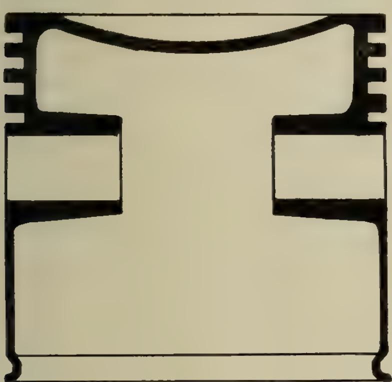


Fig. XII.

the end of the "axle" in place of the brake drums and hubs, though, of course, the gear ratio in the casing is varied to suit the altered conditions. In this application the "axle" occupies the position of the ordinary gearbox and sprocket brackets, the wheels being carried on a plain axle which is connected to the Collins axle by pressed steel radius bars fitted with the usual adjustment.

Gearboxes.

The author has not been able to find any good example of the application of pressed steel to gearbox construction, although the size of the gearbox is such that one would have imagined the construction of pressed steel boxes would have become fairly general in the case of those cars which are turned out in large numbers. There is, however, a difficulty in the case of the gearbox which is not present in that of the differential casing, and that is the trouble occasioned by the approximately rectangular form of the box at the joint between the upper and lower halves. A form like this renders the plate liable to buckling at the portion which forms the flange, and it is difficult to so secure the two halves together as at the same time to make a perfectly tight and satisfactory joint. A method which is adopted for some pump casings and other like objects of awkward shape, consists in welding or brazing the pressed metal flange a supplementary thickness of metal which affords the necessary reinforcement for permitting the joint to be machined and to be made oil-tight when bolted up. Judging by the success obtained with pressed steel for the details of the back axle, which in general are subjected to heavy stresses and to greater shock than the gearbox, it is somewhat remarkable that this detail of the car should not have received greater attention from the makers of pressed steel, in spite of the difficulties attendant thereon.

Clutch.

In the case of leather faced clutches, the rotating member has frequently been made of an aluminium casting; in this case, again, pressed steel has found its place in several designs.

The requisite stiffness for taking the thrust of the spring is obtained by making the disc dished to conical form, and where the cone has a very obtuse vertical angle, additional stiffness is obtained by embossing ribs on the disc so as to increase the effective depth of cross-section of the metal.

Engine Casing.

In very few cases up to the present has the use of pressed steel been adopted for engine cases so far as the author can ascertain. One example

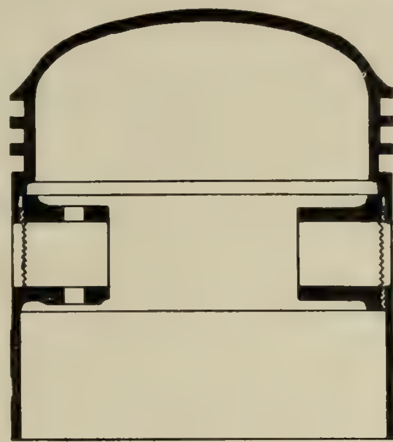


Fig. XIII.

which, so far as the author is aware, has not at present gone beyond the stage of a patent specification, proposes the construction of the engine casing in three sections, an upper casing, a central member carrying the crank shaft bearings, and a lower section forming the crank pit, the two upper sections being connected together by means of distance pieces and stay bolts. An example of a combined engine and gearbox casing in pressed steel is that of the Ford car, in which the cast iron parts have been greatly reduced in number. This casing (Fig. IX.) is an example of what can be done in pressed work of complex outline but having a one plane joint.

Pistons.

Pressed steel pistons are now made in a large number of sizes ranging from 1.8 in. to 10 in. (47 to 250 mm.) in diameter, and in a very large number of patterns, the chief variations in which may be classed as follows :—(a) form of top, convex (Fig. X.), obtusely conical (Fig. XI.), flat or concave (Fig. XII.); (b) arrangement of grooves to suit from three to six rings; (c) arrangement of one groove to effect locking of the gudgeon pin; (d) arrangement of one groove for a ring below the gudgeon pin to prevent excessive passage of lubricant; (e) an arrangement of oil-catching grooves on the external lower portion of the piston (Fig. X.); (f) arrangement of an oil-catching channel on the interior of the piston at the bottom end; and (g) variation in the arrangement and form of the

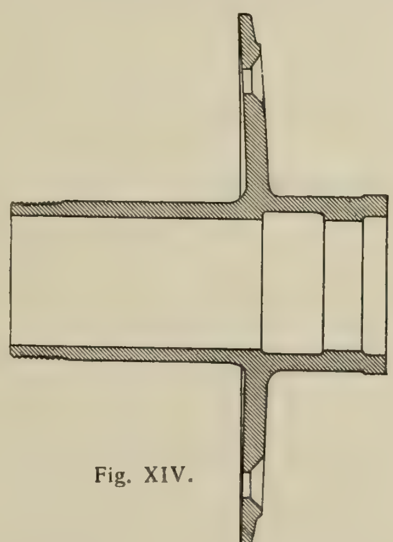


Fig. XIV.

lugs, or bosses, for securing the gudgeon pin in place.

The great importance of the pressed steel piston is evidenced by the large number of patterns and styles made (over eighty are illustrated in various manufacturers' catalogues). It is, however, generally formed by a process more closely akin to that of drop-forging than that of pressing proper of the kind previously described and considered. This is rendered necessary by the presence of the large internal bosses for carrying the gudgeon pin. In order to permit of the withdrawal of the die from the interior of the piston, it is necessary in forging to carry these bosses up to the head of the piston, and subsequently to machine away the interior so as to leave the bosses with a plane upper surface left by the tool used for boring. A piston showing the bosses finished to this form is shown in Fig. X. For purposes where extreme lightness is sought, however, the boss is turned cylindrically on its outside by means of an overhung cutter carried in a bar passed through the gudgeon pin hole and advanced towards the piston shell. In consequence of the small diameter of the interior of the piston and the relatively large diameter of the outside of the boss, it is obvious that the overhung tool cannot be made to approach very closely to the piston shell at the section through the gudgeon pin axis normal to that of the piston. To enable the cutter to remove all possible superfluous metal and terminate the boss with only a small radius connecting its external cylindrical surface with the internal cylindrical surface of the bore of the piston, a pumping motion is sometimes employed on the boring bar to effect the removal of this small amount of metal; the extra amount removed by this additional complication in the machining operation is, however, extremely small. In the majority of designs of steel pistons available, it will be found that the top is made appreciably thicker when it is of flat section, the thinnest sections being those of considerable convexity. The thickness usually adopted for the top of

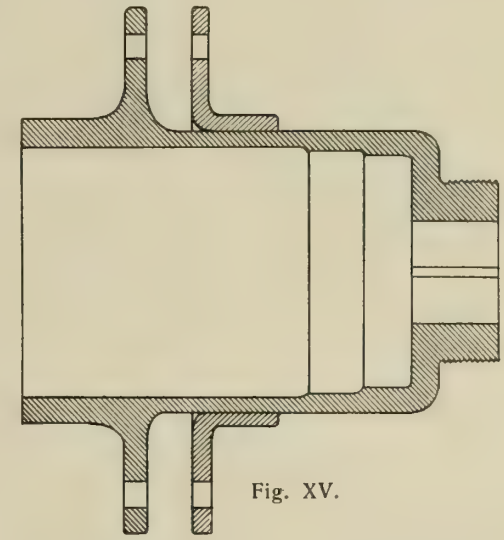


Fig. XV.

pistons from 3.5 in. to 5 in. in diameter is $\frac{1}{8}$ in. (3 mm.); from 5 in. to 6.5 in. it increases gradually to about 3/16 in. (5 mm.) and in pistons as large as 10 in. in diameter it will attain as much as $\frac{3}{8}$ in. (10 mm.).

In many cases the diameter at the top of the piston is made slightly smaller than that of the lower part of the shell in accordance with the customary practice adopted for cast iron pistons.

The thickness of the top of the piston may be taken as the ruling thickness in most cases, the thickness of the metal at the bottom of the grooves for receiving the piston rings being usually made equal to it. For pistons from 3.5 in. to 5 in. diameter the depth of groove for the piston rings usually amounts to about 3/16 in. (5 mm.), and where a distance occurs between the lower piston ring and the top of the exterior of the gudgeon pin boss, the thickness of the shell at that part is usually reduced to the ruling thickness. The thickness of metal in the bosses is generally from 3/16 in. to $\frac{1}{4}$ in. (5 mm. to 6 mm.) for the sizes just considered; the combined length of the two bosses is usually from 40 per cent. to 55 per cent. of the outer diameter of the piston. The provision for securing the gudgeon pin in some cases takes the form of an auxiliary boss stamped on the lower face of the gudgeon pin bosses towards the mouth of the piston. In some cases these bosses are made deep enough for tapping to receive a locking screw, but usually no such provision is made. In some special cases the bosses are made conical for receiving conical locking bushes for retaining the gudgeon pins in place.

Below the gudgeon pin bosses the piston is usually reduced to a very thin shell, stiffened in some cases by internal circular ribs left in boring, and grooved at the corresponding external parts to provide means for arresting the upward movement of the lubricating oil. In some cases these grooves are very narrow, while in others they attain a width of as much as $\frac{1}{2}$ in. In three

instances only has the author found examples of manufacture in which the finish of the piston mouth resembles that which would be obtained if inward curling of the metal were employed; in each case the thickness of the metal was 0.04 in. (1 mm.) and the radius of the curl about $\frac{1}{8}$ in. 3.5 mm.), in some other instances of pistons up to 0.06 in. (1.5mm.) thick, a finish is adopted equivalent to curling to the extent of bending the metal through a right angle, that is to say, one quarter of the circle is formed in this instance while one half of the circle is formed in the three preceding cases. It would appear to the author that the use of the curling process for stiffening the lower end of very thin steel pistons has hitherto not been adopted in many cases to which it was distinctly applicable, and in which it would have formed both the lightest and the most economical method of finishing the piston.

A form of steel piston consisting of a shell produced by pressing and forging, into which is screwed a drop-forged ring carrying the gudgeon pin bosses, is shown in Fig. XIII.; it has already been mentioned in the proceedings of this Institution. The advantages of this form of construction lie in the reduction in the amount of machining and in the possibility of varying the amount of compression (by altering the distance from the gudgeon pin centre to the top of the piston) should it be found necessary to make a

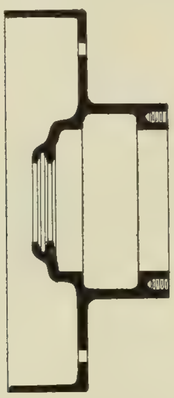


Fig. XVI.

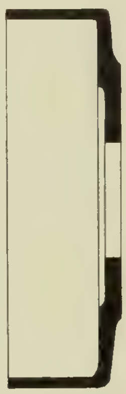


Fig. XVII.

change in this dimension after trial of the first engines of a class. A modification of this piston has been adopted by the Iris Motors for their aeroplane engines. In this instance the piston head above the gudgeon pin is pressed and the lower part with the gudgeon pin bosses is drop-forged. Where the length of shell below the gudgeon pin is short this method may prove cheaper than that shown in Fig. XIII.

The only other engine detail of any importance at present made by pressing appears to be either the complete fan, 14 in. to 18 in. in diameter, made in one piece either right or left-handed, sometimes with the fan plate stiffened by embossed ribs.

Hubs.

A large number of patterns of pressed steel hubs are available, but these again, like the piston, come more properly under the head of drop-forgings than of pressed steel work, since in many cases they are forged solid and bored out. While this applies to such parts as the main portion of the front wheel hub (Fig. XIV.), and also to many forms of back wheel hub, the loose outer plate for each, which secures the steel spokes or steel boss in place (Fig. XV.), is frequently made by a process much more closely allied in pressing, since the change of thickness of the metal from one part to another is but small, and the material does not undergo so much alteration of form in the pressing as is the case with the hubs themselves. The hub caps are also frequently made by pressing and drawing processes, though in this case the author has not met with an instance of pressing in steel.

The hub, when of large internal diameter, is in some cases made in one with the drum brake (Fig. XVI.). In other instances, however, it is preferred to make the drum separate, in which case it can be produced very nearly in its finished form by pressing alone (Fig. XVII.).

Dust Caps and Small Details.

For protecting cardan-joints and other details of like character, a thin pressed steel case is frequently used. For this class of work pressed steel can be made lighter than a casting, and at the same time it can be finished with the minimum of machining.

In a few cases radius stays have been made of steel plate about 5/32 in. (4 mm.) thick, pressed with a raised surrounding edge, and lightened with holes punched through and pressed up at the edges to give additional stiffness in the same manner as mentioned above in the case of some side frames.

The advent of Carter's chain case had such an important influence on the use of chains in bicycles that it is very strange that it should not have been adopted on automobiles at an earlier period of their development. The use of the chain case for cycles was so well known on the Continent some years prior to 1896, that the word "Carter" had become generally used by French engineers as a convenient term for any kind of oil-containing casing. With the continued employment of chains in the design of commercial vehicles, the chain casing (Fig. XVIII.) is coming into more general use, and is now a definite feature in a large number of vehicles of this class.

Amongst the minor details of car construction made by pressing can be cited the inner and outer discs for plate clutches, some of which are stamped with internal teeth and holes for receiving distance pins, while others are stamped with slots and internal teeth for engaging with the clutch-box, and for avoiding those difficulties which arise from the buckling of the plates when heated, or for preventing the rolling action of small pieces of abraded metal. Another detail, though one which does not take a very prominent place, is the small cover plate used to protect such places as the ends of the gearbox shafts; where the bosses are of such form that they project externally the cap can be made of cup form, instead of the more usual dished plate.

Steel Wheels.

Some seven years ago Messrs. Arbel, of France, proposed a steel wheel consisting of two dished plates connected at the circumference, and separated by a boss at the centre. These do not appear to have found favour, probably owing to difficulty in changing the tyres, or in inflating them properly, as has been mentioned. A somewhat similar wheel has recently been designed, and is manufactured by the Harris Wheel Co., in which provision has been made for bringing the valve to the outside of the discs, while at the same time the construction has been made such that the whole of the tyre, with the rim, can be easily removed. This form of wheel enables a spare rim to be carried.

The Sankey wheel, which is made of two pressed steel halves, welded over the vertical section normal to its axis, is at the same time a detachable wheel, and one which is completely made of steel. The wheel is similar in appearance to the ordinary wooden wheel, with the exception that the spokes appear somewhat lighter, and are connected by a somewhat larger radius at the centre, next the hub, than is usual in the case of wooden wheels. The centre of the hub is filled with hard wood packing to take the pressure of the securing nut. According to the makers' tests, the wheel is not only lighter than a wooden wheel, but capable of

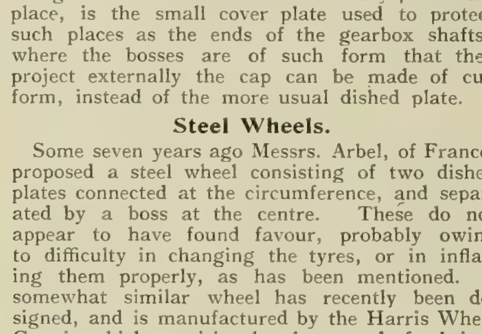


Fig. XVIII.

standing from 50 to 60 per cent. more load without collapse. At the same time, wheels which have suffered extremely rough treatment appear to have shown an immunity from that total collapse which would have been expected in the case of a wooden wheel subjected to the same shocks. The reproductions given in Fig. XIX. and Fig. XX. show two of these wheels after serious damage, and, in the author's opinion, they are of greater interest than an illustration of the undamaged wheel.

Carriage Work.

The most important use for pressed steel in carriage work is for the panels, and in particular for those panels which have double curvature and consequently great stiffness. Several makers of pressed steel now supply these panels in such form that any common type of body can be almost entirely constructed with their aid. Not only are the bucket or front sets pressed with a bottom flange in some instances, but pressed panels are made of double width for the front and back seats of torpedo bodies, and also of double and treble width for the back seats built up in two or three pieces each; in fact, a sheet metal body, suitable for a taxicab, can now be obtained almost complete.

Next in importance after body panels may be taken the mudguards. The earlier form of mudguard in which a plain sheet with a raised bead and rolled edges was used, afforded but little protection from the weather, and in many cases it was necessary to fix an additional strip of angle section, known as a vallance, round the edge of the front mudguard in order to prevent the mud from being thrown off the corner of the guard into the faces of the passengers riding in the front seats. By pressing the steel guards it is possible not only to obtain curved sections for the finished muguards obviating this inconvenience, but the mudguards can be made integral with that portion of the guarding which connects them to the chassis itself and prevents

Fig. XIX.

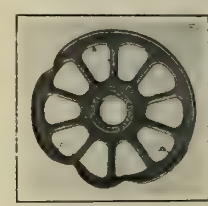


Fig. XX.

splashing through on to the bonnet and dash, in the case of the front mudguards, and on to the back of the body in the case of the rear mudguards. The use of detachable wheels has to some extent influenced this question, as it has enabled the mudguard to be approached more closely to the wheel than was the case where the tyre had to be replaced *in situ*.

Step brackets are also now made of several patterns provided with a flange for attachment to the chassis, and made of channel section where they support the floor board.

Conclusions.

In conclusion, the author desires to draw the attention of the members of the Institution to the large increase which is taking place in the use of pressed steel, particularly in those cases where parts are required in large numbers, but at the same time it is his opinion that much of the pressed steel work at present used has not that accuracy and truth which is to be found in smaller and lighter pressed articles, such, for example, as the casings and other details of cheap watches and clocks, the parts of Welsbach gas burners and of many lamps, the cases of cartridges, and a large number of other metal articles produced in large quantities and frequently in their finished form to a high degree of accuracy. Machining as a means of obtaining a finished result from pressed work involves the addition of relatively lengthy and costly operations on a piece of work which up to that point has cost but little to produce, apart from the first cost of the dies and press employed.

The connection of pressed steel parts together by their flange or by welding has proved in many cases a matter of difficulty, and it appears that much remains yet to be done with a view to facilitating and perfecting the mode of joining thin stamped or pressed work to stouter flanges or to other parts made by similar processes. At present the methods employed for welding, whether by electricity or by the use of acetylene and oxygen, or by other methods, are not such as the constructional engineer generally considers to be thoroughly trustworthy, and, for this reason, the joining of such parts in places where stresses are heavy is usually avoided. Some of the operations employed with so much success in the production of small and light work, such as re-drawing and the finishing of thin edges by curling or beading, appear to be but little employed at present, though it is obvious that their application would in many cases give increased stiffness and a better finish to the objects produced. The process of spinning which is performed largely on thin metal is at present almost unused in metal of such thicknesses as those employed in automobile parts, i.e., from $\frac{1}{8}$ in. to $\frac{1}{4}$ in. in thickness. Some of the softer kinds of steel which are capable of being pressed to form could probably be further modelled to some extent by a combination of pressing and spinning, the operation in the case of steel not necessarily being performed cold, but while hot and by means of a roller pressed against the revolving work.

It appears to the author that the employment of methods for pressing ard otherwise forming thin sheet steel to appropriate shapes will prove one of the most important factors in the production of light and cheap motor vehicles, provided always that the number of any given pattern required is sufficiently large to warrant the outlay on plant and tools for pressing. In other words, the American methods of manufacture may well result in the production of cheap vehicles of light weight having even greater strength than those produced by older methods, with the possibility of a much lower cost, provided the market available be sufficiently great.

A brief account of the Discussion.

At the close of the paper the author exhibited a number of examples of pressed steel work, one specimen in particular, to which he directed attention, being a case enclosing a cheap watch. He thought automobile engineers could learn much from the way in which it was turned out; it was so finely and accurately finished that it required no machine work and practically no hand work after it had been pressed into shape. The whole watch and case complete was sold retail at 3s. 6d.; and the price at which the article left the works would be about 1s. 9d.

Mr. F. W. Lanchester, who presided, remarked that this small article was worth coming to see, if all the other pieces of work, many of which were excellent, had been thrown in gratis.

In calling on Col. Crompton to open the discussion, the Chairman said they had already seen some remarkable examples in this country of what the Americans had done in the way of cheapening motor cars, and that had been largely due to the introduction of pressed steel work in those parts of the chassis where we in this country had not found the quantities required large enough to justify the use of that process.

Prof. Tucker said the best class of steel for pressed work was precisely that which lends itself to pressing, because the strains which led to fracture in practical working would of course bring about failure in the pressing itself. He had been associated during the past six months with matters correlated to the subject of the paper. First, as to the Collins Axle, he was aware that it was being made at Wolverhampton at the present time, and it seemed to him a very beautiful construction, not only as to lightness ensuing from the process of manufacture, but also from the neatness of the design, and great strength. All this was brought about, in his judgment, by the merit of the welding. Then again, in connection with frames, the use of the cutting blow-pipe where hydrogen was used, was an immense advantage in cutting out contour lines, and the cost was infinitely less than that for which it could be done by cutting with the cross cut, and drilling, or by other obvious methods. Successful welding with acetylene and successful cutting with hydrogen or acetylene, depended almost entirely on the purity of the oxygen. During the last few months he had obtained some interesting results in connection with that point. As to motor works practice, he found if the percentage of oxygen in the commercial gas was reduced from 99.5—at which point it could be purchased in the open market —to 97. the effect on the welding and the rapidity of the cutting were very strongly marked. He was aware that cutting by acetylene and also by hydrogen were perfectly well known operations, but in view of the paper which Mr. Legros was to read that evening he had brought along a sample which might interest members—an undercut six inches deep which had one surface not much inferior to milling. One side of the piece of steel showed the result of using a 99.5 per cent. purity oxygen, and the other a 97. per cent. He specially called attention to the latter. In the old days it would have been impossible to cut more than three or four inches. Referring to gearcases, be quite anticipated that such parts could be very easily welded and made up.

Prof. Sharp said he had had some experience in designing pressed steel work. He thought that steel makers did not quite realise what were the best materials, or what was the best specification for the materials to be used in the process of pressed steel work. The physical properties of the material needed to be studied more closely. It was a matter of the greatest practical importance to get the best material for the purpose, steel that had the greatest ductility for pressing and the necessary strength to withstand the rough treatment it was subjected to in use. Again, he thought the designers of tools for pressed steel work were working mostly by rule of thumb instead of going to the fundamental principles underlying the practical operations involved. It ought to be possible for a designer to judge what his material might be expected to do. "If steel makers knew our requirements (he said) they would be able to produce something we have not yet had from them."

Mr. H. Johnson thought pressed steel should be more largely used in making such things as lockers, drawers and wardrobes, instead of making them of wood. The advantages were obvious in large works. He also mentioned the Carter case as a great success on the Sunbeam car.

In replying, Mr. Legros thanked all for the kind way his paper had been received. With regard to the question of acetylene welding and electrical welding mentioned by Mr. D. J. Smith, he said electrical welding required great pressure to get a good grip of the material before the parts could be got into sufficient connection, and could not be done so satisfactorily. As to the question of the quantity to be turned out in a batch of any particular part, that was purely a question of the expense of the dies. In the case of a back axle, for instance, the dies were very expensive. He mentioned that he had received a letter from Mr. Hamer Read, who was laid up and unable to come. In it he said he regretted that he had not learned the art of the copper smith, as that had a great bearing on this subject of steel pressing, for the beating out of copper by the slow and tedious process of hand work much resembled what had to be effected in pressing steel by the more rapid mechanical processes. It might not be generally known, but a large number of motor frames had been pressed cold, and even hammered out cold. The limited range of the stress to which the steel was subjected in ordinary work was not sufficient to make it crack, even in such cases.

CORRESPONDENCE.

No responsibility is accepted by the Editor for the opinions of correspondents, and the right is reserved to omit any portion of a letter if it is considered desirable to do so. Letters to be inserted with a nom de plume must be accompanied by the name and address of the writer. If publication in the next issue is desired letters should reach the Editor in no case later than the 20th of the preceding month.

TOOLS AND FIXTURES FOR MACHINING PISTONS.

Sir,—I have read with considerable interest the article on above subject by Mr. D. Walters in your April issue. It is certain that no part of an automobile engine requires greater care in its manufacture than the piston, and the methods described by the writer of the article would doubtless produce the desired result, but I am sure that the work could be done in *one-third* of the time quoted and a superior job made, if it was

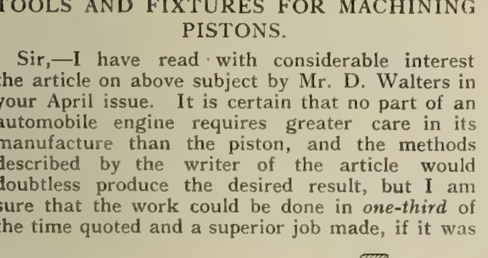


Fig. I.

handled in the turret lathe, milling machine and grinder, in the manner which I will endeavour to describe.

Pistons of modern engines are made so extremely light that it is absolutely essential to so chuck them in the turret lathe that the outer surface when turned shall be truly concentric with the core, or in other words that the walls of the piston shall have an exactly equal thickness all round.

To secure this desirable result, the method described by Mr. Walters of chucking in a 3 or 4 jaw chuck for the first operation could hardly prove satisfactory unless much time was spent in setting the piston by its inner surface with chalk marking or other tedious process.

The chuck shown in Fig. I. will centre the piston perfectly by the inside face of the casting. It is attached to the spindle nose of the turret lathe and is operated by a rod and hand wheel at the rear end of the hollow spindle. The inside walls of the pistons are held truly central by six hardened pins of rectangular section (two of which are shown). The front set of pins is expanded by the conical end of the operating rod, and the rear set by the sliding nut having inclined slots in which the pins are seated. The rotation of the piston is accomplished by two slots cut in opposite sides of the chuck body and which drive the piston by the gudgeon pin bosses,, thus relieving the pins of any work except that of centreing and supporting the walls of the piston.

The *first operation* is to turn the outside of piston roughly within .020 in. of finished size, face top end, either flat or curved as required, cut grooves for rings and drill a centre hole in the end, the purpose of which will be described later on. Time 12 minutes.

Second Operation.—Reverse piston, holding same lightly in soft jaws of 3-jaw universal chuck in turret lathe. Bore out mouth to finished size, and face lower end. Time, 4 minutes.

Third Operation.—Chuck piston on angle plate chuck, and bore gudgeon pin hole .003 in. to .004 in. under finished size. Time, 5 minutes.

As the gudgeon pin has to be a very close fit, I have not found it satisfactory to finish reaming the

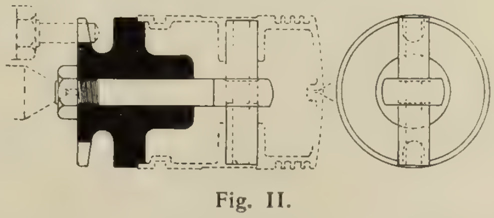


Fig. II.

holes in the lathe as the slightest error in alignment will distort the finished piston. A hand reamer made with pilot and follower bar will assure these holes coming exactly to size and lineable and the operation can be, and is, better left to the assembler.

Fourth Operation.—Hold piston in fixture, mouth upwards, in a vertical spindle milling machine, and face inside of gudgeon pin bosses with a long milling cutter of suitable diameter. A spacing bar could be arranged in connection with the fixture to locate the faces central and of correct distance apart. Time, 5 minutes.

I hardly think the attachment described by Mr. Walters is necessary for this simple operation, as it can be done readily with standard cutters.

Fifth Operation.—Grind outside of piston to finished size. Time, 3½ minutes.

The grinding is best done on dead centres, and a grinding arbor made as shown in Fig. II., should be used in duplicate, one being loaded whilst the other is in the machine. With regard

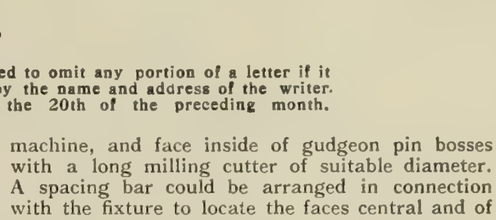


Fig. III.

to wheels, a 24 combination Norton Alundum Wheel, either Grade J or K will be satisfactory. The K wheel will give the finest finish, and the J wheel will cut the faster with slightly coarser finish. It is obvious that the extraordinarily long time (30 to 35 minutes) given by Mr. Walters for his grinding operation is partly due to the incorrect wheel which he has been using, viz., 60-M. This is much too hard and fine. On Norton plain grinding machines, cast iron pistons 3 to 4 inches in diameter are regularly ground in from 2½ to 3 minutes each. The accompanying sketches (Fig. II.) illustrate a few samples from actual practice.

I would like to point out that the amount of stock left for grinding should be from .020 in. to .025 in. on the diameter. This enables time to be saved on the lathe work, as a rough turned finish will give just as good results as a fine one when the work is completed by grinding.

CHAS. H. REYNOLDS.

A ROTARY STEAM ENGINE.

An Entirely New Design for a Rotary Engine, possessing many points of especial ingenuity.

NOTWITHSTANDING the number of years during which inventors have endeavoured to evolve a satisfactory design for a rotary steam engine, the reciprocating engine still holds its own for all but quite special work, and, in a very broad and general way, one might say that neither turbines nor other forms of rotary steam engines have proved satisfactory for small powers. The design described below has many points of peculiarity and seems to have escaped the drawbacks common to most rotary engines, while it appears to possess certain features which might make it particularly suitable for use in connection with steam tractors or launches. It is also within the bounds of possibility that it could be used for touring car work. Fig. I. is a section through the centre, A being the rotor, and B and C two other revolving pieces, while the surrounding case is stationary. D, E, F, G are exhaust ports, paired diagonally, one pair being opened when running forward, and the other pair when reversing. It will be observed that the main rotor A carries two vanes in close juxtaposition to two slots somewhat of dovetail form, and it is from these that the steam enters.

Before proceeding to describe the exact method of operation it would be well to explain the purpose for the subsidiary rotors. These are driven at the same speed as the main rotor by external gearing (revolving, of course, in opposite directions) and there is a shallow, though broad, channel in the outer casing behind each of these secondary pieces, which is just discernible in the illustration. The live steam, which normally lies in the ducts and passages, to be described hereafter, also fills these spaces, which are proportioned so as to press these "reaction rollers," as they may be called, steam-tightly against the rotor. The purpose of these reaction rollers is to do away with sliding friction, and the inventor states that there is no actual contact of metal with metal throughout the whole of the inside of the engine. The vanes, for instance, on the main rotor do not actually touch the case, but are made a very fine floating fit. That there is little or no internal friction is proved by the behaviour of a small experimental engine which delivering about 10 h.p. with eighty pounds of steam is capable of running well over 4,000 r.p.m., and maintaining this speed without noticeable heating.

The admission of steam is controlled through one of the end plates shown in Fig. II., which is, unfortunately, not too easy to follow, but with its aid the *principle* can be explained clearly. The end piece has a number of passages cast in it, all joining to the main steam pipe or the reverse steam pipe, as the case may be, at one end, and to ports cut in the inside face of the end plate at their other extremities. These ports register in turn with different openings in the rotors, so admitting steam at the correct moment for whichever direction of rotation is desired. Assuming for a moment that the intention is to go ahead, then the main steam valve is opened and steam enters the forward passages in the end plate. With the small engine it is possible to hold the rotor by means of the crow-bar, and then nothing happens except a slight leakage of

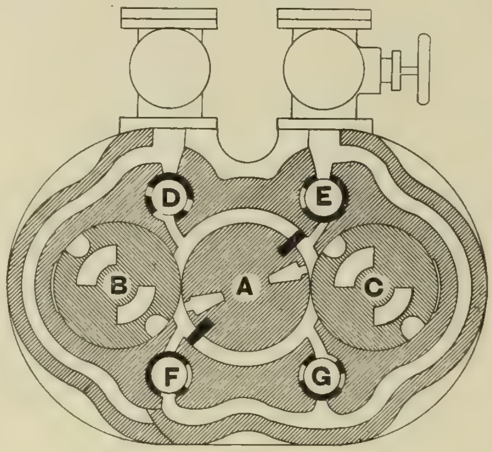


Fig. I.

steam, so small as to be scarcely noticeable. When one of the dovetail shaped openings in the rotor comes opposite a port in the main plate, the steam shoots down the passage offered to it, and thence out through the slot in the rotor until it fills the space between the nearest vane of the rotor and the adjoining secondary rotor. The sectional view in Fig. II. indicates the manner in which the steam passes out into these spaces. Of course, the rotor at once commences to revolve and steam at full boiler pressure is supplied as long as the two ports register. Those in the end plate, however, are so positioned that the supply is cut off at a pre-determined distance in the stroke, and the steam then expands, still driving the vane before it until the exhaust port is uncovered.

So far the operation is quite easy to understand, but the reverse is a little less easy to describe. First of all it must be remembered that the valves D, E, F, G are exhaust valves only, and the engine is so timed that if one particular pair are open the rotor revolves in the forward direction. To reverse, steam is no longer admitted through the dovetail slots in the rotor. The former exhaust ports are shut, and the reverse ports opened, while at the same time the main steam feeding the passages which lead to the "forward" ports is closed, while the reverse steam, feeding a separate set of passages, is opened. Steam then goes to the double ports seen dotted in Fig. II. These double ports are pairs, the inner of each being connected to the main steam, and the others to short passages leading to the spaces just inside the forward motion exhaust ports. Thus, as the engine revolves, steam so to speak "jumps" the bridges in the double ports and puffs are emitted behind the vanes.

So much for the reverse speed, but we have not yet completely described the possibilities of the forward motion. So far the engine has apparently a fixed cut-off and can be controlled merely by the degree of throttle opening. The experimental engines which have been made have this fixed cut-off, although we understand that the inventor proposes to fit an arrangement whereby this can be varied. Additional power, however, can be obtained by converting the engine into a turbine, driven all the time by steam at full pressure. To do this the exhaust ports normally used for reversing are cut off from the exhaust pipe and put into connection with the main steam pipe, thus admitting full pressure steam continually behind each vane as the rotor revolves. When set thus, either supply may be throttled or either may be cut off, but of course it is assumed that the engine is usually run in the way first described, the turbine supply being kept in reserve.

The principal advantages claimed for this novel construction are its simplicity, that is to say, the small number of moving parts, its compactness, its ability to run at very high speeds, and its smooth running when at high speed. With regard to the last claim, while this is undoubted, it ought certainly to be explained that the engine runs well at slow speeds with but little appearance of leakage. Two experimental engines have been made, one to deliver something like 12 h.p. with eighty pounds steam, and at a high rate of revolution, and another of considerably larger dimensions to give some 50 h.p. with similar steam at a speed in the neighbourhood of 1,000 r.p.m. In the latter engine the arrangement of rotors and passages is duplicated, giving both a high and a low pressure system, the exhaust steam from the high

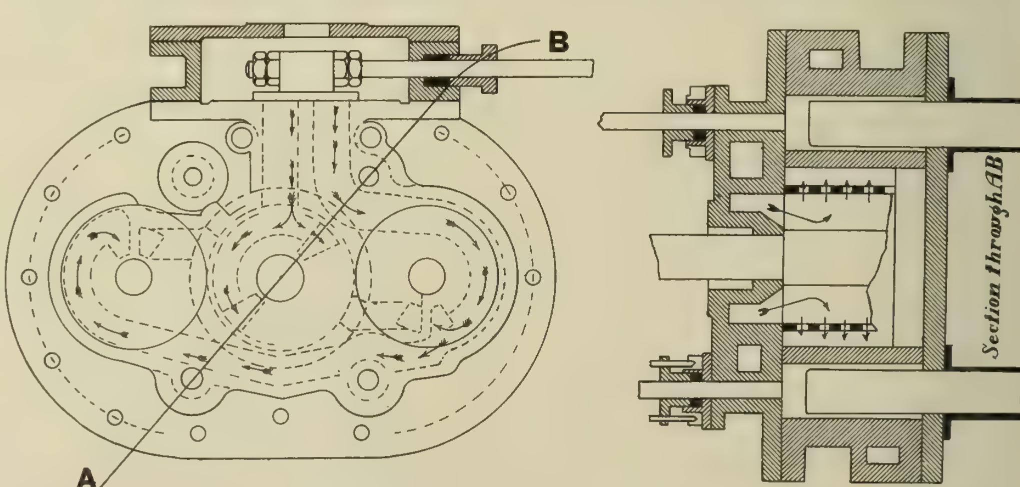


Fig. II.

pressure box being passed over to the second portion by external pipework. The latter has been found to cause very considerable loss, but the arrangement is only experimental and there appears no doubt whatever that quite good economy would be obtained by combining the high pressure and the low pressure rotor in the same containing case. It might, perhaps, appear difficult to make an engine of this kind with sufficient accuracy to avoid friction, while at the same time preventing leakage of steam, but it must be remembered that the essence of the whole invention is the method whereby the "rotary stators" are backed by pads of high pressure steam, so that they practically float, and it is this that makes the engine so much more promising than the vast majority of broadly similar designs.

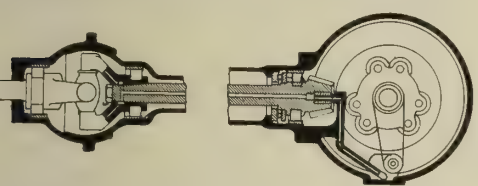
We understand that Mr. Marks—the inventor—is prepared to show the experimental engines (which are now in London) to any engineers who may be interested.

RECENT AUTOMOBILE PATENTS

By Eric W. Walford, F.C.I.P.A.

Universal Joint Lubrication.

A pump is mounted in the bevel casing on the back axle which draws a supply of lubricant from the casing and pumps it along the propeller shaft to the universal joint. In the construction illustrated a rotary pump is chain driven from the axle and delivers by means of a tube to a duct

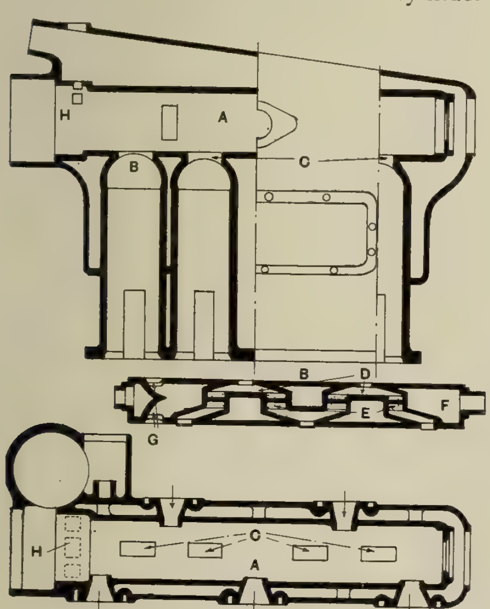


passing through the centre of the propeller shaft. This duct communicates with passages leading to the universal joint bearings, whilst the lubricant passes out from these into the universal joint casing and returns down the torque tube to the axle. The inlet to the rotary pump is at such a height that a small amount of lubricant is always maintained in the axle casing.

No. 17,137/10. W. F. Rainforth and D. Napier and Sons, Ltd.

A Rotary Valve Engine.

Over the cylinder head is arranged a tubular casing A, which contains a rotary valve member B. The casing A is formed with inlet and exhaust ports as shown by the arrows, and these are put into communication with the cylinder

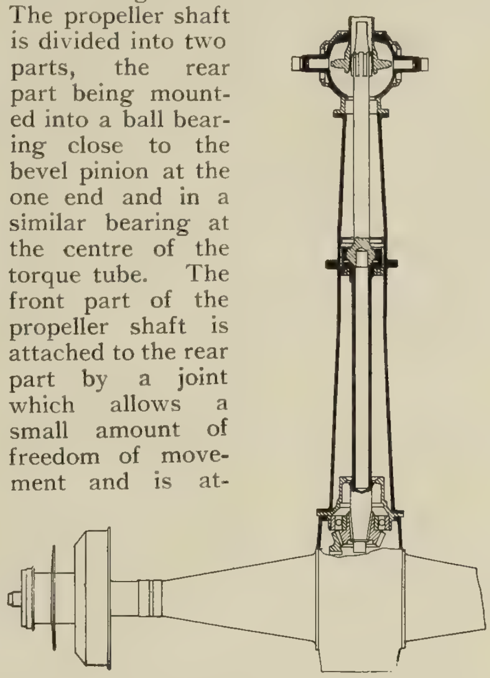


ports C by the rotating valve member B, which is formed with inlet passages D and exhaust passages E. The cooling water enters the valve member at F, passing helically along the hollow casing and issuing at G through passages H into the main water outlet. The rotary valve member therefore constitutes a water circulating pump.

No. 1,731/11. La Société Anonyme L'Auto Metallurgique.

Propeller Shaft Construction.

In this construction a torque tube is used which is connected to the car frame by a universal ball joint, the torque tube transmitting the axle thrust to the frame. The propeller shaft is divided into two parts, the rear part being mounted into a ball bearing close to the bevel pinion at the one end and in a similar bearing at the centre of the torque tube. The front part of the propeller shaft is attached to the rear part by a joint which allows a small amount of freedom of movement and is at-



tached to the main universal joint. The front shaft length is consequently not supported in bearings of its own, and floats between the main universal joint and the rear propeller shaft part which, however, is supported by the bearings mentioned at both ends. Consequently the propeller shaft is relieved of all stresses due to errors of alignment and spring. The invention also includes a special form of ball joint casing attaching the torque tube to the main frame.

No. 25,485/10. F. H. Royce and Rolls-Royce, Ltd.

A Radial Cylinder Engine.

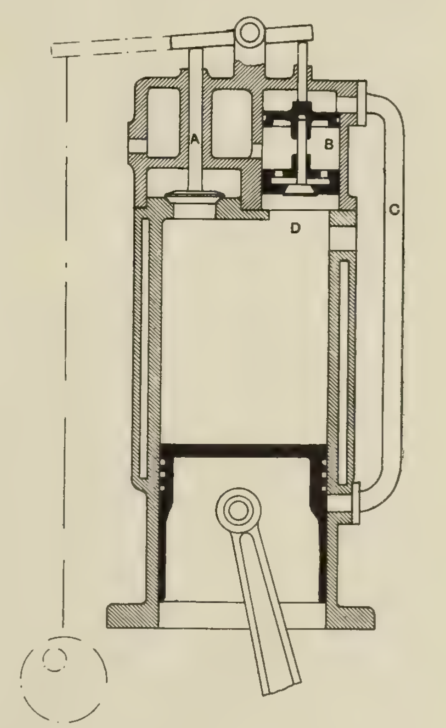
In order to enable radial cylinder engines of large powers and light weight to be constructed, the outer cylinder parts are of large bore, accommodating a shallow piston or disc. This is attached to a trunk portion which works in a guiding cylinder of smaller bore, and is connected to the crank through a connecting rod. It is not possible to use large bore cylinders throughout without considerably enlarging the crank chamber, which increases the weight, and, in revolving cylinder engines, brings the cylinder masses far from the axis of rotation, increasing the centrifugal stresses.

The engine operates on the Gnome principle in which gas passes through an inlet valve in the piston and exhaust takes place through the cylinder head.

No. 25,391/10. Société des Moteurs Gnome.

Pneumatic Exhaust Control.

The exhaust valve shown at A is of inverted conical shape, and is actuated through a rocker by a piston B, which is open to the cylinder pressure on the one side, and on the other to the pressure obtaining in the pipe C, the end of which is uncovered at both extremes of travel of the main piston. The inlet stroke is arranged at D. On the inlet stroke the exhaust valve A is sucked on to its seat, and the inlet valve D opened. On the compression stroke the tendency for the exhaust valve to open under the pressure in the cylinder is resisted by the greater total pressure on the piston B on account of its large area. The same applies to the firing stroke, but at the end of this stroke the pipe C is uncovered by the main piston and put into communication with the cylinder. The piston B is now balanced so that the pressure on the valve A causes it to open. It may be held open by the rocker being positively actuated by its eccentric, as illustrated, it being understood that the actual opening is effected pneumatically, mechanical means being used only to hold the valve open, a



somewhat complicated process without any apparent advantages, though undoubtedly the method of control is ingenious.

No. 793/10. J. Scott.

THE SOCIETY OF AUTOMOBILE ENGINEERS.

Summer Meeting.

In our last issue we announced that this meeting would take place at Dayton, Ohio, on June 15th, 16th and 17th. Below we give the programme of events :—

THURSDAY (8.30 *a.m.*).

1. Opening Address of President Henry Souther. Business Matters.

Business Matters.

2. Reports of Tellers of Election of Members.
3. Treasurer's Report.
4. Announcement of Membership vote on Constitutional Amendments (as provided by Paragraph 59 of the Society's Constitution).

Professional Matters.

5. Reports of Standards Committee Divisions (10 a.m.).
6. Illustrations of Physical Facts relating to Metallurgy. Paper by Radclyffe Furness (11 a.m.).

FRIDAY.

Professional Session.

1. Elements of Ball and Roller Bearing Design. Paper by Arnold C. Koenig (8.30 a.m.)
2. Worm Gears and Wheels. Paper by Warren Noble (9.30 a.m.).
3. Reports of Standards Committee Divisions (10.30 a.m.).
4. Topics for Discussion (11.30 a.m.).
   1—Long stroke motors.
   2—Underslung frames.

SATURDAY.

Professional Session.

1. Paper and Discussion (8.30 a.m.).
2. Paper and Discussion (9.30 a.m.).
3. Standards Committee Division Reports (10.30 a.m.).
4. Topics for Discussion (11.30 a.m.).
   1—Multiple-disc clutches.
   2—Six cylinder *versus* four cylinder motors of equal rating.

Additional Subjects for Discussion if the Opportunity Affords.

1. Three-Point *versus* Four-Point Suspension.
2. Current Practice in Lubrication and practical results obtained.
3. Piston Ring Fittings and Piston Ring Friction.
4. Elimination of noise in motor cars.
5. Present trend in compression of gasoline automobile motors.

Report of Steel Tube Standards Committee.

Some time ago the Detroit Seamless Steel Tubes Company prepared a table which the members of the Society have probably seen, giving sizes of proposed standard tubes. It is the idea of the seamless steel tube division that it will move a little slowly in the matter of recommending standard sizes. The committee feel that there is a great deal to be gained by the engineers of the different automobile companies working together in this matter, getting down to a limited number of sizes. Various radical changes have been suggested like going to entirely new gauges for thicknesses, so as to cut down the number of gauges.

At the present time however, we want to make our report so that it will not disturb in any way the manufacturer or the tube mills, or current practice among automobile makers. Consequently the committee have taken the above-mentioned list, which covers tubes from ½ in. to 5 ins. in diameter and gauges from 20 (.035 in.) to 1 in., and have eliminated certain sizes which are used so rarely that they need not be considered. Both 14 (.083 in.) and 12 (.109 in.) gauge tubes have been eliminated entirely. Gauge 14 comes so near 13, which, the tube makers say, is one of the most common sizes, that the committee feels safe in cutting it out. This also applies to 12 gauge, as well as to tubes of 7/32 in. (.219 in.).

The committee are going to get together from the automobile builders and the tube makers a list of all the tubes useed by the automobile industry to-day. The table prepared by the Detroit Seamless Steel Tubes Company covered the field of not only the automobile builders but of the trade at large. The committee are interested in only what the automobile builder wants, and will consider only the sizes in which the automobile trade is interested.

The committee feel that in a year's time, when they will have had a chance to canvass the situation, they will find that 50 or 75, probably 50, sizes of steel tubing will supply the entire automobile trade with everything it has any use for. Doing that means that New York, Detroit, Cleveland and Chicago, can carry tubes in stock. If we can reduce to 40 or 50 tube sizes, the warehouses can and will carry these in stock and we can get them for less money and more quickly, serving all interests.

One very important thing which the automobile designer ought to bear in mind is that steel tubing is a raw product, and that it should not be expected to get it down to thousandths of an inch unless paid for proportionately. It seems to be the concensus of opinion of tube makers that tubes smaller than 2 in. O.D. will vary from .005— to .005+ on both I.D. and C.D.; and tubes 2 in. and larger will vary from .010— to .010+ on both I.D. and O.D.

Of course, there will always be classes of work where none of the proposed standard sizes can be used, but for 85 to 90% of the work of the automobile engineer about 50 different sizes should suffice.

MISCELLANEOUS.

WE ARE INFORMED by Mr. J. D. Roots that the claims made for his engine, which we described last month, are stronger than quoted in the article. Writing concerning the power absorbed by the clutch when slipping, he remarks as follows :—

"I claim that this amount would be about one-tenth of the present losses with the ordinary box of gear between the engine and the driving wheels."

"As there is always in this engine a practically direct drive the losses must be less than those at present occurring with the ordinary gearbox, and moreover as I know well from experience with similar clutches, that the clutch does not heat after a whole day's run, and as the measure of the loss of power must necessarily be the degree of heating, the loss by the friction on the rotary device cannot be *considerable.*"

ELECTRO-PLATING AND POLISHING is dealt with very fully in a volume recently issued by W. Canning and Company, who are suppliers of every kind of apparatus for use in connection with electro-plating. The book describes in detail the precautions which have to be taken when depositing any metal, gives formulæ for a very large number of different baths, and gives equal attention to the special problems concerning all sizes of work from the smallest to the largest. To a casual reader not the least interesting section of the work is that devoted to polishing articles in all kinds of metal, both plated or solid, and many useful hints will be found also on lacquering and the protection of polished work from atmospheric action without spoiling its appearance.

BRITISH MOTOR VEHICLES is the title of a handbook compiled by J. S. Critchley. It contains some statistics concerning historical automobile events, certain legal particulars, tables of customs duties, etc. The chief feature of the work however, is a list of British cars with their specification and price. This is arranged in alphabetical order and would be convenient for reference to those who require such information. The form of the book is good, and reference to any section of it should be simple. (*C. D. Clayton, Ltd.*, 1/-).

SPANNERS of various types are dealt with fully in a leaflet issued by J. H. Williams and Co., of Brooklyn, New York. Amongst them are some excellent car kit sets, which are most reasonably priced and can be made up to suit the needs of any particular chassis.

ALBERT DURST AND FRANKL have sent us a sample of a machine sponge cloth which appears to be extremely strong and absorbent. We understand that these cloths can economically be washed for a very small charge.

HANS RENOLD, LTD., have recently issued an extremely interesting pamphlet on the subject of chain drives for camshafts, together with a translation of a French article on the same subject. The publications should be really useful to any who are considering the adoption of chains for engine work.

C. A. VANDERVELL & CO., LTD., inform us that they have arranged with the Royal Automobile Club to conduct a second and even more searching test of their car lighting dynamo. It will perhaps be remembered that in a recent test several unfortunate accidents occurred, which marred the otherwise good record of the machine.

IN THE ARTICLE ON THE theory of the Norton Running Balance Machine last month by Mr. G. S. Bower, there were several printer's errors in a portion of the issue. They are largely obvious, but we should be glad to hear from any reader who has been unable to follow any portion of the article from this cause.

SPEEDOMETERS.—A new and complete list of Jones' Instruments has recently been issued by Markt and Co., Ltd.

ACCUMULATORS, ETC.—Van Raden and Co., Ltd., have a new catalogue of their electrical goods especially intended for automobile use. A number of new batteries appear in it for the first time, some of them being designed especially for use with car lighting outfits. Amongst other novelties are some new head lamps—of course, electric—while the ignition specialities of the firm are also dealt with fully.

THE SPEED OF RECIPROCATING ENGINES.—Note errata in previous instalment in last issue. Page 340, line 10, col. 2, for "by increasing the *stroke*" read "by increasing the *diameter.*" Line 39 in the same column—read "curves. . . . 1 and *4* Page 341 last line but one, first column, for "*slow*" read "*stop.*"

AUTOMOBILE ENGINEER VOLUMES.

Our last issue was the twelfth to be published, as *The Automobile Engineer* first made its appearance in June, 1910. It has, however, been decided to continue the first volume until the end of the present year, it being believed that it is more convenient for a periodical to commence each volume with a January issue. The index will therefore be issued in December next.

We take this opportunity of informing readers that we have only a very few copies of the first few issues left, but these will be supplied to any desirous of keeping a file for binding purposes, although it has been necessary to fix special prices (which may be obtained on application) for certain issues.

THE
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Articles of a technical nature relating to the design or construction of automobiles for land, air, or water, will be carefully considered by the editor. Matter must be clearly written or typed on one side of the paper only, and a stamped addressed envelope must be enclosed for return. No responsibility can be accepted for the safety of contributions although every reasonable care will be taken.

Correspondence on interesting subjects, which come within the scope of this journal is cordially invited. All letters must contain the name and address of the writer, but not necessarily for publication. If a reply is desired by post enquiries must be accompanied by a stamped addressed envelope.

CONTENTS.

THE IMPORTANCE OF TRIALS.

In our last issue it was suggested that there was a tendency to under-rate the amount of really useful knowledge to be gained from racing cars, and it was pointed out that racing, as conducted in this country, ought not to be regarded solely as a means for advertisement. Many of the same arguments apply even more strongly perhaps, to the running of cars in reliability trials, and, though this once popular form of testing has been banned by the Society of Motor Manufacturers and Traders as useless extravagance, some day no doubt the falseness of this view will be realised. Of course, there is no secret about the facts that reliability trials have been declared anathema by the trade representatives of the industry for two reasons, one being the expense (which pressed more upon the small maker than upon the large) and the other the "flukiness" of the results (which pressed with equal rigour on both, because the large maker had more reputation at stake for which he was compensated usually by his financial ability to run a team of cars instead of only one). Be this as it may, the engineering side of the industry has never had a say in the matter: in fact it has certainly never been generally realised that the latter body had anything to gain or lose by the running of trials, with the possible exception of a few personal reputations.

It is, of course, easy to argue too, that much is to be learnt by conscientious car testing conducted privately by individual makers and their staffs, but it is doubtful whether this argument is really sound, because it requires very exceptional strength of mind on the part of all individuals concerned in the test to make that test as arduous as a competitive trial, and besides far more is to be learnt from the comparison of one car with another of different type than by the observation of a series of the same type.

It has been claimed that success in a two thousand miles trial even is an insufficient guarantee of reliability, because, barring accidents, any good car ought to be able to cover this distance without trouble. While it would be impossible to deny the accuracy of this statement, as a statement it is not complete, because the fact remains that even in the last trials the number of mechanical casualities was enormous. If material is good and workmanship is good as well, no parts of a chassis ought actually to wear out under, at the very least, fifteen thousand miles running, but if there is any bad design it is likely to show up well within the limits of 2,000 miles.

To say that such and such a feature is bad design may be taken as equivalent to saying it is so proportioned that some part of it is stressed unduly, is inadequately protected, or is inconvenient in use. With respect to the last two characteristics, these are obvious very quickly, but nevertheless they still continue to be found as conspicuous faults in very many cars. With respect to the first named, this might still exist and yet fail to cause actual breakdown within the limits of a trial, but it is safe to say that, if the design was bad enough to be really injurious, some minor fault in running would appear.

Probably had all trials been conducted as well as the majority of those organised and controlled by the Royal Automobile Club, the outcry against them would have been delayed some years, if indeed it had ever arisen, and of course a trial is of no value unless it is conducted in a scientific way. The rules need to be strict and the enforcement of them equally so, while, ideally, trials should always be carried out with absolutely standard cars arguing from the point of view of the maker alone and not considering the public point of view, because it is the behaviour of his standard type, which is of greatest importance to an engineer—and to the manufacturing firm as well, if they have any desire for a lasting reputation.

It is quite safe to say that no maker has yet run a team of cars through a long trial without gaining some knowledge concerning them, and we believe that practically every trial which has been held has resulted immediately in a number of small improvements which, if they were all taken together, would decidedly raise the average of excellence of the competing cars, and, if the British-made car is to continue to hold the extremely high position it occupies at the present time, improvements must continue to be made. It is easy to argue that alterations are directly opposed to commerce, and like most good things improvements in a design can be overdone. Never to turn out two cars alike is an inconvenience and an expense to everyone from the manufacturer himself down to the user, but to refrain from altering known bad points simply because they are not very bad and it is cheaper to continue with them, is decidedly foolish as a season to season policy.

There is a point which has been overlooked perhaps by the Society of Motor Manufacturers and Traders, but is worth quoting as an instance of possible bad effects which may arise from the total neglect of open competitions. This is that immediately the makers in France gave up their encouragement of races and trials the exportation of French automobiles declined steadily. It might easily be argued that this was the effect of loss of advertisement, but anyone who has examined French design at the last Salon could tell a different tale.—Absence of competitive stimulus has prevented the advance of French design and allowed England to take the lead. If we are to keep that lead we must maintain our present rate of progress even if it be "uncommercial." Let the opponents of trials ponder well on that ancient but most true of quotations *quem deus vult perdere prius dementat.*

THE INSTITUTION OF AUTOMOBILE ENGINEERS.

A Critique of the Work Done during the Past Season.

IN commenting upon the growth and development of the Institution of Automobile Engineers in our issue for August, 1910, we remarked that the new secretary would receive his appointment at the close of a session then the most memorable in the history of the Institute, and we further said that year by year the work done by this, the youngest engineering body, has increased in importance and usefulness. It is interesting now to review the proceedings of the Institution during the last session to see whether the importance and usefulness of the work done is still upon the upward grade.

Firstly, it is pleasant to be able to chronicle the fact that the membership has increased very considerably, which, of course, means that the Institution and its sphere of usefulness have expanded in direct proportion, although the total membership is still far smaller than it ought to be, having due regard to the magnitude of the automobile industry in this country. Secondly, the past session has seen the installation of the Institution in its own offices, which it holds independent of any other body, although the principal meetings still continue to be held at Storey's Gate (by the courtesy of the Institution of Mechanical Engineers), owing to lack of a suitable hall in the new premises. Undoubtedly therefore, it may be said that the status of the I.A.E. has undergone a quite appreciable improvement.

Turning now to the actual proceedings, and the papers which have been read since last November, it is lamentably difficult to direct attention to any definite point in either the papers themselves, or the discussions which followed, which may be said to have advanced the knowledge of automobile engineers generally. As this may perhaps be thought to be rather a sweeping statement, it is proposed to take the papers one by one and to comment upon them in the order in which they were given.

The Presidential address is scarcely comparable with ordinary papers, because it is generally—and rightly—accepted that such oratorical efforts should be devoted to dealing with broad issues in a very wide way, rather than to the narrow consideration of some one particular branch of research. Still, those members of the Institution who attended the first meeting of 1910-1911 session were given ample food for thought, even though they did not learn much. The paper was, it is not too much to say, a particularly clear and lucid statement of the principal problems with which designers are now faced, and even though most of the audience were doubtless perfectly well acquainted with them, the martialling and arranging of our difficulties and in order of merit before our eyes is an excellent and practical discipline.

Carburettor Action.

The first controversial paper of the session was therefore the second to be read, and this, as will doubtless be remembered, was entitled "Carburettor Action," and was given by Professor W. H. Morgan and Mr. E. B. Wood, being based upon some elaborate experiments carried out partly at the works of the Daimler Company, at Coventry, and partly in the excellent automobile engineering laboratory attached to Bristol University. In this paper Prof: Morgan sought to prove by deductions from his observations that there were certain definite laws connecting the flow of petrol from a jet with the speed of passing air, the suction of which causes that flow. His conclusion was that all so-called automatic air-adding devices were detrimental to good carburation, and that the best carburettor was obtained by quite a simple construction consisting of a plain jet and air opening with means for adding a constant additional quantity of petrol on the principle now employed in the "Limit" carburettor. Prof: Morgan's deductions have been derided by some critics, and a hot controversy followed the reading of the paper, both by word of mouth the same evening and by correspondence in various periodicals at later dates.

Carburettor theorists are always faced with the troublesome but undoubted fact that some of the most successful carburettors for ordinary everyday use are the very worst in point of theory. It is easy to recall three or four types at least with elaborate air valve attachments which give all the flexibility that could possibly be desired, together with a reasonable economy of fuel, and yet to demonstrate that these same carburettors give very far from a constant quality of mixture is extremely easy. It is greatly to be regretted that a certain animosity between Prof: Morgan and his principal critic has prevented the academic discussion of the investigations which led up to the writing of the paper. Prof: Morgan has certainly put forward one of the most constructive theories of carburettor action which have yet been published; even if the deductions are some of them erroneous the facts ascertained in the preliminary researches are undoubted. We think therefore that Prof: Morgan's paper may be classed as one of the most important in the session, believing as we do that some of the theories found in it will be heard of again at no very distant date.

Two Cycle Engines.

In December a paper was to have been read by Mr. F. B. S. Bircham on the develoment of internal combustion engines for marine purposes, but this had to be cancelled, and instead of it Dr. Watson read his paper on the efficiency of the two cycle engine a month earlier than had been arranged for on the programme. Dr. Watson is quite rightly reputed to be one of the most thorough amongst the physicists who who have paid special attention to the problems of the automobile industry, and his paper was therefore perhaps unusually complete, while the conclusion arrived at, as the result of an analysis of an enormous number of investigations, was that even an extremely roughly constructed two-cycle engine could give an efficiency very little inferior to that of an average four-cycle petrol engine. One trouble however, is that some of the chief objections to what might be called the standard pattern of two cycle engine (that is to say with crankcase compression, cylinder ports at the bottom of the stroke, etc.) are by no means so apparent when the engine is being run in a laboratory as when it is in practical use. Putting the matter in quite plain language one might say that the question of efficiency is of secondary importance, so long as it is reasonable. Undoubtedly some forms of two stroke engine will eventually displace the four stroke, but the deciding factor will not be whether its efficiency is greater or less than that of the four stroke so much as whether its flexibility and reliability are better or worse. Dr. Watson's paper is therefore largely academic. It is of comparatively small importance to designers however interesting it may be to students, which means that the ideal two stroke engine is scarcely brought nearer by the investigations of the efficiency of the particular engine which was used.

Castellated Shafts.

In January, when Dr. Watson's paper should have been given, an addition to the programme was made by the introduction of Mr. C. R. Larrard's paper on the strength of castellated shafts. This was again largely academic, but it was also practical, as the author's aim was to show the way in which castellated shafts failed when twisted to destruction when made of various materials treated in various ways. It tended to show that certain forms of shaft and certain proportions of splines and key ways, etc., appeared to make the strongest shafts. In course of preparing the paper many hundreds of shafts were twisted to destruction in the engineering laboratory of the Northampton Institute by the author, and some interesting cinematograph pictures were shown exhibiting the surface actions which occurred during the destruction of several shafts of different materials. While extremely interesting to those automobile engineers who have to make metallurgy their own peculiar study, it can scarcely be said that the conclusions were such as appreciably to alter our existing knowledge and employment for automobile work, simply because castellated shafts must be so strong that no deformation worth considering would take place under maximum stressing in service. That is to say, no castellated shaft will ever be called upon to twist beyond a small fraction of its elastic limit.

One point which may be recalled worthy of remembrance was that these shafts almost invariably failed firstly, by radial cracks running inwards from the bottom corners of the splines, and that the larger the splines were in proportion to the shaft diameter the more pronounced this cracking became. This may probably be taken to be another example of the importance of good radii at corners particularly on high tensile steels.

Horse Power Rating.

In February there was published what has been referred to as the *magnum opus* of the Institution, the report of the Joint Committee of that body, the Society of Motor Manufacturers and Traders, and the Royal Automobile Club, who have been discussing horse power formulæ for upwards of three years, followed by Mr. G. A. Burls' paper proposing a maximum rating formula. This report has been dealt with very fully in these columns both descriptively and critically, so at the present time there is no need to dilate upon it. To designers and manufacturers generally, far the most important and most valuable portion of the paper was the sheet of data giving the proportions of the one hundred and fifty engines from which the formulæ were deduced. As regards the formulæ themselves, at the present moment they are possibly more accurate than any which have hitherto been in use, but the constants employed in them could be made to vary by small differences in standard practice as regards the design of engines—everything depends upon the purpose for which a formula is desired. A rough measure of an engine's power is given by the amount of fuel it burns, and this is again roughly proportional to the displacement volume of the engine. If the data on which horse power is to be calculated is to include nothing but bore and stroke, it will not be easy to obtain a formula which will give a better all-round rough comparison than this volume rating The suggested formulæ can give much more accurate determination for certain engines, but if means were found to obtain a higher mean effective pressure or to increase rates of revolution, then new constants would be required at once. Now all these things alter together, and for each of them separate constants are involved, wherefore to alter the formula would be as difficult as to re-make it entirely from the beginning, because a whole series of fresh engines would have to be examined in every detail in order that the calculations for their proportions could be made and the average case again deduced. On the other hand, with a purely arbitrary formula like a volume rating, no great discussion would be required to change the constant.

On the other hand, as we have several times remarked in these columns, a formula is needed badly which will take everything into consideration, and which will therefore be perfectly useless for all kinds of rating but would be extremely valuable to designers because it would enable them to calculate the effect of altering a given dimension or proportion in their own engines. The proposals of the horse power committee and those of Mr. Burls come much closer to this "Engineers'" formula than anything which has before been put forward, but they do not come quite far enough—they still leave too much in the "constant" form. While the committee are to be congratulated on the immense amount of valuable data which has been collected by their extremely able secretary, we think that their work is not yet complete.

Air Craft.

In March, Mr. Mervyn O'Gorman presented a paper which in the light of what we have said in the commencement of this article would have been obviously suitable for some presidential address; it was in fact very similar to Mr. Lanchester's opening of the session, but dealt solely with the navigation of the air instead of that of the land by road, that is to say it was an exposition of the more immediate problems in connection with flight. It was a summary of the different principles of aeroplane construction as shown by the various types now in existence. It dealt briefly (all too briefly unfortunately) with methods of construction, and it stated in particular the requirements of the army in respect to aeroplanes. Noticeably little was said about the dirigible balloon, although it had been anticipated in some quarters that Mr. O'Gorman would deal particularly with this more cumbersome form of flying machine. Mr. O'Gorman's paper therefore may be commented on in precisely similar terms to those which we have already used when referring to the presidential address of this session.

Wheels and Roads.

Following Mr. Mervyn O'Gorman's conversational paper on airships and aeroplanes, Prof. H. R. A. Mallock, F.R.S., in April read a paper of the purely mathematical sort concerning the effect of road irregularities on wheels rolling over them, and the effect of the wheels on the road, dealing also with the motions of a carriage body as supported from the wheel by springs. Although interesting in a lecture room manner, it is very hard to see in what way this paper has helped towards the improvement of automobiles, or in what way it has tended to the assistance of automobile engineers. All it did was to show that the spring designer has no easy task (which indeed he knew before), and that road constructors are faced with certain difficulties contingent on building roads to withstand heavy, high speed traffic. Without further comment therefore, we can turn to the next and last paper of the session.

Pressed Steel.

Mr. Legros' paper was reported in the last issue, and there is, therefore, but little need to enlarge upon it here. Broadly speaking, it was valuable as a catalogue of the possible uses of pressed steel, as exemplified by practice, while containing also, many suggestions as to the enlarged use of die-formed steel in place of castings. Most automobile engineers will be in agreement with Mr. Legros in this respect.

In quite another way, too, is the paper commendable. The introduction describing briefly the different ways of treating steel by pressure should be studied by all who are not most thoroughly acquainted with them, because a knowledge of methods of pressing is essential to the designer who wishes to use stamped or pressed parts to the best advantage.

General Management.

Looking back over the previous few columns it certainly seems that the Institution of Automobile Engineers has done very little to increase its prestige during the past session. It has undoubtedly increased its membership, but this, we believe, is owing very largely to the energy and enthusiasm of the secretary, Mr. Basil H. Joy. Throughout the season the attendance at meetings has been deplorably bad, with the solitary exception of the paper read by Mr. O'Gorman, when the lecture hall of the Institute of Mechanical Engineers was full to overflowing—unfortunately more with visitors than with actual members. There is full reason to believe that the scheme suggested in the May issue of The Automobile Engineer might change the sad state of affairs as regards attendance at meetings, but the time has come when it is essential for the good of the Institution, to point out one thing concerning the administration which has a most evil influence.

This is that the number of practical men—that is to say men who are actually and successfully engaged in the industry—upon the Council is comparatively small. It seems a hard thing to say, but it must be said none the less, that more than one of those who have done most towards the starting of the Institution, and who are the most ready to open discussions are, by their very lack of knowledge of modern conditions, doing more damage than they do good. There is, of course, no need to mention individuals, but if it is desired to make the Institution a really "live" body, and to obtain really "live" papers, it is absolutely imperative that the Council and the principal spokesmen thereof should be representative of modern knowledge. On the part of those at present in power, there is far too strong a tendency to treat all proceedings as an amusement; to regard the discussion of papers as a battle of wits far more than as a means to obtaining useful information for the membership at large. It is quite conceivable that an excellent and most useful paper might be read, dealing with the very latest phase of some particularly modern problem, but in such a case the only people living who could assist usefully in the discussion would be those who had been conducting experiments similar to those of the author of the paper, and whose knowledge on that particular subject was therefore absolutely up-to-date. To open a discussion on a good paper by three-quarters of an hour spent in time-honoured and age-weary platitude, put forth with the discursiveness of self-satisfied middle-age, is quite sufficient to ruin the effect of any paper upon a practical audience.

The Institution of Automobile Engineers, as it is at present, ought to be the nucleus for a really powerful and useful body. Unless its control is better carried out however, there is grave reason to fear that it will remain as futile as it is at present. Although this last adjective is a strong one, there is no doubt that the circumstances deserve its use as things are at the moment. That this is so is no reflection on the Executive nor upon the authors of papers; it is directed entirely against the Council. In the early days of corporate bodies it is essential that the governing Committee should be active and young—in mind at least. This cannot be said of the now existing Council, and if present methods are continued for many years more, we doubt whether it will be easy to obtain the "new blood" which is now so sorely needed. At present nothing would be more simple than to elect good men from the membership—men who would rapidly change the status of the Institution with respect to the profession it was formed to serve. There is, happily, a plenitude of such individuals.

SOME PROBLEMS OF THE TWO CYCLE ENGINE.

By R. W. A. Brewer, A.M.I.C.E., M.I.M.E., M.I.A.E.

THE propulsion of motor vehicles has been achieved and developed by engines working on the Otto cycle, and the brains of designers have been concentrated almost entirely on the perfection of this type of engine, since the first motor car made its appearance upon the road. It is only natural, therefore, that the modern four stroke engine has reached such a state of perfection at the present time, in spite of several inherent disadvantages under which it works. When one studies the development of the explosion engine the name of Mr. Dugald Clerk at once comes into one's mind as being that of one of the most scientific and persistent investigators of the problem relating to internal combustion engines. Since 1877 Mr. Clerk has been at work upon the two cycle engine, and his first practical result was exhibited at the Kilburn Exhibition of the Royal Agricultural Society in London in 1879. This early engine consisted of two cylinders, with pistons connected to a two throw crankshaft, one cylinder being a compresser for air and gas, delivering the mixture into a chamber at the back of the second cylinder, in which it was eventually fired.

An improved Clerk engine had a conical combustion chamber, into which the charge was delivered at its apex through a mushroom valve, the exhaust products being expelled when the piston uncovered ports in the cylinder walls at a crank angle of 40 degrees before the outer dead centre. In this engine the pump piston was actuated by a crank set at 90 degrees to the power crank. Thus the mixture of air and gas was slightly compressed until such a time as the exhaust ports were uncovered and the pressure in the cylinder was less than the pressure in the pump receiver; then the mixture passed automatically into the working cylinder. The final compression in this engine was carried out by the working piston on its returning stroke. The development of this engine is the Körting Bros.' design.

In engines of these dimensions it is only possible to obtain economical working, without the risk of firing the incoming mixture, by supplying air and gas by means of separate pumps, and in the Körting engine each double acting power cylinder is fitted with a double acting air pump and a double acting gas pump in line with each other on the same piston rod. The length of the power cylinder and its piston are so arranged that at each end of the piston stroke a ring of exhaust ports round the centre of the cylinder is uncovered by the piston. A charge of air is then admitted, sweeping out the products of combustion, and is followed by a charge of gas. The advantages claimed for this type of engine are perfect scavenging of the products of combustion with cool, fresh air, elimination of risks of pre-ignition from the presence of incandescent gases left in the cylinder, the absence of exhaust valves which have to be lifted against a high pressure, and a cylinder diameter of about one-half that required in a single acting Otto cycle engine of the same power.

One might be tempted to think that conditions prevailing in large engine practice were of little interest to the automobile engineer, but, on the contrary, what has already proved a practical system on a large scale is daily being more appreciated for smaller powers. American engineers have for a long time been fully alive to the possibilities of a two cycle engine, and there are a large number of firms in that country constructing such engines, principally for use in motor boats, while there is no reason why the small two cycle engine should not be used for car work.

It may be argued that the Otto cycle engine is at present so simple and works so satisfactorily, with great controllability and silence, that it is unnecessary to push forward another type of reciprocating engine with a view to supplanting the Otto cycle system, even to a small extent. During the past twenty years' internal combustion men have concentrated upon the Otto cycle, and it is only to be expected therefore, that this type of engine should be practically in a state of perfection as the result of so much thought and labour. On the other hand the two cycle engine problem has only been tackled by comparatively few, but even now there are several very excellent two cycle engines doing regular work with economy. Still, the problems of the two stroke cycle are many, and in the past the following defects have principally been present :—(a) The difficulty of controlling the speed and power; (b) the loss of unburnt charges down the exhaust pipe; (c) the variation in the composition of the mixture due to attenuated charges and misfiring; (d) the difficulty of entirely displacing the burnt gas from the cylinder and filling the latter with a new charge; (e) excessive fuel consumption.

I shall first compare the losses in a two cycle and a four cycle engine, and investigate their causes, by this means endeavouring to come to some conclusions as to whether a small two cycle engine can reasonably be expected to compete with a four cycle engine.

Supposing that with equal cylinder dimensions we can obtain more power from the two cycle engine, it will not be necessary to increase the size and weight of the connecting rods or the crankshaft, and these portions of the material will therefore be utilised more efficiently. Conversely for equal powers the two cycle engine will be smaller, lighter and, in all probability, the cheaper of the two, power for power. Now taking losses, there is very little to choose between the loss of heat to the water jacket in either case, this being about 3,400 to 3,880 B.T.U. per i.h.p. hour, say 35 per cent to 40 per cent. of the total heat in the fuel.* With regard to friction and pumping losses, we will take the case where the charge is supplied by a separate pump. Here a two cycle engine is somewhat under a disadvantage owing to the fact that the volumes of air required for combustion must be handled by two separate cylinders. Assuming an equality of the other conditions, this might be expected to involve greater pumping losses in a two cycle as compared to a four cycle engine, but the increase of movement of air and gas does not really involve a large percentage of extra work, and we may take it that in a small engine the pumping losses amount to a total for a two cycle engine of 8% to 10% of the i.h.p. and for a four cycle engine of 6% to 7% of the i.h.p.

Referring to the proceedings of the Institute of Automobile Engineers, Volume 3, the pumping losses will be found to vary according to different authorities. Professor Hopkinson shows a pumping loss of 3%, and slightly less at lower speeds than 930 r.p.m. Mr. L. G. E. Morse shows his observed pumping loss at 720 r.p.m. equals 2.9 per cent.; at a thousand r.p.m. equals 3.8 per cent.; and at 1,220 r.p.m. equals 6.8 per cent. These were on an engine of an i.h.p. equal to 21 at the maximum speed.

The four cycle engine is at a disadvantage as far as rubbing friction is concerned, for the reason that the same piston rings which must hold up against 300lbs. to 400lbs. per square inch for half their time are normally holding up only about 1 per cent. of these pressures. The two cycle engine pump can be designed much more efficiently, and it is doing work practically the whole of the time. At the low pressure for which it is designed, its losses therefore are much lower.

According to an old rule the rubbing friction of material in contact with similar piston engines is proportional to the volume of piston displacement per horse power, hence four cycle and two cycle engines would show equal losses if the pump volume was added to the cylinder volume, but we have seen that a lower frictional loss is to be expected in the two cycle engine. In the pumping cylinder of a two cycle engine large clearances can be avoided; the valves can be of large area and light weight, and the fresh charge is not heated up during the pumping to the same extent that it is when it is pumped directly by the working piston.

We will now consider the most difficult problem, namely that of displacing the products of combustion by a complete cylinder-full of fresh mixture, for if this can be done, the power obtained from any one cylinder working on the two stroke principle should be more than double that obtained from the same cylinder on the four stroke principle, theoretically by a proportion represented by the ratio of the compression volume to the total working volume. It is obvious that this should be the case, as in a completely scavenged cylinder the compression volume, as well as the working volume, will be filled with explosive mixture.

Several methods have been tried for charging the cylinders of a two cycle engine in order to comply with important

* Taking calorific value of fuel at 19,500 B.T.U. per lb. and consumption = 0.5 lb. per i.h.p. hr
= 9,750 B.T.U. per i.h.p. hr

The loss of heat to the jacket $= \frac{3,400 \times 100}{9,750} = 35\%$

conditions which must be fulfilled in order that the two cycle engine should justify its existence. We may take it that the two cycle engine will stand or fall with the efficacy of the scavenging arrangement of the power cylinder, and also that its reliability, capacity and thermal efficiency will depend upon the scavenge. As a generality also an excess of air is indispensible in order effectually to drive out the burnt gases, for during this process turbulence is bound to be set up. There will be to a certain extent an intermingling of the air and the burnt gas, and some of the air must therefore be blown out through the exhaust port in every case where complete scavenging occurs. It will thus be seen that in any arrangement of crankcase displacement, (i.e. where the piston volume swept is the same on the crank chamber side of the piston as on the power side, and the chamber itself is utilised as a vessel in which partially to compress the charge), the maximum that can be hoped for in the way of new charge is a volume equal to piston displacement. Even this will scarcely be reached in practice on account of the losses occasioned by the transference of heat from the cylinder walls and other masses of metal to the new charge.

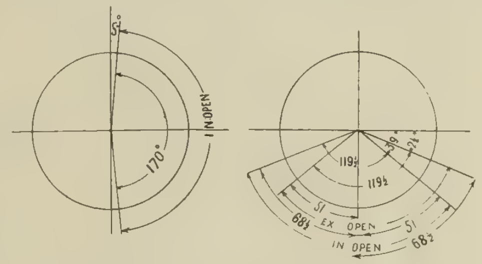
There is another point to be borne in mind where crank case pumping is adopted, namely, that the positiveness of the scavenge ceases when the piston reaches its outer dead centre, unless a receiver of sufficient capacity is interposed and a non-return valve is fitted to prevent the air returning to the crank chamber on the upward stroke of the piston. It will be pointed out later how such an arrangement also suffers by reason of the falling off in pressure of the air supply at a time just before the inlet closes, when a slightly higher pressure would be of considerable benefit. One further feature in the crank case system is that, when the front of the piston acts as a compressor, the heat transferred to the charge is considerable on account of the unsuitable shape of the volumes in which the compression takes place, and the leakage that is likely to occur through the bearings. It cannot be denied, however, that such an engine has the feature of simplicity in its favour and, where high efficiency is not a great point, it has many things to recommend it.

The Day engine of 2.5 h.p. at 900 r.p.m. having one single cylinder of $3\frac{1}{4}$in. bore and stroke, is an example of the type of engine just considered. Such an engine has been tested by Professor Watson and full details appeared in his paper read before the Institution of Automobile Engineers recently (see the AUTOMOBILE ENGINEER for January, 1911). I shall refer to these later.

In order to get over the disadvantages connected with crank chamber compression Mr. B. T. Hamilton and others have designed engines in which the working piston terminates in a second piston of larger diameter, and compression takes place in the annulus between these two diameters. Such an engine of course must be built in multiples of two cylinders so that one piston on its upstroke compresses air and gas into a receiver to be liberated by the piston in the neighbouring cylinder and covering the inlet port at the end of its outward stroke. In Mr. Hamilton's engine a very neat arrangement of rotating valve is employed between a neighbouring pair of cylinders so that the admission of the charge to the pumping annulus and its delivery therefrom to either cylinder can be regulated directly and mechanically.

There is, however, one inherent defect in all such engines in that the piston upon descending uncovers an exhaust port wider than the inlet port, as it is naturally necessary for the exhaust gases to escape before the new charge is forced into the cylinder. When a single piston uncovers the two ports, one on either side, it is obvious that the exhaust port closes after the inlet port, and so it is impossible for a complete cylinder full of mixture to be supplied. In such an arrangement also, where the inlet and exhaust ports exist on the opposite side of the same cylinder, it is necessary to employ a baffling piece on the piston head to deflect the incoming charge upwards into the cylinder barrel and as far as possible prevent it from blowing straight across the piston head and out of the exhaust port.

In order to get over this objection, what is known as the syphon cylinder arrangement has been employed by many designers dating back at least to 1900 by Binn, and later in 1903 by Messrs. Bickerton, Bradley and Dugald Clerk, of



Port setting for Lamplough two cycle engine.

the National Gas Engine Co., Ltd. The arrangement of cylinders in syphon form is convenient as offering considerable length of path for the incoming charge to traverse; for it enters one cylinder and has to pass along its barrel, across the combustion head and down the barrel of a second cylinder before it can emerge through the exhaust port.

∩-shaped cylinders, such as these, are most popularly known in the construction of the Lucas "Valveless" engine, and Mr. Reginald Lucas, at an early stage of his work, departed from a single straight cylinder and adopted the syphon arrangement with great success.

There is no objection to a straight cylinder when air scavenging only is employed, as it is quite possible to clear out the products of combustion from such a cylinder by air, which costs very little, and the loss of air alone through the exhaust port does not matter. Working on this basis Mr. G. F. Mort, of the New Engine Company, has perfected an arrangement in which air alone is pumped in the initial stages of charging and is followed by a charge of carburated air immediately before the piston closes the inlet port. Such an arrangement is almost ideal, as the highest possible economy can thus be obtained, and figures given by the New Engine Company show that they are able to obtain a power equal to twice that obtained from similar cylinder dimensions in a four stroke engine; and with the low petrol consumption of 0.6 pint per b.h.p. hour.

It is not always possible to arrange the pumping in two stages, though in large gas engines this is done because the extra mechanism involved is of small consequence, but in the Mort engine the separate pumping arrangement is carried out very cleverly and a rotary distributing valve is employed for the proper transference of air and gas to the cylinders at the correct moment. In this engine the introduction of the new charge is carried out by means of a rotary blower of the Roots type, which has been brought up to a high state of efficiency, and the pressure which one is able to obtain with such a blower, namely, about 5lbs. per square inch, is sufficient for the purpose. The object in admitting air alone in the initial stages is that, if slow, burning occurs, flame may possibly be existing within the cylinder at the time the inlet port opens. An admission of combustible mixture to the cylinder at such a moment would be dangerous, as this flame might strike back into the charging device. The explosive mixture is therefore delayed until the internal contents of the cylinder have been sufficiently cooled down and expelled. In the absence of accurate figures referring to the N.E.C. engine, it is impossible to strike comparisons with the simpler form of engine now so largely used in America, but, as far as consumption of oil and fuel are concerned, there is no doubt about the great advance which this engine shows in comparison with the American type.

I shall now proceed to discuss the thermal efficiencies of two stroke engines, and I feel compelled to make a few remarks with reference to what Dr. Watson states as a feature of the Day engine. In paragraph 10 of his paper he draws attention to the difficulty of carburation in a two stroke engine, and states that "unless the richness of mixture is adjusted within comparatively narrow limits, particularly at the high speeds, the engine refuses to work on the two cycle and only fires on every other stroke, the intermediate stroke acting as a scavenging stroke. The result of this peculiarity is that, unless the carburettor provides a mixture of uniform richness at different speeds and for different throttle openings, satisfactory working cannot be obtained." This is one of the great troubles in the ordinary two cycle engine, and as I have had considerable experience with the Lamplough engine working on the two cycle system with a charging pump, it is as well to consider the results obtained with this type of engine.

As many of the readers of the AUTOMOBILE ENGINEER doubtless know, I was able to show a small car at Brooklands working most satisfactorily at the meeting on March 25th. This car had a 4 to 1 gear ratio in the back axle, and I have since been driving it on the top speed and without manipulating the clutch in London traffic at speeds as low as five miles per hour, and with the same carburettor adjustment on the track at nearly fifty miles per hour. Practically no misfiring was noticed under these or any intermediate conditions, so that it is obvious that a two cycle engine can be constructed to work with great flexibility as far as range of speed is concerned.

Touching now upon the question of the loss of fresh charge to the exhaust

SOME PROBLEMS OF THE TWO CYCLE ENGINE.

Tables of data referred to in the article commencing on page 396.

Table I.

Test.	Revs. per Minute.	Petrol. Galls. per hour.	Petrol. Pints per B.H.P. hour.	B.H.P.	Air, cubic ft. per minute.	Air / Petrol by weight.	% Blower full.	Carburettor.	Brake.	Exhaust Gas Analysis. CO_2	O_2	CO	N, etc.	Excess of air %.	Loss of Charge %.	Mean Effective Pressure calculated on the B.H.P.
C 10	400	Running		Light.				Lamplough receivers. Connected.	8½" × 11" in 4th hole.	6·4	6·5	4·3	82·8	19·8	30	
B 8	800	0·88	1·67	4·2	17·6	12·8	61	Claudel 18 × 1·00 Half open.	8½" × 11" in 7th hole.	5·0	6.6	6·9	81·5	14·3	30·6	30·5
C 13	1240	1·33	0·37	11·0	42·0	20	94	Claudel. 18 × ·95.	8½" × 11" in 4th hole.	7·1	4·6	5·5	82·8	8·4	21·4	51·2
B 6	1560	1·58	1·0	12·6	56·2	22·5	100	Claudel 18 m.m. dia. × 1·0 jet. Full open.	6" × 6" in 15th hole.	6·4	5·5	5·2	82·9	13·2	25·5	46·7
C 12b	1370	1·89	1·0	15·0	49·5	16·7	100	Claudel 18 m.m. dia. × 0·95 jet.	8½" × 11" in 4th hole.	9·0	5·9	1·5	83·0	22·7	27·4	63·2

Table II.

Test.	Revs. per minute.	Petrol cons. Pints per B.H.P. hour.	B.H.P.	Air / Petrol by weight.	Exhaust Gas Analysis. CO_2	O_2	CO	N, etc.	Excess of air %.	Loss of charge %.	M.E.P. calculated on Brake H.P.
3	638	1·20	2·1	12·45						34	48·3
20	1218	1·22	3·3	11·0	7·4	4·2	5·6	82·8	5·64	19·7	40
28	1502	1·06	3·7	11·6						7·0	36·2

Table III.

Revs. per minute.	B.H.P.	Pints per B.H.P. hour.
1000	5·0	1·28
Do.	9·5	1·05
Do.	11·6	0·91
Do.	14·0	0·82
Do.	18·0	0·78
Do.	19·1	0·75
1200	21·5	0·71

Fuel used: Pratt's spirit.
Calorific value (lower) by Janker's Calorimeter= 17,800 B.T U. per lb.

Table IV.

Revs. per min.	B. H.P.	I. H.P.	Mean Effective Press. lbs. □"	η P.	Mechanical Efficiency %.
400	7·8	9·0	81	69·7	86
600	11·8	14·0	84	70·5	84
800	15·7	19·15	86·2	70·7	82
1,000	19·5	24·7	89·0	70·3	79
1,200	22·5	30·0	90·0	67·5	75
1,400	23·0	34·8	89·0	58·0	—

Table I.—Details of tests of the Lamplough two cycle engine.

Table II.—Doctor Watson's observations in testing the Day two cycle engine.

Table III.—Fuel consumption tests of the Dolphin two cycle engine.

Table IV.—Other tests of the Dolphin engine.

port referred to in paragraph 4 of the paper mentioned, Dr. Watson states, "in the experiment on the four cycle engine that whenever there was carbon monoxide in the exhaust there was no free oxygen." That means to say that all the oxygen present combined with a hydrocarbon, and that, as far as the air was concerned, combustion was complete; and he goes on further to say that "as a test of the accuracy of the deductions which can be made by his method we may calculate what would be the composition of the exhaust gases supposing we were able to eliminate the proportion of the gases which escape combustion." Dr. Watson's method of calculating the proportion of mixture escaping unburnt is based on the supposition that under certain conditions the oxygen in the air is completely combined during the firing stroke and, although I do not wish to enter into any controversy with Dr. Watson upon this point, I express it as a matter of opinion only that in a short stroke two cycle engine combustion is not so complete. The important bearing of this matter upon the calculation of the proportion of unburnt charge escaping cannot be too strongly emphasised.

As the cylinder contains a larger proportion of inert gas at the end of the compression stroke in many two cycle engines than in the average four cycle engine the presence of this inert gas may delay combustion in the short stroke engine and permit a larger proportion of oxygen to escape than would be the case under normal conditions. I have had a considerable number of chemical analyses of exhaust gases made from the Lamplough engine, and even now I am unable to come to any definite conclusion on this point. Furthermore I have run the engine with various excesses of air, and I give on page 399 a table of some of my results, and should be glad if Dr. Watson or any other investigator has the time to offer any explanation as to the results obtained. One noticeable figure is the low percentage of CO_2 which does not appear to bear any direct relation to the excess of air. The column showing the loss of charge is calculated by Dr. Watson's method, but until I have made further investigation I am not prepared to accept the high figure shown.

In Dr. Watson's table, for example, Test 20, his engine indicates 4.29 h.p. and gives on the brake 3.3 h.p., the petrol consumption being 0.0498 lbs. per 1,000 revolutions per minute. In order to make this figure comparable with my figure for the Lamplough engine I take 8 pints of petrol as weighing 7.2 lbs. and calculate as follows :—

$$\frac{0{\cdot}0498 \times 1{\cdot}218 \times 60 \times 8}{3{\cdot}3 \times 7{\cdot}2} = 1{\cdot}22 \text{ pints.}$$

Calculating the percentage of charge escaping unburnt we may take it that 21.3% of the total air is represented (by volume) as oxygen. Of this total, in Test 20 above, 4.2 appears as O_2 in the exhaust, so that 21.3—4.2 = 17.1 volumes only of O_2 were burnt.

The proportion of escaping O_2 to the total O_2 is represented by $\frac{4{\cdot}2}{21{\cdot}3} = 0{\cdot}197$ or as a percentage 19.7, the figure given by Dr. Watson.

Calculating now for excess of air, we may proceed in the same manner, and still taking the same test, No. 20. The amount of CO is 5.6%, and to burn it to CO_2 $\frac{5{\cdot}6}{2} = 2.8\%$ O_2 is required. The total O_2 present is 4.2%, so that 4.2—2.8 = 1.4% O_2 out of a total of 21.3, or in round figures say 22.

Therefore $\frac{1{\cdot}2 \times 100}{22} = 5{\cdot}5\%$ excess of air only is present in this case. All the figures which I give in the foregoing tables are worked out in the same manner.

Now we come to the last column giving the mean effective pressure as referred to the b.h.p or ηp and one example for the Lamplough engine will suffice.

Take the cylinder dimensions as 2.5 ins. diameter by 3.5 ins. stroke the area of four pistons

$= 4 \times 2{\cdot}5 \times 2{\cdot}5 \times 0{\cdot}785 = 19{\cdot}5$ sq. in.

and working test C 12b we have a b.h.p. of 15.0, produced at 1,370 r.p.m.,

$$\text{therefore } \eta p = \frac{33{,}000 \times BHP}{A \times L \times N}$$

$$= \frac{33{,}000 \times 15 \times 12}{19{\cdot}5 \times 3{\cdot}5 \times 1370}$$

$$= 63{\cdot}2 \text{ lbs. sq. in.}$$

In the above case N = the number of revolutions per minute as the pistons receive an impulse every revolution.

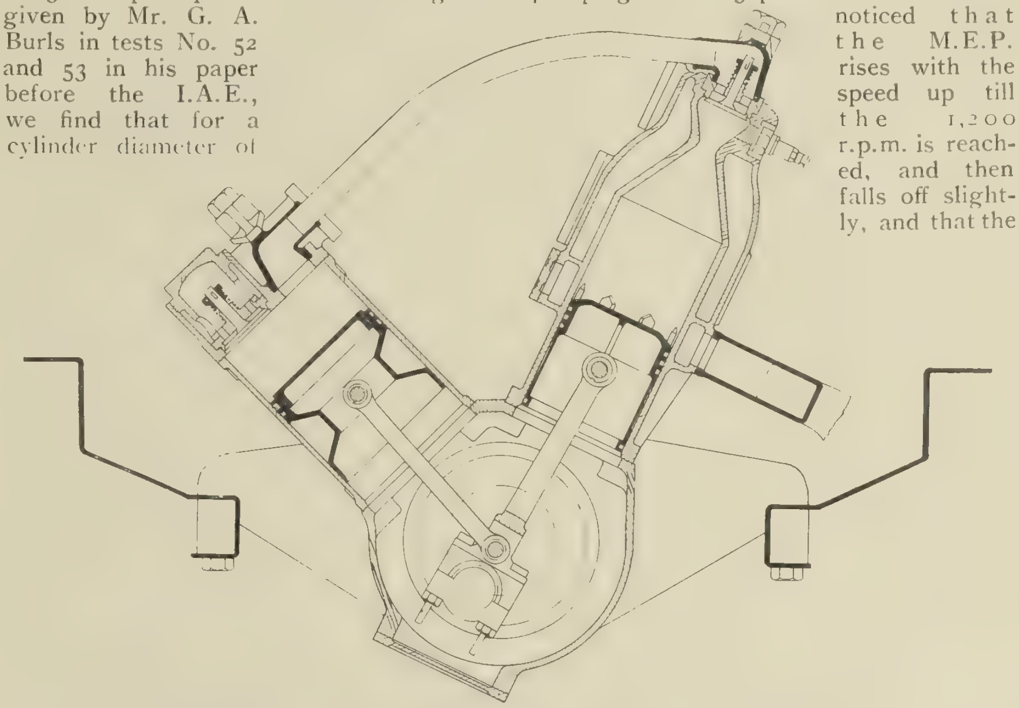
If we now compare this value of ηp of 63.2lbs. per square inch with the figure given by Mr. G. A. Burls in tests No. 52 and 53 in his paper before the I.A.E., we find that for a cylinder diameter of 2.4 inches in the first place the value becomes 61.2 lbs. per square inch with a 10 h.p. engine, and in the second case, test 53, the cylinder diameter is 2.56 inches, the horse power is 11.5, and the value of ηp = 63.8 lbs. Making a comparison therefore, we find that the two stroke engine shows up very favourably against the four stroke engines referred to in Mr. Burls' tests. As the engine in question is in no way a racing model, and no special care is taken to lighten the reciprocating parts, we will make a comparison of the value for this pressure with the results obtained by Mr. Poppe's formula in the calculation of brake horse power where b.h.p. = 0.81 (d-0.79) 2, and taking Mr. Burls' equation where σ denotes the piston speed in ft. p.min. b.h.p. per cylinder $= 1/68000\ d^2 \eta p \sigma$.

Mr. Poppe's formula therefore gives $\eta p = 136 \times (1-0.7/d)\ 2$.

So that for a cylinder diameter of 2.5 inches ηp 63.5 lbs. per square inch for a piston speed of 1,000 feet per minute as against 63.2 lbs. per square inch at a piston speed of 800 feet per minute in the Lamplough engine.

Referring again to the table for the Lamplough two cycle engine the loss of charge appears very high when worked out by Dr. Watson's method, but if these figures are re-calculated upon the R.A.C. standard of 2 per cent. of oxygen in the exhaust gases a very different percentage appears. Test C10 for the engine running light would now show 20 per cent. loss of charge, test B8 15.6 per cent.; test C 13 9.5 per cent., and test B 6 14.3 per cent.

Before concluding this article I take the opportunity of giving a few figures referring to the two cylinder Dolphin engine of 20 nominal h.p., the cylinder bore being 3¾ ins., and the stroke 5 ins. The engine has its working and pumping cylinders set at a vee as indicated in the accompanying diagram. With reference to the mechanical efficiency of this engine, taking the figure given of 75 per cent. at 1,200 r.p.m., the losses are made up of frictional losses 1 per cent., and pumping losses 13 per cent. It will be noticed that the M.E.P. rises with the speed up till the 1,200 r.p.m. is reached, and then falls off slightly, and that the fuel consumption per b.h.p. steadily falls as the b.h.p. increases when the engine is running at 1,000 r.p.m., and it is still less when the speed is increased to 1,200 r.p.m.



Sectional diagramatic view of the Dolphin two-stroke engine.

It must not be supposed that any sort of finality in two cycle engines has been reached, but the intention of this article is to attract the notice of automobile engineers to the fact that a large development has lately been made in two cycle practice, and will continue to be made during the next year or so by those who are of the firm belief that the two cycle engine is at last merging from a totally unsatisfactory state of a too great simplicity into that of an engine with large prospects, if indeed it be not the engine of the future.

THE 16-20 H.P. SUNBEAM CHASSIS.

This has a number of interesting details, an unusual gear box suspension, and a particularly strong and simple back axle.

THE 95 mm. x 135 mm. Sunbeam chassis is quite a distinctive design, as it departs in many ways from the standard practice of the 80 mm. x 120 mm classification, although it very closely resembles the other Sunbeam cars with these dimensions. Notwithstanding the extremely high efficiency of most modern 80 mm. engines, some larger cylinder dimensions are really to be preferred for the ordinary work to which a touring car is put, because the larger an engine is the less often it needs to be used at its full power. For this reason, perhaps, the 16-20 h.p. Sunbeam is particularly smooth running and comfortable at all ordinary road speeds when the engine speed is, of course, comparatively low. With an open body there is an ample reserve of power for fast hill climbing, and also the engine is capable of rates of revolution almost as high as that of its smaller prototype, so that a very high horse power indeed can be developed if necessary.

The Sunbeam company is one of a very few firms who still pin their faith to the tee head cylinder casting with the valves on opposite sides and two camshafts. This construction has, of course, many advantages on the score of accessibility, and its only real disadvantages are its necessarily greater cost by comparison with the one-sided arrangement, the presence of another pocket, and (perhaps still more important) the need for an extra camshaft driving gear, which means that it is not quite so easy to make the engine quiet. However, the timing gears in this particular engine are not noticeably noisy, and the drawback in this case is chiefly cost of production.

The cylinders are cast in pairs, and are a very simple foundry job, being completely open at the tops, and having very large water jackets round the valve pockets, so avoiding small and intricate coring. One small point which should be noticed is that water is carried completely round the valve caps keeping both these and the sparking plugs well cooled. A peculiar point in connection with the cylinders is that they do not spigot in the crankcase, being solely positioned by their holding down bolts. This is a debatable point, but, where the jigging is good, as is the case in the Sunbeam works, there is no strong objection to the absence of the spigot, while the machining of the cylinders and crankcase is, of course, very considerably simplified.

The valves are exceptionally big, being 50 mm. in diameter at the smallest part of the head, whereas the average is certainly not in excess of 45 per cent. of the cylinder diameter, which should make the normal for a 95 mm. engine almost exactly 43 mm. The large radii under the heads should be noticed, and it may be added that, in the racing engines, the central portion of the head is drilled out from above for the sake of lightness. Very curiously, because it is a strange defect in an otherwise well-thought-out design, no tappet adjustment is provided, and the tappets are of the intermediate lever type now very seldom used. The arrangement as a whole has the advantage of simplicity and it is not expensive to manufacture, but it is probably no cheaper than the ordinary design because, although both the tappet itself and its

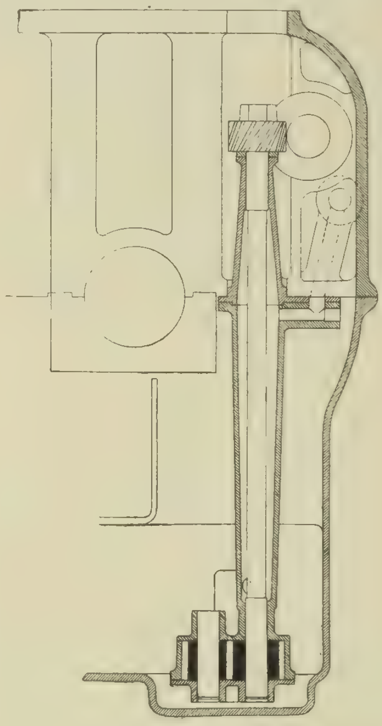


Fig. II. Section of crankcase showing oil pump arrangement.

guide are extremely simple, the fixing of the intermediate levers is so much additional work, while the total number of parts to be machined is quite as high. It is also more difficult to obtain quietness in valve operation with this arrangement than with a roller-ended tappet of the ordinary type. It is not meant to imply that the Sunbeam valve gear is really noisy, because, owing to the cover plates and the light weight of the valves, they are not more than ordinarily audible.

Both the camshaft and crankshaft are provided with three bearings only, and both are stampings of Vickers oil hardening crankshaft steel, the diameters being —camshaft 22 mm. and crankshaft 44 mm. The bearings are die castings of a particularly hard white metal, those for the crankshaft being 80 mm. in length, with the exception of that nearest the flywheel, which is some 20 mm. longer.

Great lightness of reciprocating parts has received special study, and the lantern form of piston shown is the standard for this engine, the material used being malleable cast iron. It is to be doubted whether the scraper ring beneath the lantern orifices serves any particularly useful purpose, as it is usually found that cutting away the skirt is in itself sufficient to prevent the passing of too much oil to the combustion chamber. However, the extra ring can do no harm, and is an additional safeguard at all events. In the section of the piston shown in Fig. I. the gudgeon pin fixing can be seen, this consisting of a small grub screw in the pin, which prevents its rotation, and a thin steel ring slipped over the ends. The pin itself is a fairly tight fit, and the steel ring is only added to prevent cylinder wall injury in the unlikely event of the pin shifting. This was the device which was used on the first 8-10 h.p. Humbers, and was always extremely satisfactory. There is unusually good bearing surface on the gudgeon pin, the diameter being 24 mm. and the width of bearing 50 mm. Like the valves, the connecting rods are nickel steel stampings, and they do not require any special description.

It is noticeable that there are no ball thrust bearings to take the endwise pressure of the timing gears, but each camshaft is provided with a hardened steel thrust button, one of which can be seen in the side sectional elevation in Fig. I. Lubrication of the main crankshaft bearings and the big ends is performed entirely by pressure, the pump being situated midway in the crankcase at the near side of the sump. There is a separate view in Fig. II. showing a section through this pump, which is of the gear type, but acts in an unusual way. The oil is filtered through the central gauze box as it passes from the crank chamber into the sump and through another gauze at the pump intake. After passing through the pump it is forced up in an annular space surrounding the driving shaft which is, in reality, an extension of the pump casing, being part of the same phosphor bronze casting. From this annular chamber the oil goes to the drilled hole in the upper half of the crankcase, thence to a longitudinal cored channel, and thence to the main bearings, the whole path being traceable easily in Fig. II. Needless to say, neither the pump nor the filter can be removed without loss of oil, but the defect is shared by almost all other engines and is, perhaps, not too serious, because, when the filter needs cleaning it is probably time to replace the old oil by an entirely fresh supply.

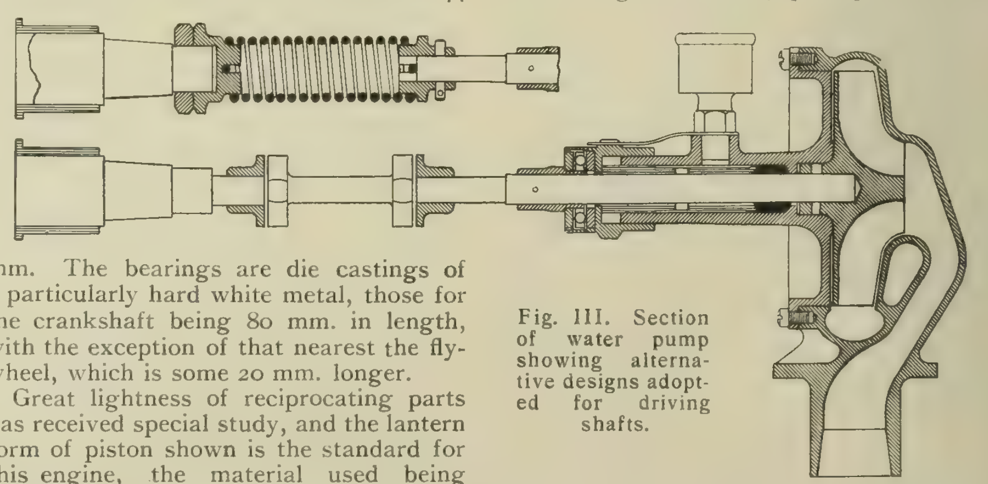


Fig. III. Section of water pump showing alternative designs adopted for driving shafts.

The magneto and the water pump are

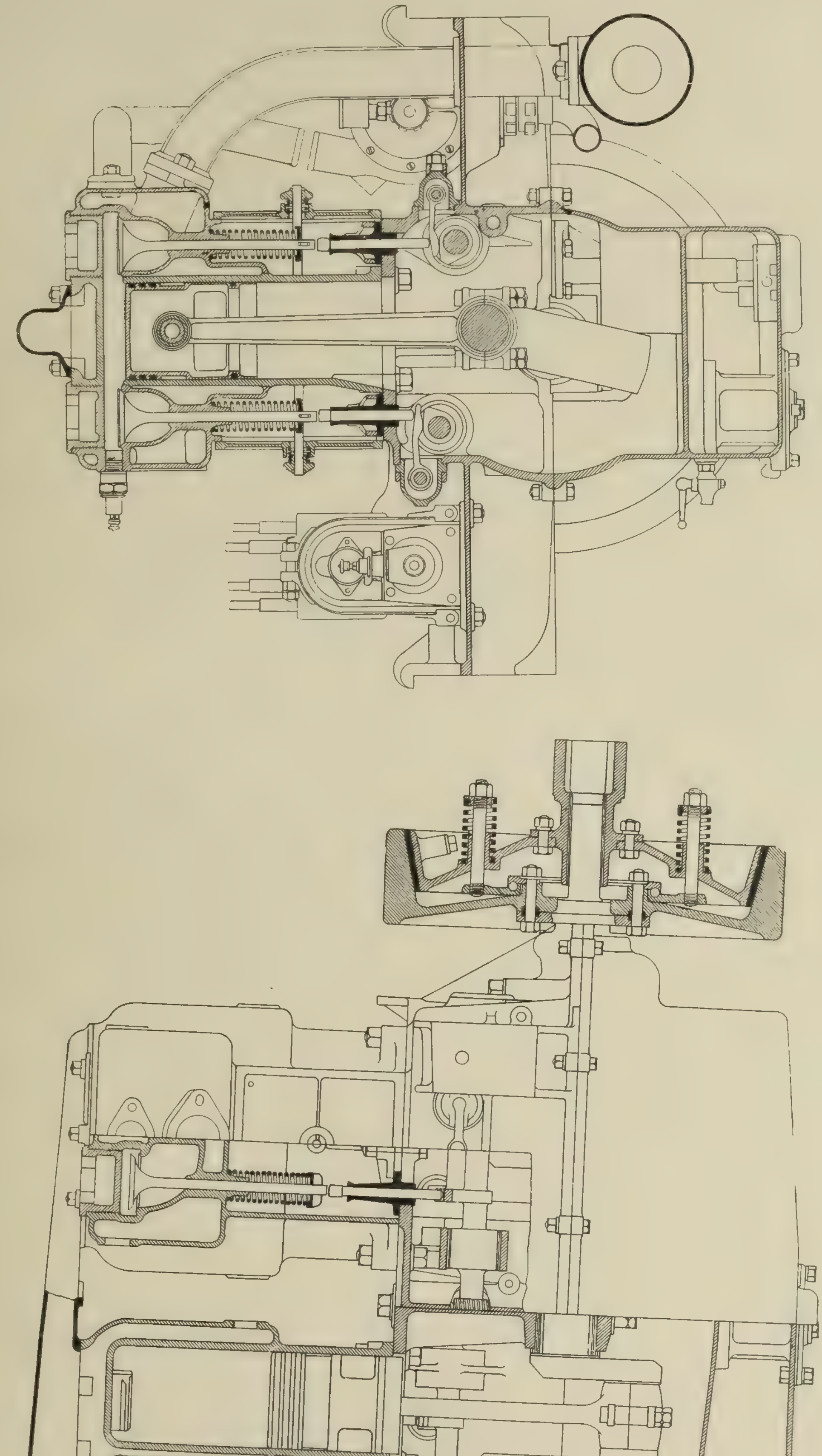


FIG 1. THE 16 H. P. SUNBEAM ENGINE.

This illustration shows the extent to which the engine is inclined in order to obtain a normally straight line drive to the worm, which is situated beneath the worm wheel in the back axle.

situated on opposite sides of the engine towards the front end, and are driven by means of additional gear wheels meshing with the large timing wheels. For the pump drive a rather neat form of spring coupling is used, and this is shown in Fig. III., together with a section of the pump on a larger scale, showing that the latter is of the vane type, while a point which should be noticed is the situation of the greaser, behind the packing and so well away from the water. Adjustment for the gland is provided by the threaded cap with spring trigger locking gear, and the ball bearing takes the end thrust of the drive, so eliminating any risk of the pump vanes fouling the box should slack develop by neglect to adjust the gland.

Concerning the crankcase it is only necessary to say that, regarded as a foundry job, it is simple, and the machining is still further simplified perhaps, by the method of housing the tappet lever arms in small brass dogs bolted to the aluminium.

Leaving the engine, the clutch requires a little explanation additional to that given in the illustration, the two springs being carried on the aluminium cone-piece, while the inner plate carries the bolts from which the thrust is taken. This arrangement gives a very large area of ball bearing to take the thrust when the clutch is held out, and there is, of course, no resultant end thrust with the clutch engaged. The divided spring arrangement is advantageous on the score of accessibility, for it renders the removal or replacement of the clutch very simple and quick. It is perhaps a little more difficult to adjust than the usual single spring, but this is not so for any but the most amateur mechanics.

Between the clutch and the gearbox there is a De Dion type coupling, made a little peculiarly. This is shown separately in Fig. IV., and perhaps requires some explanation. The steel piece which is bolted to the shaft, is forked, and carries the bearing surfaces for the inner part. To prevent the latter falling out, a divided cover encircles the fork, being itself prevented from longitudinal motion by a ring in the main piece, while the two halves are kept together by an external ring, which may be slid off sideways and is secured by two set screws. Thus it may be said that, on removing these two screws, the joint falls to pieces without the further use of tools, and this naturally adds still further to the ease of detaching the clutch. Probably the most notable point about the gearbox is the size of the bearings and of the layshaft. Inside the spigot a specially made roller bearing is used, and this should give a rigid as well as a free running support to the forward end of the layshaft. For the gears Vickers case-hardening steel is employed, and the teeth are six diametral pitch in every case. Both the first and second speed wheels are made solid with the layshaft, and the latter would seem to be unnecessarily heavy and perhaps expensive as well. It might certainly be as strong and would be cheaper if the gears were separate pieces attached by dog tooth distance pieces in the usual way: still no complaint can be made with it on the score of strength or rigidity. The following are the speed ratios, and the reverse is given by bringing a double gear into communication with the first speed pair :—

First speed	15.3 to 1
Second speed	8.69 to 1
Third speed	6.45 to 1
Fourth speed	3.5 to 1

the worm proportions being 8 threads to 28 teeth. Great care is taken to prevent leakage of oil, the pieces seen at each end of the main shaft immediately inside the stuffing boxes acting as thrower rings for returning oil to the box.

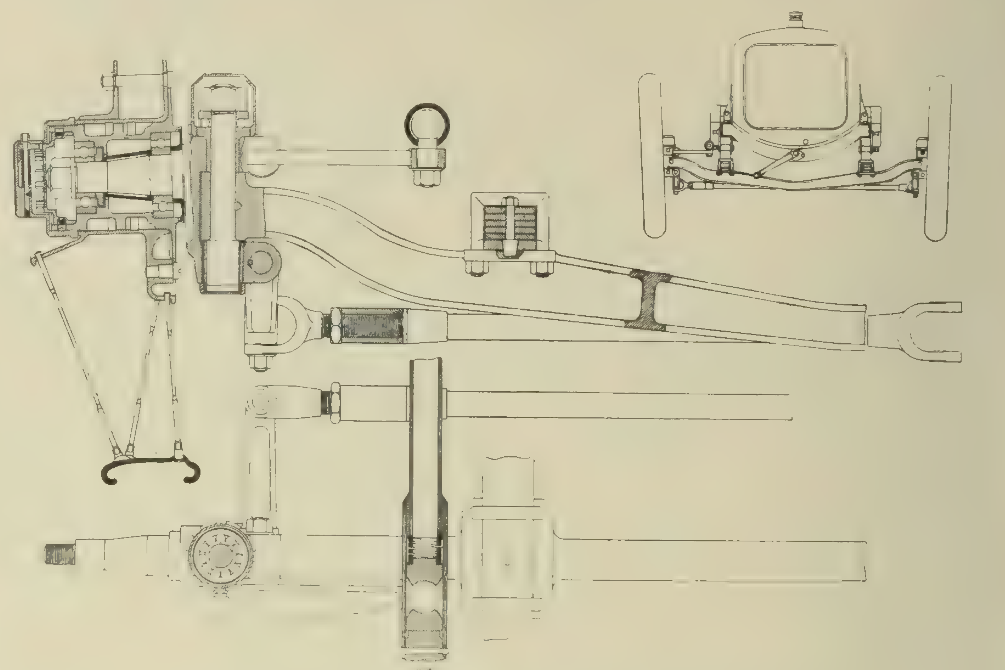


Fig. VI. Sunbeam front axle and steering arrangement.

The change speed control is locked by a pendulum arrangement as is made clear in the illustration, but the change lever shaft, instead of being mounted on a tube as is usual, is secured to a shaft turning and sliding on phosphor bronze bearings at each end of the tube. The tube is used for the application of the brake, and, owing to the bearings, there is no tendency for tube and shaft to stick, so the sideways sliding of the shaft is always perfectly free. Though perhaps not quite so easy to handle as a hinged arrangement, this device has the advantage of simplicity and low cost, while it is possible for even a hinge to bind if it is allowed to become encrusted with mud. The chief drawback is that the brake lever is necessarily brought inside the change speed lever, but this is a point of extremely small importance, and one which is certainly not noticed in driving, even to one unaccustomed to the car. In the chassis plan and elevation, the method of supporting the box in the frame can be seen, the two arms observable in Fig. IV. being slung from a cross-member by means of a pair of eye bolts and cross bolts, while the front end of the box hangs from one other bolt. This arrangement is extremely convenient in a shop, because it makes lining up so simple, and it is also an advantage when taking down the box. The chief point which can be raised against it is that, if the bolts are insufficiently strong, the box has a tendency to move about or vibrate irregularly in the frame when the full power of the engine is passing through it. When observing the behaviour of one of these cars on the road particular attention was given to this point, and there was certainly no noticeable movement and no sense of vibration, while the makers are perfectly satisfied as to the rigidity. It seems, therefore, safe to assume that the question is merely one of size, and that a three-bolt suspension can be quite satisfactory if well designed. The internal expanding foot brake, the universal joint and the cover of the latter need no explanation.

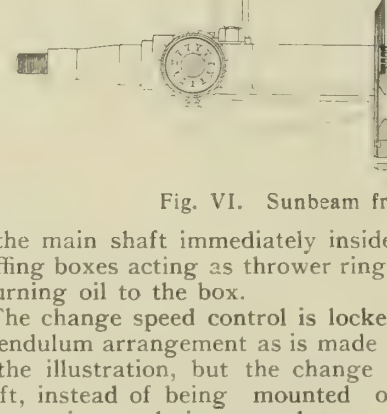


Fig. VII.

The next point requiring mention therefore, is the back axle. This derives its strength from a pair of malleable iron castings, strengthened by a tie rod, but the road wheels are carried on steel sleeves socketed in the ends of the castings, the way in which this is done being shown clearly in Fig. V., in which it may be remarked the hub is designed to take a detachable wheel. The worm casing is also malleable iron, bolted to the cast sleeves and divisible in a central vertical plane, while positive control in both directions is given to the brakes. The brake rods are mounted Rover fashion, being socketed in the differential casing at one end and then operated by means of a toggle instead of the usual cam. As we have already remarked, the worm has eight threads, and is straight sided, cut from the same piece of steel as the shaft and meshed with a phosphor bronze wheel. Here again the size of the bearings is noticeable, particularly the double thrust bearing, and the adjustment of the latter is placed in as conveniently accessible a position as such an adjustment very well could be. The large gland at the forward end of the worm spindle is also adjustable with but little difficulty, and it is, of course, necessary to keep this fairly tight owing to the worm being

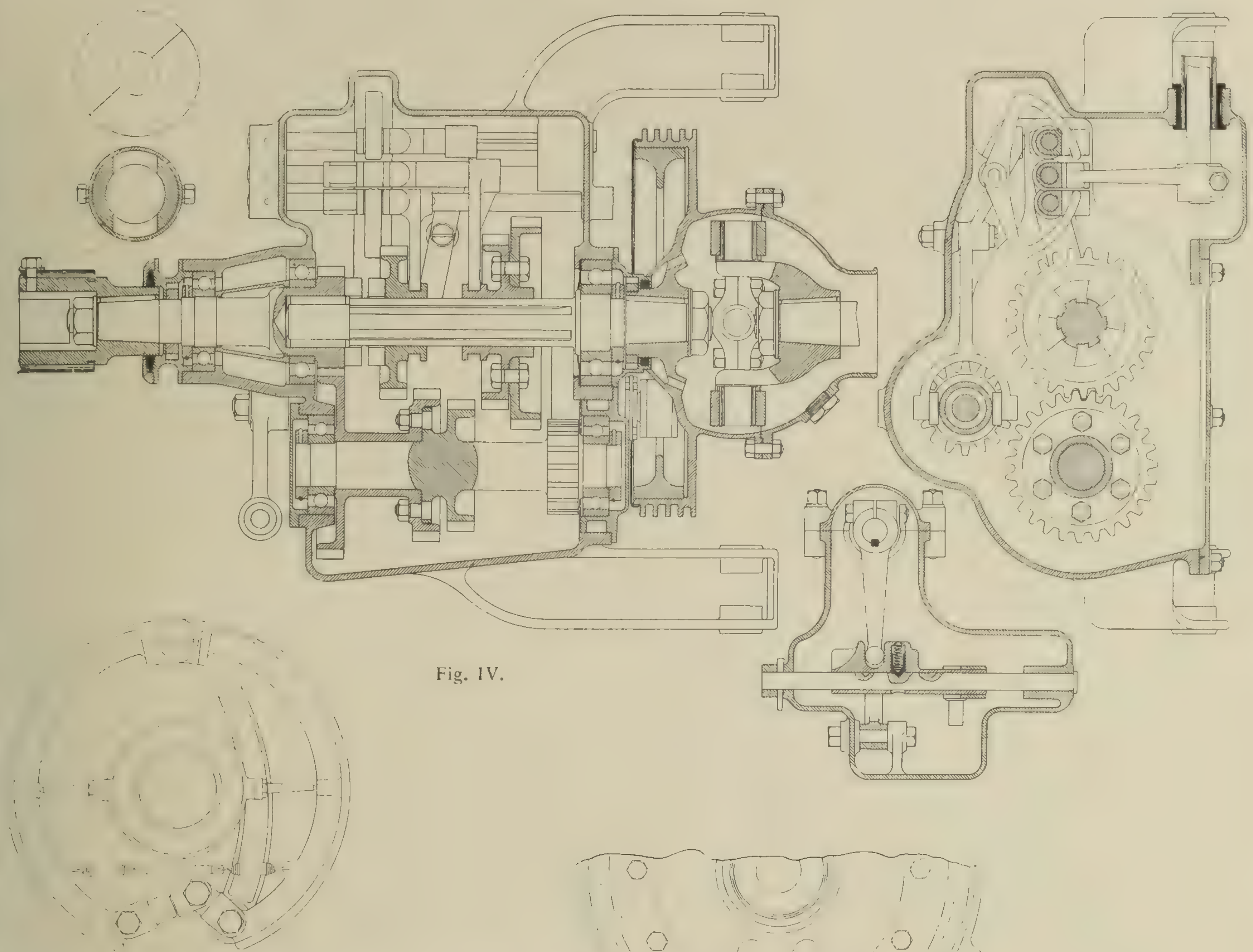


Fig. IV.

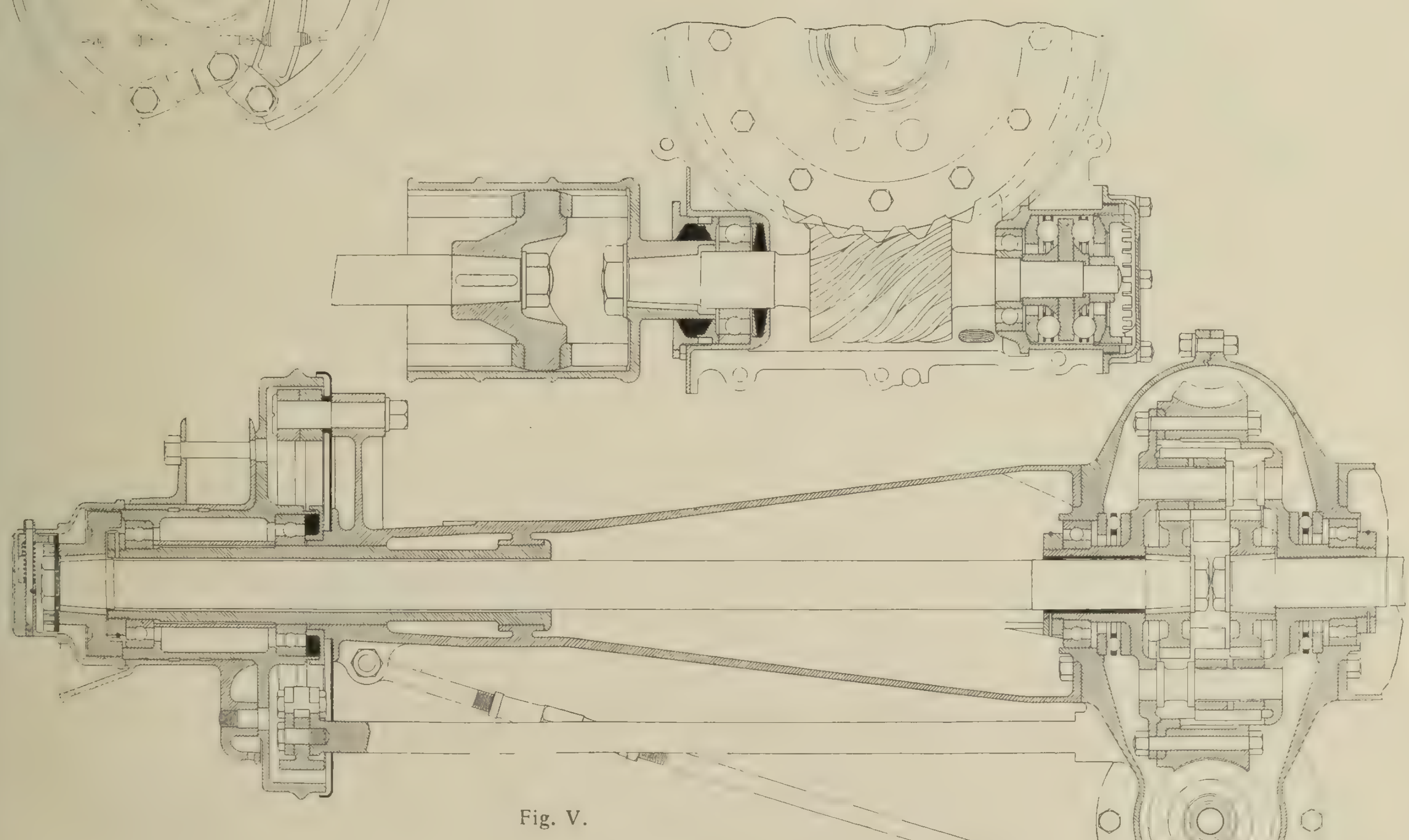
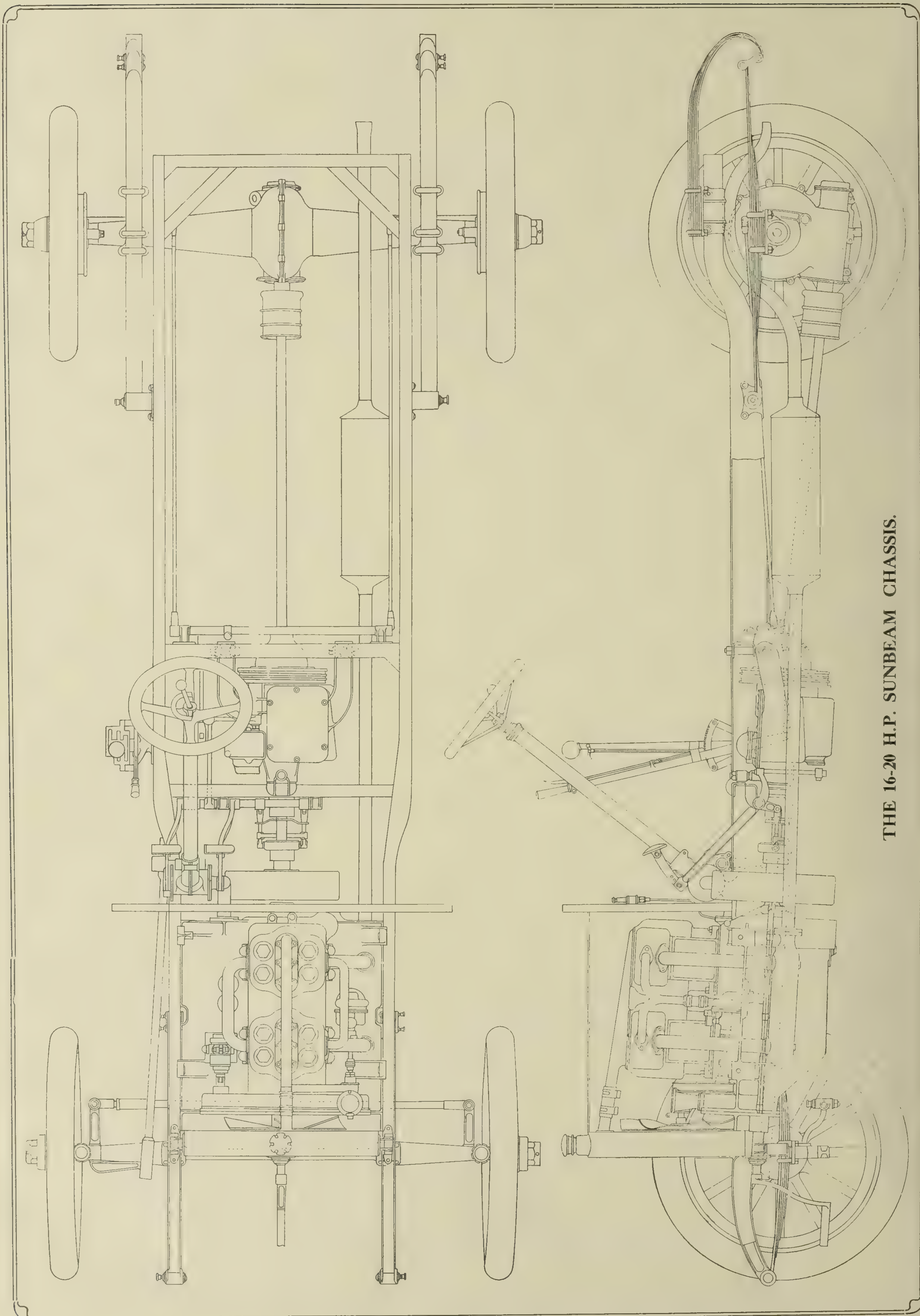


Fig. V.

Transmission details of the 16-20 h.p. Sunbeam Car.



THE 16-20 H.P. SUNBEAM CHASSIS.

beneath the worm wheel. To prevent the accumulation of oil in the sleeves, small holes are drilled through both the sleeve and central casting to allow any oil which finds its way through the differential ball bearings to fall back into the sump surrounding the worm. It is perhaps to be regretted that no adjustment is allowed for the thrust races on either side of the differential box, but otherwise the axle is certainly exceptionally strong, albeit perhaps somewhat heavy.

Turning now to Fig. VII., which shows the front axle, a point about this part is the situation of the ball thrust bearing which is at the top instead of at the bottom of the steering pivot. Here it is much better protected from water, being entirely covered by the brass cap, and its ample lubrication is practically ensured by the provision of a cap, which is filled with grease occasionally. The adjustable end of the tie rod is an unusual feature, but one which possesses advantages from the erecting shop point of view (and might conceivably be useful in case of slight accidental damage on the road) while the pins of the steering joint are certainly well above the average of size. The simple form of the separate steering arm should be observed, in fact, the whole steering shaft is very clean, every link being straight, throughout from the worm segment arm to the front arm.

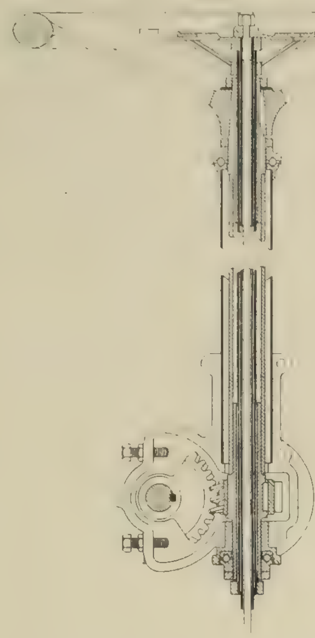


Fig. X.

The Sunbeam steering gear showing the peculiar bearings and the control levers.

The steering box is mounted on the frame by a neat and simple bracket shown in the sketch Fig. VIII., and also in the chassis, plan Fig. IX., though it appears to much greater advantage in the former. Although this arrangement necessitates the removal, or at any rate the partial detachment of the steering gear before the engine can be lifted up, this is a matter of very small moment and is more than compensated for by the rigidity with which the worm-box is supported. Fig. X., which gives the section of the gear, has been included particularly to show the peculiar type of ball bearings used to support the worm spindle at the top and bottom only; otherwise this portion of the mechanism is not deserving of special mention—with the exception possibly of the control levers which are shown in section, and are rather neatly arranged. The quadrant is, of course, fixed, and does not revolve with the wheel, while the control levers are brought up beneath the sector. On each lever and inside the quadrant there is a small projection to which a piece of flat spring steel is rivetted so that the latter presses down on top of the sector thus gripping it. There are no teeth, but the device provides sufficient protection to hold the lever in position while being very easy and smooth to move, and satisfactory in working.

THE RATING OF PETROL ENGINES.

By C. F. Dendy Marshall, B.A.

IT is agreed, for various reasons which I need not go into, that it is impossible to devise an h.p. formula which will do more than give an approximation. Consequently anything in the nature of complications should be avoided, unless they are justified by unmistakable superiority. The same consideration applies to factors like d + a constant, where d is the diameter, and also to roots of the stroke, etc., which cannot be defended on any ground except that they profess to be based on the careful observation of different engines. I use the word profess advisedly, because anything more inconclusive than the first three diagrams in the recently issued report of the Horse Power Committee I cannot well imagine. The dots are literally all over the place, yet they are the record of observations from which the highly empirical formula proposed has been deduced.

There is an exceedingly simple formula which, under average conditions, is capable of holding its own with all the more complicated ones that have been put forward. By "average conditions" I mean within a fairly wide limit of engine speeds, for, if the engine is being overdriven it gives too high a result, but it does not seem to be limited to any particular range of dimensions, as far as engines have at present developed. It has been produced by several independent investigators, of whom I was not the first, though it is frequently associated with my name. I refer to $\frac{n\,s\,d^2\,r}{12{,}000}$, or $\frac{n\,s\,d^2\,r}{200{,}000{,}000}$, according to whether the dimensions are expressed in inches or millimetres. Here n is the number of cylinders, s the stroke, d the diameter, and r the number of revolutions per minute. It is derived from the "PLAN" of steam practice, and assumes the ηp, or effective mean pressure corresponding to the brake horse-power, to be 84lbs. per square inch. η is a fraction representing the efficiency of the engine, and effects the change from i.h.p. to b.h.p. It may be objected that this assumption of the mean pressure is a large one to make, but the conditions are very different to the enormous range of pressures in steam engines of different kinds, and this is a fair average, and is, I think, fully justified by the results obtained. If the revolutions are unknown, and an idea of the power that would be obtained from a given engine on the road is sought for, they may be assumed at 1,000, and the formula becomes $\frac{n\,s\,d}{12}$, or $\frac{n\,s\,d^2}{200{,}000}$. Strictly speaking this assumes that the product of ηp and the number of revolutions is 84,034.

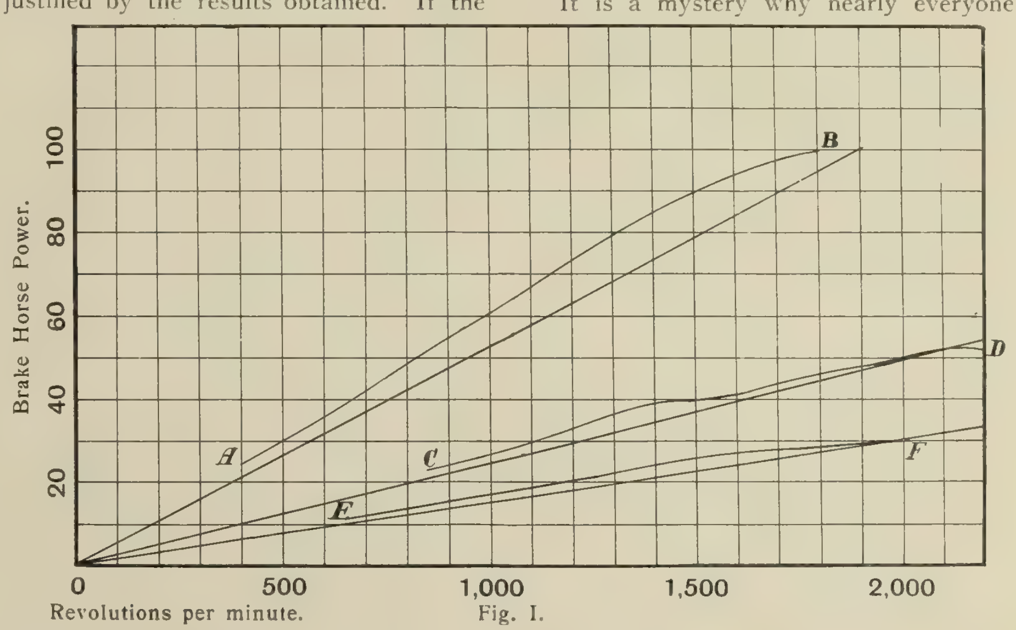


Fig. I.

This expression is superior to that of the R.A.C. $\left(\frac{nd^2}{2\cdot5}\right)$ because, although the latter is considered on theoretical grounds by its supporters to take into consideration both stroke and m.e.p. the practical result of it is that two engines with the same diameter and different strokes are rated the same, which is absurd, more especially in these days when the tendency is to increase the length of the stroke.

It is a mystery why nearly everyone who invents a formula should treat the stroke in such a cavalier fashion; sometimes they practically ignore it, as in the case of the R.A.C. Others notice it to the extent of taking its root, and the report that has just been issued is full of the ratio of stroke to bore, as if it were preferable to work with that, rather than the stroke pure and simple. This sort of

thing has never been suggested, so far as I know, in connection with steam engines, so why should it be necessary with explosion engines?

As a matter of fact it should be much easier to devise a formula for the latter, because the whole power is both developed and utilised in the cylinders, whereas in a steam engine it is developed outside. And there is nothing like the enormous variation of design, dimensions, initial pressure, and rate of revolutions, etc., that obtain with internal combustion engines.

The power here is produced by a certain volume of gas, which is equivalent to d^2s, if the correction for the fact of the piston being round instead of square is made by adjusting constants.

The Committee's formula is put forward as "the maximum practicable b.h.p., as determined by a bench test under onerous but safe conditions." I gather it is based on the figures given by Mr. G. A. Burls in the paper accompanying the report. Now, it so happens that in his tables we can see very nearly what is the maximum power that has been obtained from many of the engines. For shortness, I will call the Committee's proposed formula—i.e., $.45\ (d+s)\ (d-1.18)\ N$—A; Mr. Dugald Clerk's modification of this, in which he multiplies it by .6, or, in other words, substitutes .27 for the first factor—B; and the nsd^2 formula I shall in future designate as C.

Take engine No. 1. It developed by test at a piston speed of 1,275 feet per minute, 37 h.p.; at 1,535 and at 1,705, 41.5. We may take it, therefore, that 41.5 is about the most that can be squeezed out of this engine. The power by formula A is 67.2; by B, 40.3. By C it is 39.6 at 1,275, and 47.7 at 1,535, at which point the engine was already being overdriven. In the case of No. 2 we have 21 h.p. at 937 r.p.m., and 22 both at 1,205 and 1,340. Hence 22 is not very far from the maximum.

Formulæ: A, 63; B, 38; C, 22.4 at 937 and 29.5 at 1,205.

No. 3 is a large six-cylinder racing engine, developing 104 h.p. at 1,666 feet per minute, which is 2,000 revolutions. It is probable that this is not very much below the maximum.

Formulæ: A, 163.4; B, 98; C, 121.

No. 4 gave 45 h.p. at 1,082, and only 47 at 1,333, so it is clear little would be gained by driving this engine faster.

Formulæ: A, 103; B, 62; C (at 1,082), 48.4.

No. 5 gave 32 at 1,166, and only 33 at 1,417.

Formulæ: A, 48.5; B, 29; C (at 1,166), 39.8.

No. 6 produced 54 h.p. at 2,040, which is 2,300 revolutions, practically maximum.

Formulæ: A, 80; B, 48; C, 57.

In No. 14 we have an excellent indication of the maximum, as it gave 87 at 1,416 and 1,600, dropping to 86 at 1,665.

Formulæ: A, 155; B, 93; C (at 1,416), 95.

No. 15 gave 32 at 960, 32.7 at 983, and 30 at 1,035. Here again the limit is clearly shown.

Formulæ: A, 90; B, 54; C (at 983), 35.5.

No. 16 gave 37 at 1,160, and 37.5 at 1,320.

Formulæ: A, 107; B, 64; C (at 1,160), 47.

No. 19 gave 202 at 933 and 204.5 at 955. Possibly this engine might have been pushed considerably higher, but I have worked it out because it is such an unusual size, namely, 12 inches diameter by 8 stroke.

Formulæ: A, 474; B, 285; C, 275.

No 20 developed 108 at 1,120 and 109 at 1,200.

Formulæ A, 264; B, 158.5; C (at 1,120), 143. The last two engines are both considerably below Formula C at all speeds. The cause is possibly bad cylinder proportions. In steam engines it has been found best for the stroke to exceed the bore, and I think the same holds good for these engines.

Nos. 27 and 28 are similar engines, except that one has four cylinders and the

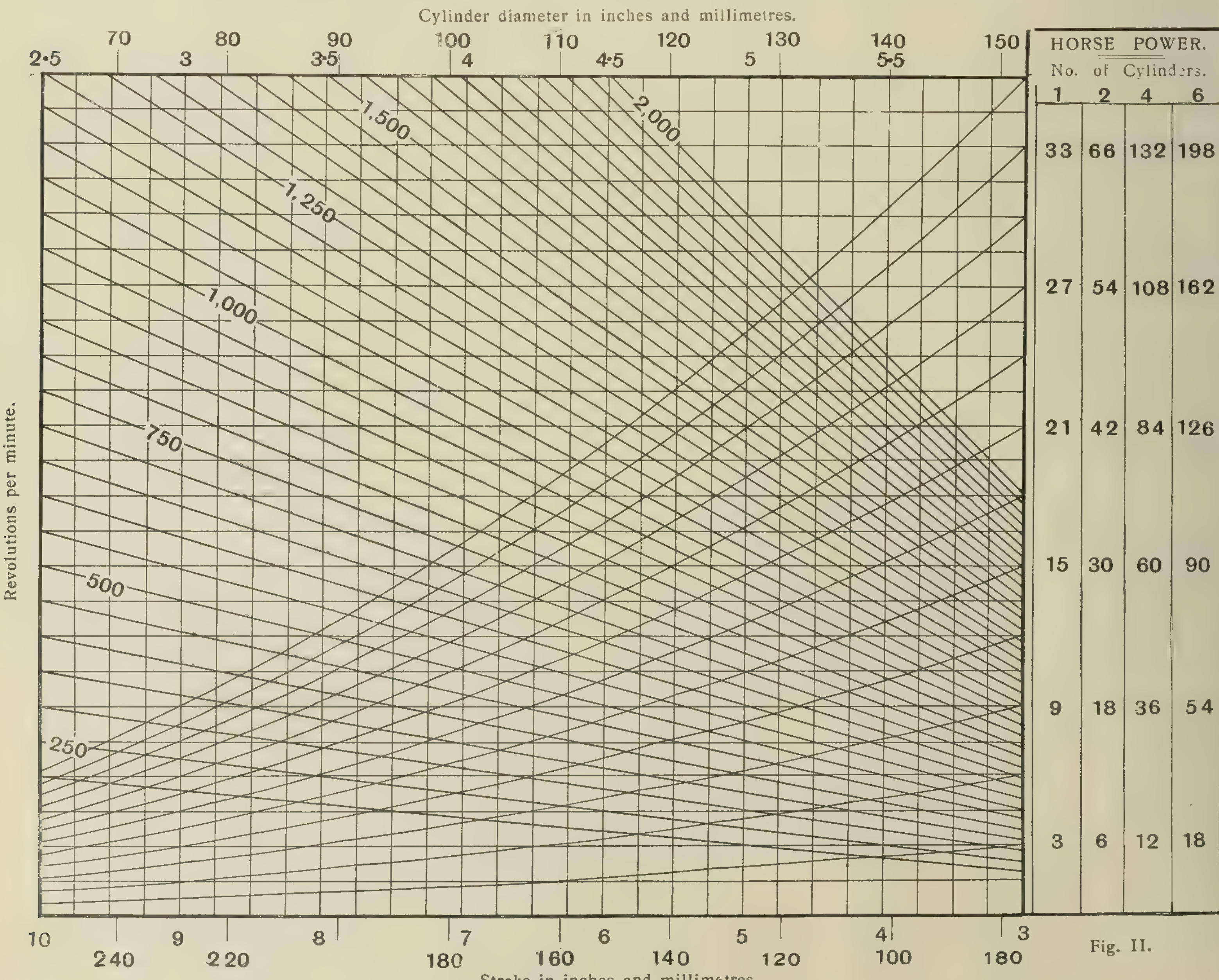


Fig. II.

other six. It is a very remarkable thing that at five different speeds the powers should be *exactly* proportional to the number of cylinders. This is a case of practice agreeing almost too closely with theory, and one is inclined to suspect the accuracy of the tests, especially the last one, which is at 2,500 revolutions, when one would expect the six-cylinder to have rather the best of it. At this rate they are being driven to death, as No. 28 gave 47.2 at 2,000, and only 41.2 at the higher rate.

Formulæ: A, 87; B, 52.5; C (at 2,000 revolutions), 45.7.

No. 29 gives us a maximum also, as the power was 37.5 both at 1,580 and at 1,900 ft. per minute.

Formulæ: A, 70; B, 42; C (at 1,580), 39.

The series 37 to 40 are extremely interesting, because they are engines with 1, 2, 4 and 6 cylinders all the same size. Here the figures are very close to the proportion of the number of units, without being absolutely correct. As they do not seem to have been driven up to their maximum power, I have not worked them out. In fact, I have done enough to show, from their own figures, that the formula proposed by the Committee is of very little use for practical purposes. It is true I have only taken some of the cases, but as I have simply chosen those which show clearly within small limits what is the maximum power that can be got out of the engines, there is no reason to suppose that those selected are specially unfair to the formula.

I give below four others, which are interesting engines, and show how extraordinarily well the nsd^2 formula adapts itself to widely different types.

I. Sunbeam car; 6 cylinders, 80 × 120 mm.: b.h.p. 46 at 2,000 revolutions.

Formulæ: A, 92; B, 55.2; C, 46. 1.

II. Panhard motor for Lebaudy airship: 4 cylinders, 185 + 200 mm.: b.h.p. at 1,000 revolutions by test 128, by formula C, 136.9.

III. Gnome aeroplane engine: seven revolving air-cooled cylinders 4.3 by 4.7 inches: b.h.p. at 1,000 revolutions, 50 by test, 50.7 by formula C.

IV. Westinghouse motor for petrol-electric railway cars: 6 cylinders, 140 × 160 mm. At 950 revolutions, 90 by test, 89.4 by formula C.

The results of all the above figures seem to be that the Committee's formula for maximum h.p. more often than not works out at about double what can be obtained on test; that Mr. Clerk's modification, which is supposed to be the power usually to be expected on the road, gives somewhere about the real maximum, and that the one I have always advocated is slightly too much at high rates of revolution. The reason for this is not far to seek. When an engine is being tested at various speeds, after it has got into its stride, the h.p. rises steadily with the revolutions, until a critical speed is reached, after which it goes up more slowly, finally coming down again. Fig. I. gives the characteristic curves of three petrol engines. The line AB is that of a six-cylinder Napier, cylinders 4 1-16 by 7 ins. By the Committee's formula it should be capable of developing over 156 h.p. CD is from a four-cylinder Sunbeam, 95 by 135 mm., and EF from a Crossley engine. The straight lines adjacent to the curves represent the nsd^2 formula for each engine. If it is desired to estimate the maximum power by this formula, and the revolutions are unknown, it can be done more or less accurately by assuming a number for them, based on experience, and the dimensions and design of the engine. Fig. II. is a chart giving the h.p. by the nsd^2 formula.

This is used as follows:—Enter at the bottom of the measurement for stroke, and run up until the correct revolution line is intercepted, then pass horizontally to the right and take the curved line which starts at the point of arrival. First, I will assume there is one just right. Go down it to the left until you are under the diameter on the top scale. The height of this point from the bottom gives the horse-power at the right side. If there is not a curve available, and it is necessary to interpolate, it is quite easy; you estimate the ratio in which the point arrived at divides the space between the two adjacent curves, then the required point below the diameter will occupy a similar position.

To take a practical example, the h.p. of the Sunbeam, my No. 1, is found as follows:—Enter at 120 stroke and pass up until you intercept the 2,000 revolution line, then go across to the right. This lands you just about midway between two curves. Follow the curves back until immediately beneath the diameter, 80 mm., and take a point just below halfway between them, then run across to the right, where you arrive at 46 h.p. It is worth pointing out that the exact equivalent for the constant 12 when millimetres are used is 196,634. The effect of taking the round number of 200,000 gives a result just under $1\frac{3}{4}$ per cent. lower, which is quite near enough for all practical purposes, a variation which, of course, is quite negligible, considering that an approximation is all that one can aim at in devising a formula.

THE 15 H.P. GERMAIN CHASSIS.

This design is peculiar in many respects and has a number of good points of detail.

BEFORE proceeding to discuss the details of design of the new 15 h.p. Germain chassis, it is proposed to make some mention of the works in which the car is produced, because they are interesting in broad comparison with factories of similar size in this country. Situated in the peculiarly dirty neighbourhood of Charleroi, in Belgium, the works are surrounded by *pavé* roads with a surface so execrable that the best sprung car is subjected to the most violent oscillation even at quite moderate speeds. As a testing ground these roads could scarcely be improved upon from the point of view of roughness, and this fact possibly accounts for certain details in construction of the chassis which will be mentioned in due course. The manufacturers were primarily interested in the building of rail vehicles of various kinds, and the automobile factory is an addition and a development of the original business. It is situated alongside the old works, but is entirely separate and can be regarded as complete in itself, although it benefits doubtless by the use of a finely equipped testing laboratory, which is common to both departments. The works themselves are of the single building, gallery divided type. Stores are arranged in the basement; on the lower two galleries most of the machining is performed, and on the upper two, erecting in various stages. Engine testing proceeds in a walled-off portion of the ground floor, and test cars are also accommodated on this level. Fitted with enormous windows and roof lights as well, the factory is one of the best illuminated by day, that the writer has ever been in, and the gallery arrangement whereby everything progresses upwards floor by floor, appears to work very smoothly. A single lift, of course, feeds all departments from the stores upwards, and as this is the only means (bar rough stairs which are not used) of transference from one department to another, the checking is extremely easy.

The machine tool equipment is in no way strikingly different to that generally found in works of similar size in this country; that is to say, there are a large number of American made tools and a fair sprinkling of German origin, although English products are noticeably absent. Perhaps there is rather less grinding done than is now common here, but otherwise there is no noticeable peculiarity in the manufacturing methods, and the most interesting feature of the whole place is undoubtedly, to the casual visitor, the material testing laboratory. This is equipped with a tensile machine, an impact machine and a hardness testing instrument, together with some smaller apparatus. The chief fitting, however, is a large micro-photographing outfit, and it is upon this that the greatest reliance is placed. The writer was given to understand that samples of material are tested with a frequency quite unusual here, and that — after the micro-photograph—the impact test seemed to give the quickest gauge of the quality of the material, or rather, perhaps, seemed to be the most liable to show up changes in a material supposed to be standard. This laboratory is, of course, comparatively old, having been in existence before the automobile factory had grown to be of much importance, and it might, in conclusion, be remarked that it is staffed as a laboratory rather than as a section of a works, being really in the position of an outside advisory concern run by fully qualified men.

Turning now to the chassis itself, our readers will doubtless be well acquainted with its general outline. Although the designers have for many years pinned their faith to separate cylinders with brass water jackets, they have this year carried the block casting idea probably as far as it could possibly be carried.

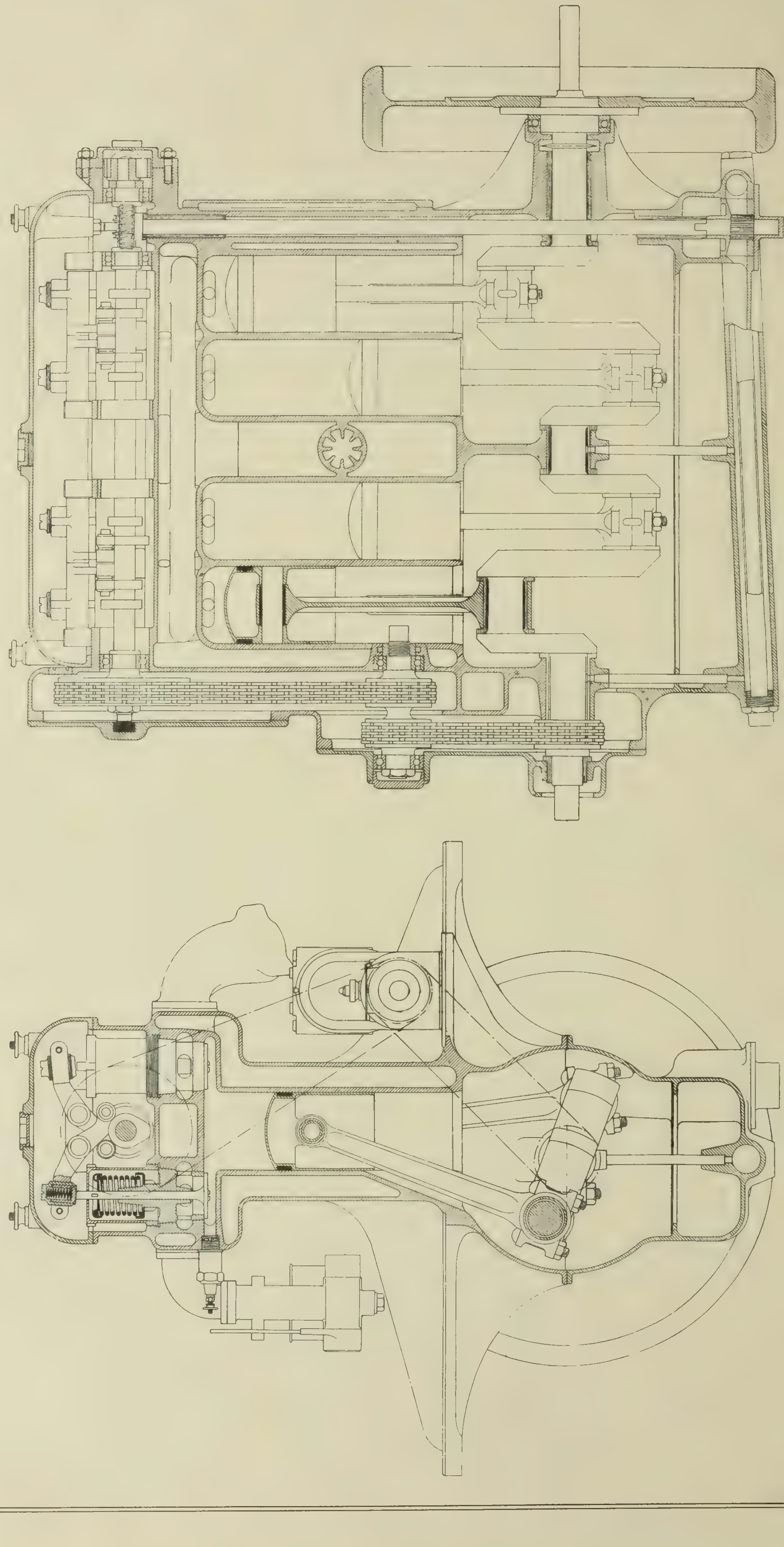


Fig. I.

THE 15 H.P. GERMAIN ENGINE.
Front and side sectional elevations.

Everything is enclosed in the main casting except the carburettor. A long stroke is used, the cylinders being 80 mm. in diameter, while the crank throw is 65 mm., and there are no gear drives in the engine, Hans Renold chains being used for the camshaft and magneto shafts, with a skew gear from the camshaft to the oil pump.

Fig. I. shows a sectional off-side elevation, and a section through the front cylinder with the centre lines of the chains dotted in. Taking the valve arrangement as a first detail, this has been described already in our annual issue "AUTOMOBILE ENGINEERING for 1911," and it is shown in greater detail here. The valves themselves are flat faced, and 40 mm. in diameter inside the faces. Cast iron is used for the cages, and these are held down by phosphor bronze caps which enclose the springs entirely, and, as they can be removed easily, complete detachment of a valve is therefore comparatively easy. A special spanner is used to unscrew the cap and, with a clean engine, the valve can then be withdrawn without trouble. After considerable use the long conical seatings are of course a little apt to stick, but even then they can always be removed without much trouble. The shallow pockets would prevent the valve head from falling into the cylinder were it to become broken off, because a valve is not likely to fracture in the immediate vicinity of the head, owing to the very large radius between the stem and the former. It might be thought, however, that the pockets are small enough somewhat to throttle the gas flow, and this may possibly actually be the case since the engine is not supposed to run at extremely high speeds. At the same time, if there is not much depth, there is ample breadth of pocket, and the flatness of the valve also assists to increase the opening area, while rapid action cams are used.

The camshaft lies in bearings carried directly in the main casting, and the valve tappet arms rock on shafts which terminate in bridge pieces fitting over the same studs as the bearing caps, although they are above and separate from the latter. To get out a valve, therefore, it is necessary to undo four nuts, when two bridges and the tappets for four valves can be pulled up, the whole operation taking an extremely short space of time as the fitting is very accurate. Once the tappet set for a pair of cylinders is taken off its holding down studs, it can be dissembled by merely pulling apart the end pieces, as the shafts on which the tappets rock merely fit in holes in the bridges and are not locked. Further to ensure silence the tappets are adjustable in the tappet arms, the latter being split so that the cap can be locked by a clamping screw, and inside each cap is a spring plunger which bears against the valve stem.

At the rear end of the camshaft, behind the worm gear which drives the oil pump, there is an eccentric driving a small plunger pump to supply air pressure to the petrol tank, and it is perhaps worth remarking that this air pump is not made in the main casting but is attached by the flange seen in the sectional view of the engine.

No adjustment is provided for the chains which are of considerable size, the width being 22 mm., while the pitch line diameter of the smallest wheel is 65 mm. This absence of adjustment means necessarily that the chains will have to be renewed more frequently than would otherwise be the case, because it is scarcely conceivable that an engine would continue to run until a half link could be inserted. However this may be, the durability should be quite reasonable on account of the large bearing surface, and time alone can show whether the adjustment is really necessary.

Contrary to modern practice, while there is separate support for the lower half of the centre bearing of the crankshaft, the lower bushes of the two end bearings are carried in the bottom half of the crankcase, which also contains the oil pump and the oil leads to the three main bearings. In Fig. I. the arrangement of the lubricating system can be seen quite clearly, and also the joint in the oil pump driving shaft which enables the lower half of the crankcase to be detached. The pump itself can be taken out separately, and also the long filtering tube can be withdrawn from the front end of the case. Additional to the oil pump shown there is an external copper pipe which carries oil from the bottom of the front end of the main pressure oil channel and delivers to a trough just above the intermediate chain wheel shaft, whence it overflows to the chains and keeps these continually lubricated. Lubrication of the valve gear is by splash, and is performed entirely separately, oil being poured in round the camshaft to such a depth that it just touches the cams and is replenished at rare intervals. This means that the valve gear works under the best possible conditions, but it has the disadvantage that the oil must be run out before a valve can be removed, as otherwise it would all run down into a cylinder. For this purpose a special drain is provided, whereby the oil can be drawn off quickly when warm. The cover of the valve box is, of course, aluminium, as is the bottom of the crankcase, and the front of the chain case. Pistons are made of steel pressings machined practically all over, and they are fitted with the special type of ring illustrated as a regular feature. The gudgeon pins are quite above the average of size being 20 mm. in diameter with a bearing length of 45 mm: they are, of course, hollow, and the connecting rods are bushed with phosphor bronze. All other bearings, with the exception of the three ball races on the camshaft, are phosphor bronze bushes lined with white metal. A point which should not be missed as indicative of the careful way in which the bearings are fitted, is the ball thrust immediately in front of the fly-wheel, because this is in operation only during such time as the clutch is held out of engagement.

Possibly the intricacy of the main castings is the most interesting feature of the engine, notwithstanding its other peculiarities. In one piece are the cylinders, the engine arms, the crankcase upper half, the inlet and exhaust passages and the whole of the valve arrangements. That such a casting keeps cool with thermo syphon water circulation is owing doubtless to the truly enormous water spaces which, it should be noticed, are carried well round the valves, preventing the cracking of the seatings from which some other makes of overhead valves have suffered. Notwithstanding the elaborate nature of this casting, the writer was assured that the percentage of bad castings is so small as to be negligible, being no greater than with ordinary pair-cylinder castings of normal type.

As regards the engine fittings, the standard carburettor is a Zenith, the air being taken through a hole in the middle of the main casting (seen in the section) provided with inward projecting radiating fins, and the standard magneto is an Eisemann.

Having dealt with the engine, we have disposed of the most interesting part of this car. The internal expanding clutch which works with notable sweetness, is an old design with which most of our readers will probably be familiar. The method of assembling the universal joint is worth noticing, as is the lubricating arrangement to be seen in the same view as the clutch: the two pins which terminate in the central floating chamber are slotted and fixed by ordinary cycle crank cotters, so that when these are knocked out, the joint falls to pieces immediately. Thus the detachment of either clutch or gearbox is rendered a very easy job A sketch view of the clutch coupling is also shown in Fig II.

Passing on to the next illustration, Fig. IV. This shows the gearbox in plan sec-

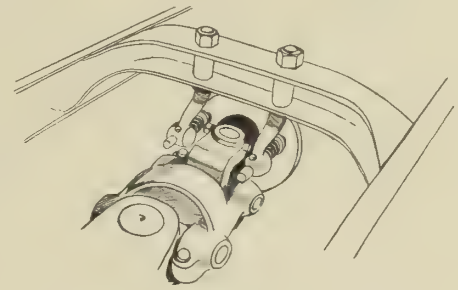


Fig. II. Germain clutch coupling.

tion and also a front end view. Most noticeable of course is the large diameter of the gears, and as this is commonly supposed to make for noisy running it should at once be said that the box is rather quiet than otherwise, a particularly striking feature being that on the second and third speeds the pitch of the note emitted by the gears does not rise very much with the engine speed. It is at all times a low note, and it does not seem possible to obtain a high pitched scream from either of the ratios we have mentioned. The fourth direct speed is quite quiet, there being no audible sound from the layshaft drive.

The fitting of the bearings by means of pegged collars is not usual, but there seems no reason why it should not be satisfactory, and the thrust bearing at the front end of the main shaft is likewise quite an uncommon design, which is presumably intended to relieve the main journals of any accidental axial load, while yet another unusual point, which cannot be commended too highly, is the provision of large thrower rings at each end of the main shaft, the box being intended to run with comparatively thin oil. These rings, together with the felt packing which comes outside them in each case, work with complete satisfaction, so that leakage of oil from the box is only possible when it is greatly overfilled. The gears themselves are made of a casehardened steel, and are not treated in any special way after hardening, their comparative ac-

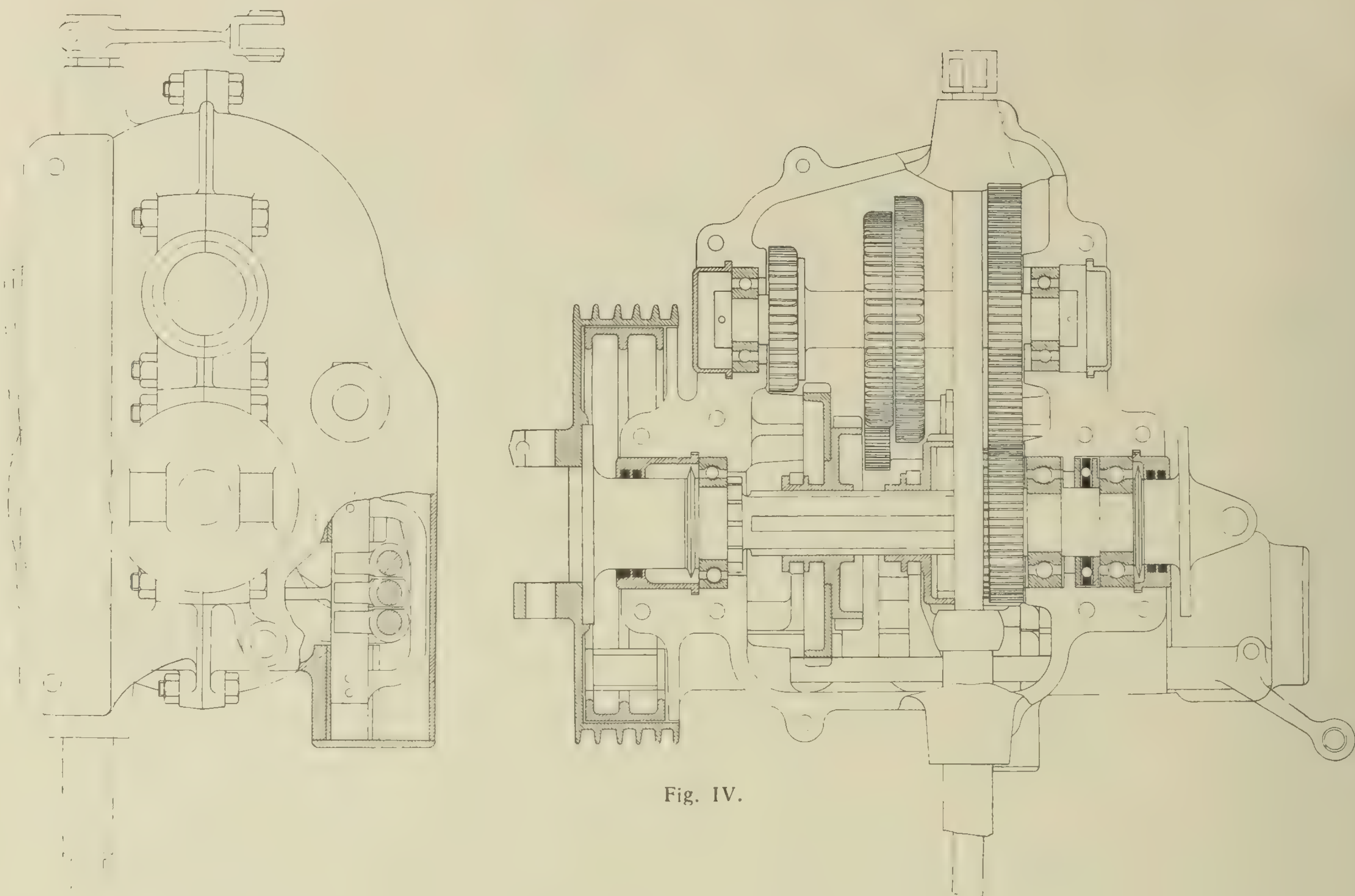


Fig. IV.

SECTIONAL VIEWS OF THE TRANSMISSION ON THE 15 H.P. GERMAIN.

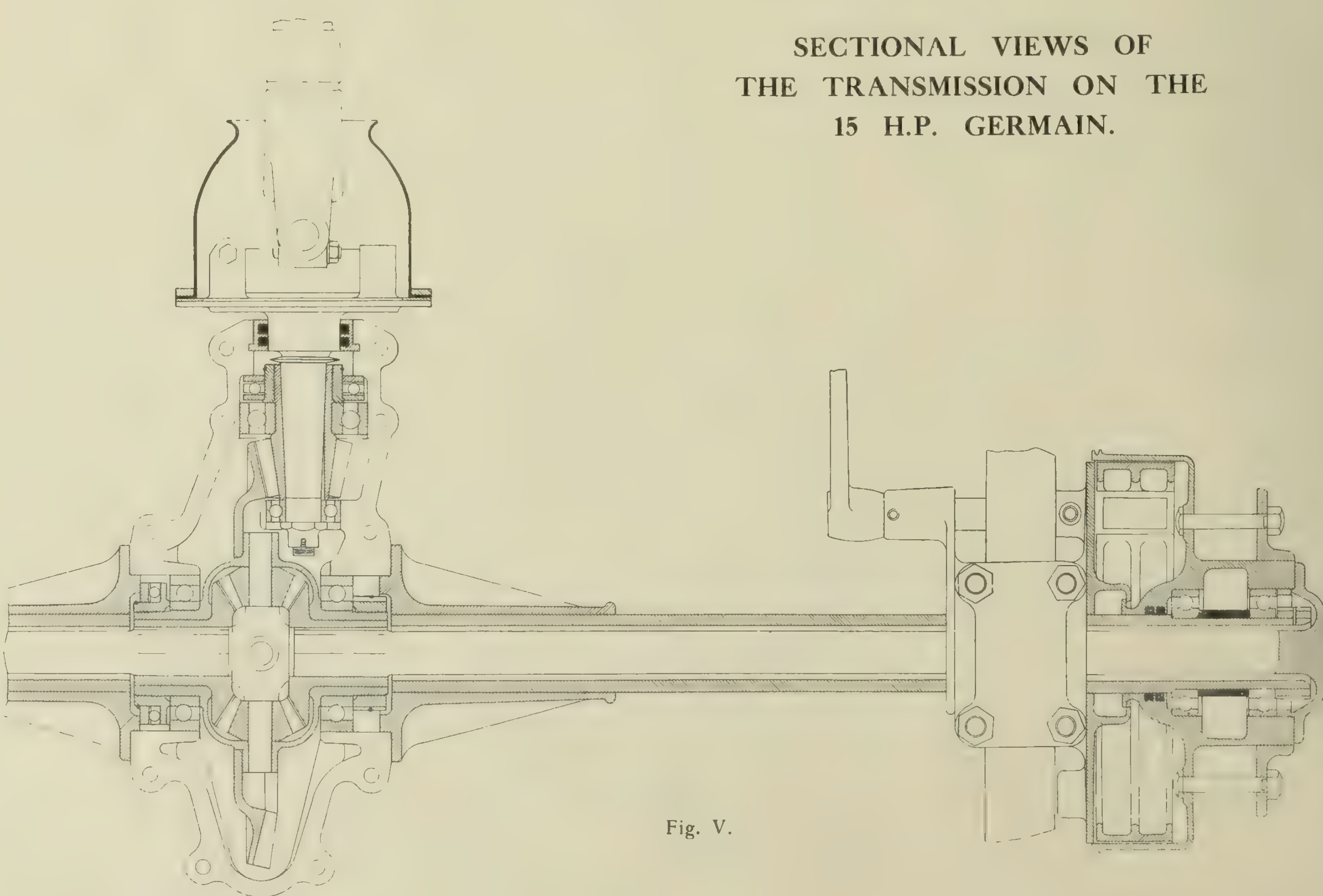


Fig. V.

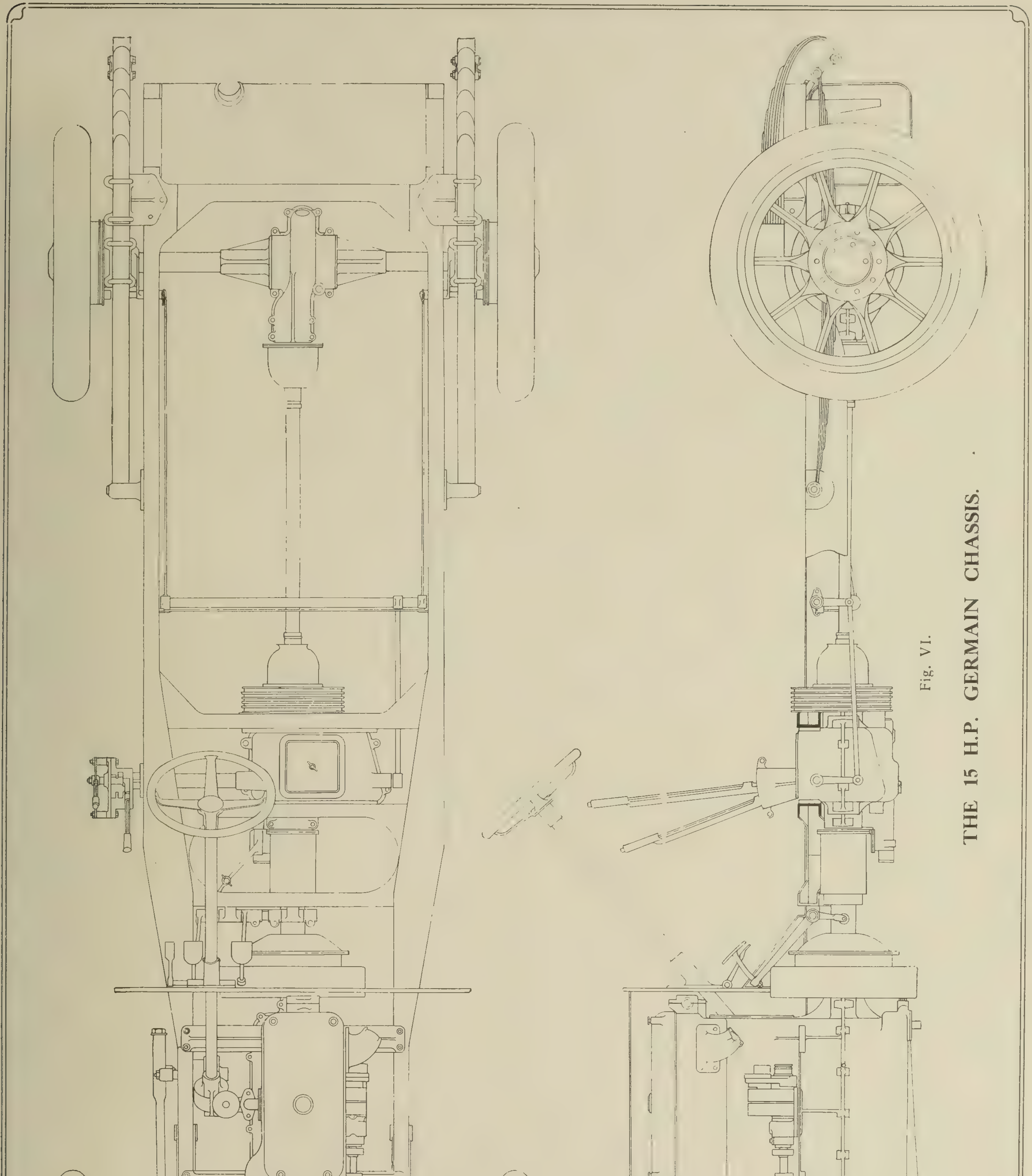


Fig. VI.

THE 15 H.P. GERMAIN CHASSIS.

curacy depending upon the procedure actually during hardening. For changing speed an ordinary sliding tube gate is used, but enclosed in an outer brass casing to exclude dirt and prevent possible damage, while the "sliding U" locking gear is employed, as can be seen in the sectioned part of the front end view of the gearbox. The shaft for the foot brake control passes right through the box and operates expanding cast iron shoes inside the cast steel drum. On the gearbox there is a disc corresponding in diameter to that of the brake drum, and enclosing the shoes completely; while this excludes all dirt it makes it necessary to take down the universal joint if it is desired to examine the inside of the drum or the surfaces of the shoes, though hand adjustment is, of course, provided in a position convenient to the pedal.

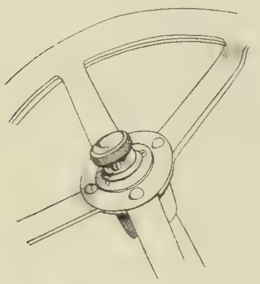


Fig. VII. The switch on the 15 h.p. Germain.

For the propeller shaft a rather unusual arrangement of two pin joints is used, the rear end of the shaft being squared so that it is free to slide in the fork. The joints themselves are precisely the same pattern as those between the clutch and gearbox, and are enclosed by pressed brass covers with leather wrapping to protect the joint. The diameter of the shaft is above the average, being 40 mm., and the joints themselves are fairly well proportioned, although they have plenty of work to do in the absence of any torque or radius rods. In Fig. V. a section of the axle is shown, and this again needs but little elucidation, it having a malleable cast iron central casing with steel sleeves, spring pads being fixed solidly to these sleeves. The driving shafts can be withdrawn through the hubs, and there is an oil trap for any lubricant which may leak from inside the hub shells, so preventing accumulation of oil within the brakes, which themselves are of precisely the same pattern as the foot brake. On the bevel pinion shaft another thrower ring may be observed, and adjustment of the engagement of the pinion can be made through the peculiar type of thrust ring, while the crown wheel can similarly be set over to one side or the other.

In the introduction of this article it was remarked that the very bad roads near the works probably exercised some influence on the design of the chassis. This is specially noticeable with regard to the frame, which is of extra strong section, as well as of the somewhat unusual design seen in the chassis views shown in Fig. VI. In the plan view it can be seen how the gearbox is hung between two cross members, while the engine arms rest directly upon the main side pressings. This arrangement would seem to make for difficulty in lining up the engine and gearbox, and this possibly accounts for the large universal joints between these two parts. The steering gear is well arranged, practically every arm and connection being straight, but it otherwise calls for no particular comment. Similarly the springs are quite normal, although the three-quarter elliptics at the rear have to take both the torque and the drive. In Figs. VII. and VIII. two interesting and neat details are shown, the first being a switch incorporated with the steering wheel, and the second a catch used for holding up the undershield.

On the road the car runs smoothly, and the periodicity of the engine at no point appears bad. A certain amount of engine vibration is just perceptible at slow speeds, but it increases strikingly little as the speed rises and, even when the engine is running at its maximum on the lowest gear, the vibration is not really unpleasant, which, as it is rather unusual with a long stroke engine, is therefore particularly worth noting. The four speeds change easily, and the standard gearing is sufficiently low to enable the chassis to carry a heavy body, in fact it is rather for ordinary touring work with an open body. The springing is very much above the average on rough roads, being equally good when tested from either the front or the rear seats, and there are no small noises, the cable actuation of the rear brakes eliminating brake rod rattles, and the number of links and connections otherwise being small.

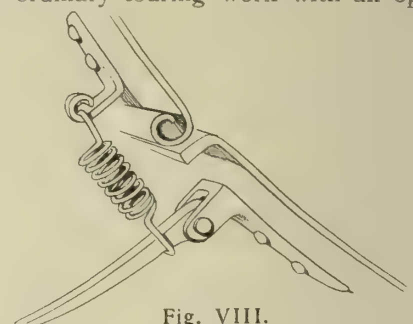


Fig. VIII.
Undershield catch on the 15 h.p. Germain

MACHINING OVERHEAD VALVE CYLINDERS.

By C. T. Schaeffer, M.S.A.E.

A SUBJECT which has had considerable discussion in all the leading technical journals of the world is the machining of automobile engine cylinders. Doubtless this is largely because the machining of these cylinders is always a difficult problem, regardless of what type the cylinders may be, and much study has been given to this subject by manufacturers with the object of increasing their quality, increasing the output and, as a result, decreasing the cost.

Engine cylinders are subjected to more severe usage than almost any other casting, and this adds to the care required in their design, casting and machining. The object of the present writer is to detail the methods and fixtures used in machining the overhead valve cylinders of an American car, the processes described having been in use for several years. Methods of machining cylinders vary, the machinery at hand usually deciding the method to be used, and therefore it would be advisable to state the operations in order as they occur before turning to the details of the fixtures. These are :—

(1) Facing the water jacket opening boss and cylinder base.

(2) Drilling the bolt holes in base.

(3) Boring the cylinders.

(4) Boring and tapping valve cage ports.

(5) Drilling and tapping water jacket opening boss, and cylinder top.

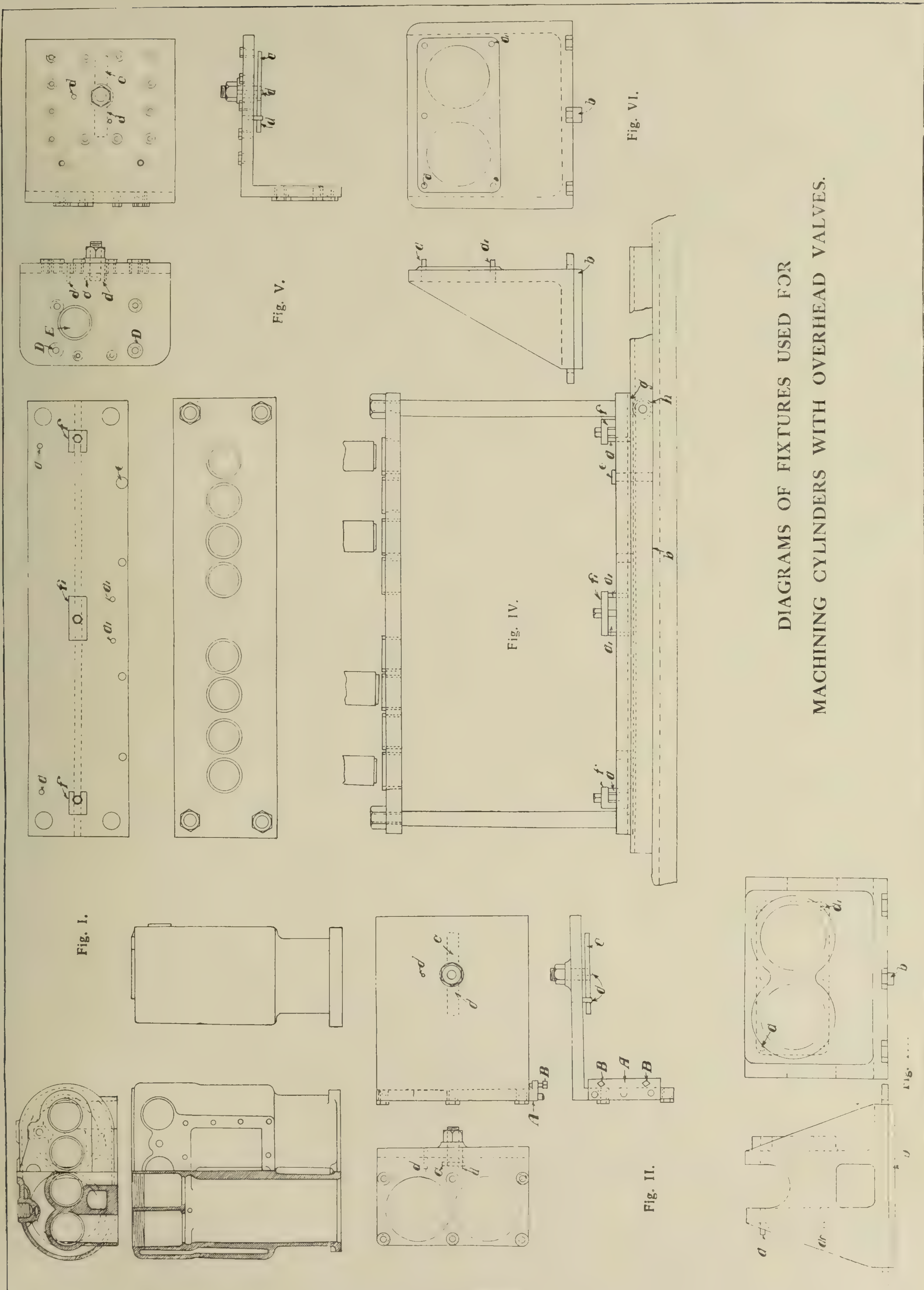
(6) Grinding.

There are, of course, various methods for locating cylinders, but in this instance the method of locating from the bolt holes in the base was used for the reasons given below, and it may be added that this is the most popular method, while it seems to give good results whenever it is used. It might also be well to describe the type of cylinder to be machined and the locating points, in order thoroughly to depict the method of procedure in machining.

Fig. I. therefore, shows the cylinders, which are cast in pairs and have all manifolds on the right hand side together with the water inlet pipe, the water outlet being on the top. The brackets for the rocker arms are located on the left hand side of the top and the valves are carried in cages. The large opening in the water jacket is used for several purposes, namely, as a core print in moulding, for cleaning out sand and for locating the inlet and exhaust manifolds, which rest on the upper face of the cover plate, as well as the water inlet pipe which is fastened to this cover plate. As the engine has six cylinders, this feature alone settles that the cylinders must be true in line upon the crank case in order to ensure good alignment for their pipework, and therefore a locating point must be found having a direct relation to this face.

The necessary point is fixed in the first and second operations by machining the water jacket opening boss and the cylinder base and locating the bolt holes in the base from the water jacket opening boss. Thus we have locating points for all subsequent operations, and are assured of true alignment with accurate location throughout.

The cylinders are first given a pickling to remove all sand, from both the jackets and the inside of the cylinders. After the pickling has been completed, the various core plug holes are drilled and tapped for pipe plugs, so that the castings may be tested for soundness. This is accomplished in the following manner: A plate is placed on the water jacket opening (this plate being connected to the compressed air supply in the factory) and clamped tight, using a light copper washer to obtain an air-tight joint: only two clamps are used to hold this plate. The casting is placed in a tank of water and the air in the jacket, which is circulated under pressure, will readily show any leaks in the jacket walls, port walls, and cylinder walls. This is regarded as a preliminary, and we may now proceed to the detailed consideration of the machining.



DIAGRAMS OF FIXTURES USED FOR MACHINING CYLINDERS WITH OVERHEAD VALVES.

First Machining Operation.

The first operation, as mentioned above, is that of machining the water jacket opening boss and the cylinder base, this being accomplished with cut jigs by merely clamping the casting on the machine table and using a gauge to obtain the proper distance of jacket face from cylinder bore; 1/64th in. is allowed on both surfaces for grinding, to insure a tight joint at these points. Both the manifolds and water jacket plate are ground, as well as the top of the crank case, the purpose being to avoid the use of jointing washers at any point, which is a feature worth striving for. The machining is done in a vertical milling machine with a large diameter of cutter permitting a very heavy cut to be taken for roughing, the grinding being done in a surface grinder. It may be remarked here that there is thorough inspection after each operation.

Second Operation.

The first operation being completed, the cylinders are passed on to the inspector, after which they are returned to the machine shop for the next process, which is that of establishing a permanent locating point for drilling the bolt holes in the cylinder base. This is accomplished with the fixture shown in Fig. II. which is a cast iron angle plate finished all over on the inside and using a clamp C to hold it in position. The small pins *d* act as positioning stops in clamping the cylinders into position. The fixture has two guides only at each end, with two set screws B to locate the holes endwise, and in true relation with the cylinder bore, to insure an equal wall thickness at all points, the centre of the bore being found with the use of a master dial, which is required only for the first setting of the jig. Afterwards it is only necessary to back off two set screws on one side, loosen the clamp and place the jig on the next pair of cylinders.

Third Operation.

The most vital point in the machining of cylinders is the boring. Fig. III. shows the jig used for this operation, using two pins *d* as locating points and holding the cylinders by bolts through the remaining clamping bolt holes. In this way the castings are held securely, and at the same time not distorted in any way. The jig is held to the machine table by lugs and has a tongue *b* to locate it relatively to the cutter bars and the work.

Probably the most interesting feature of this operation is that the two cylinders of one casting are bored together. This was accomplished by re-designing a Beaman and Smith horizontal boring machine, using a pair of gears to obtain a drive for both cutters, and also the proper centre distance which on this particular pair is $5\frac{1}{2}$ ins., the cylinder bore being $4\frac{1}{2}$ ins. This is a fast operation in that a high cutting speed is used and a very stiff boring bar. In this way the cylinders are machined rapidly, as it is said to be advantageous to rough machine at a high rate of speed, because the internal strains will be released as a direct result of the high speed work. The compression chamber head and walls are also machined as far as possible, in order to obtain equal compression in all cylinders. Small pockets at each side are used to prevent the valves from falling into the cylinders and so breaking the piston head on its upward stroke. Two cuts are taken, a roughing and a finishing cut, after which the cylinders are allowed to age for several weeks.

By referring to the jig it will also be seen that large bearings are provided at one end for the boring bars, every precaution being taken to bore accurately, so as to eliminate unnecessary grinding. After the ageing process has been completed the cylinders are given another test in order to make sure that no defects have been disclosed by the boring.

Fourth Operation.

For the fourth operation another interesting jig is used, and by referring to Fig. IV. it can readily be believed that this has been given a considerable amount of study by the tool designer. The jig has a base which carries a tongue *b* for locating it and lugs to hold it to the drill press table. On the base there is a pinion *h*, having a squared shaft to receive a handle and meshing with a rack *g* on the bottom plate of the fixture. Through this construction the travel of the jig in both directions is attained, making a simple and durable arrangement. This jig is made large enough to receive two castings, so that one may be finished while another is being rough bored, and as the finishing operation is faster than the roughing, it permits the removal of one casting and the replacement of another to take its place while the remaining one is in work. This makes a fast operation, and has the advantage of keeping the machine and operator working all the time. The rack *g*, as will be noticed, performs several functions in that it acts as a guide for the travel of the bottom plate and also as a locating point in building the fixture. It is made of steel, while the top and bottom plate and the base are of cast iron. The locating of the cylinders is obtained by the pins *d* for reasons as given above, and the cylinders are held by the clamps *f*, which are slotted to permit the easy removal of the cylinders. In this way it is only necessary to slack back the nut, leaving the clamp still in place on the jig. The jig is provided with one set of bushings for drilling and another set for tapping.

As only two cutters are used for both roughing and finishing, it is necessary to establish a locating point for each cylinder casting. This is given by the stud *e* which is laid out with reference to the centre line of each cylinder, the bottom plate having four holes drilled and reamed while the base has but one. The stud *e* is removed and placed into the next hole, and the jig moved endwise by the end of the pinion until this hole communicates with the hole *e*.

Fifth Operation.

The fifth operation is the drilling and tapping of the right hand side and top of the cylinders. The fixture for this operation is shown in Fig. V., employing a method of locating which is simple and fast. The clamping of the fixture is accomplished in the same manner as in Fig. II. with a clamp C. The holes D are for drilling the rocker arm brackets and, as two of these rocker arms are carried on each bracket, they must be located with reference to the valve cage ports. This is readily accomplished by dropping a stud into the large hole E which communicates with one valve cage port. The drilling of the side is done in a multiple drilling machine specially designed for this purpose, while the holes in the top are drilled in the ordinary way. When the drilling is completed, the drill bushings are removed and tapping bushings inserted while, as a guard against poor threads, the cylinders are re-tapped by hand by an apprentice.

Sixth Operation.

The sixth and last operation is that of grinding, and the fixture for this operation is shown in Fig. VI. Pins *d* are again used for locating, while the cylinder is held by bolts through the balance of the clamping bolt holes. This fixture also has a tongue *b* for locating it on the machine table, and is held to the latter by lugs and tee bolts. Care is also taken not to distort the cylinder in any way, but no bearing is provided for the grinding wheel and bar, because the wheel cuts so freely that the small stresses acting upon the bar can be neglected.

The tools for inspection comprise a complete set of gauges, plugs and master templates, and the machine work in all operations must come up to these standards. The writer might add that only a very small proportion of work has to be scrapped, which means that the inspection is sufficiently close after each operation to prevent mistakes being carried through.

THE STRENGTH OF CONNECTING RODS.

By George S. Bower, B.Sc.

Owing to the combined effect of end thrust and inertia loading, the accurate determination of the maximum stress in the connecting rod of a high speed engine is a matter of some complexity, and any formula giving this stress directly is necessarily approximate.

The rod will be assumed to be of uniform cross section from end to end, and will be taken to be in the position at right angles to the crank, this being the one for which the inertia loading is greatest, whilst the end thrust is not much less than at the commencement of the stroke, owing to the inclination of the rod to the line of stroke.

In Fig. I., AB represents the centre line of the unstrained rod, A being the "little" end, and ACB shows the form taken by the centre line under the action of the external loading. Let

l = length of rod between centres.
I = moment of inertia of cross section of rod about an axis through its centre, and parallel to the crankshaft, i.e., about XX (Fig. II.).
m = mass of rod per unit length (near its mid point).
w = angular velocity of crank.
r = radius, or throw of crank.
L = $w^2 r$ = acceleration of 'big' end B.
P = end thrust on rod.
x = distance from A to any point X in rod.
y = deflection of rod at X from the initial line AB.
R_A and R_B = reactions at right angles to AB, called into play at A and B to balance the inertia forces on the rod.

Since, in its assumed position, the acceleration of A at right angles to the rod is zero, whilst that of B is L, it follows that the acceleration of X, in the same direction, will be : —

$$= \frac{x}{l} L$$

so that the inertia force acting on an element at X, dx long, will be

$$= m dx \frac{x}{l} L$$

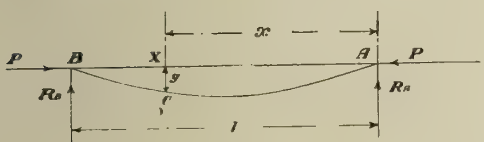


Fig. I.

From this it may easily be shown that

$$R_A = \frac{m L l}{6}$$

Reckoning bending moments as $+^e$ which tend to make the rod convex upwards, the bending moment at X due to inertia loading alone will be

$$= - R_A x + \frac{m L x^3}{6 l}$$

the second term expressing the sum of the moments of the inertia forces on AX about X.

Owing to its deflection, the resultant bending moment on the rod at X will be

$$M_X = - R_A x + \frac{m L x^3}{6 l} - Py \quad \ldots (1)$$

But we have the well-known relation :—

$$\frac{M}{E I} = \frac{d^2 y}{d x^2}$$

whence, substituting for Mx, we obtain :—

$$\frac{d^2 y}{d x^2} + \frac{P}{E I} y = \frac{m L}{6 E I l} x^3 - \frac{m L l}{6 E I} x \quad \ldots (2)$$

The complete solution of equation (2), the arbitrary constants being determined so that y is zero at A and at B, is :—

$$y = - \left(\frac{m L l}{6 P} + \frac{m L E I}{P^2 l}\right) x + \frac{m L}{6 P l} x^3 + \frac{m L E I}{P^2 \sin \sqrt{\frac{P}{E I}}\, l} \sin \sqrt{\frac{P}{E I}}\, x \quad \ldots\ldots\ldots (3)$$

Now, changing the sign of each side of equation (1), and substituting for y from equation (3), the former equation becomes, numerically :—

$$M_X = \frac{m L E I}{P} \left(\frac{\sin \sqrt{\frac{P}{E I}}\, x}{\sin \sqrt{\frac{P}{E I}}\, l} - \frac{x}{l} \right) \ldots (4)$$

On differentiating equation (4), and equating the result to zero, it is found that the maximum bending moment in the rod will be at the place whose distance x from the little end satisfies the equation :—

$$\cos \sqrt{\frac{P}{E I}}\, x = \frac{\sqrt{\frac{E I}{P}} \sin \sqrt{\frac{P}{E I}}\, l}{l} \quad (5)$$

and the value of the maximum bending moment may be arrived at by substituting the value of x found from equation (5) into equation (4).

By expanding $\sin \sqrt{\frac{P}{E I}}\, x$ and $\sin \sqrt{\frac{P}{E I}}\, l$, each to two terms, and then finding the maximum bending moment, as before, by differentiation, the following simpler, and more useful, approximate formula is obtained :—

$$M \text{ max.} = \frac{\cdot 385\, m L l^2}{6 - \frac{P}{E I} l^2} \quad \ldots\ldots\ldots\ldots\ldots\ldots (6)$$

Mmax being the greatest bending moment in the rod.

If d = depth of cross section,
and A = area of cross section,
the maximum compressive stress, q, in the rod may be determined by means of the equation :—

$$q = \frac{\cdot 385\, m L l^2}{6 - \frac{P}{E I} l^2} \cdot \frac{d}{2 I} + \frac{P}{A} \quad \ldots\ldots\ldots\ldots (7)$$

It would be impossible to design a connecting rod of the usual type rationally from equation (7), owing to the number of independent variables involved, and the easiest method will be to first of all assume a section, and, using equation (7), find the stress induced in it, increasing the section if this stress comes out too high.

The diameter of a rod of circular cross section may be obtained at once by substituting for I, m, and d, in terms of the diameter of the rod, into equation (7), and solving the resulting equation by trial.

Having determined a section for the rod which is strong enough to resist buckling about the axis XX (Fig. II.), this section must be examined as to its capability of resisting buckling about the axis YY.

The crippling load for failure in this direction may be found roughly from Rankine's formula :—

$$P_C = \frac{f_c A}{1 + \frac{1}{7{,}500} \frac{l^2 A}{I_{YY}}} \quad \ldots\ldots\ldots\ldots\ldots\ldots (8)$$

in which, for safety, the ends are assumed to be free to swivel, and where :—
P_C = crippling load of rod.
f_c = compressive stress at elastic limit.
A = area of cross section.
l = length of rod.
I_{YY} = moment of inertia of cross section about axis YY.

P_c should be about five or six times P.

It may be of interest to indicate the actual magnitude of the stress to which a connecting rod may be subjected, and so the stress will now be determined in a rod whose section is shown twice full size in Fig. II., its length being 10.62 in., the diameter of the piston 3.15 in., and the crank circle radius 2.56 in.

The thrust, P, along the rod may be taken as being equal to the force due to a pressure intensity of 300 lbs. per square inch on the piston area, and the speed of rotation may be taken as 2,000 R.P.M.

From the above data it is found that :—
I = .0424 inch 4 units.
I_{YY} = .00506 inch 4 units.
A = .3128 square inches.

$$\text{Hence } m = \frac{\cdot 3{,}128 \times \cdot 283}{12 \times 32 \cdot 2} = \cdot 000{,}229$$

P = 2,340 lbs.
L = 112,294 ins. per sec. per sec.

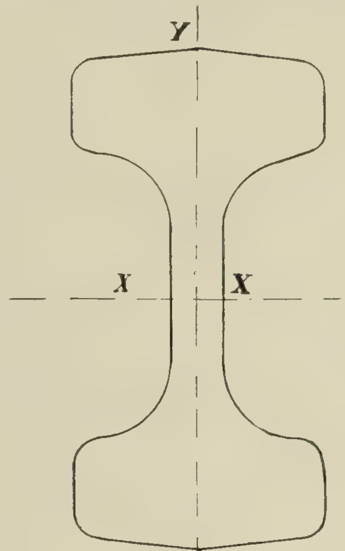


Fig. II.

Substituting the above values in equation (7), it is found that the bending stress is 2,416 lbs. per square inch, whilst the uniformly distributed compressive stress is 7,480 lbs. per square inch, so that the maximum compressive stress is 9,896 lbs. per square inch.

The crippling load for the above rod may be calculated, using equation (8), to be 7,630 lbs., so that an ample factor of safety has been allowed.

It must be remembered that the stress in a connecting rod is continually changing in amount as the engine rotates, so that it is liable to fail under a low stress.

Assuming the stress at the elastic limit to be 12 tons per square inch, and taking the same end thrust, P, to be acting as before (i.e., 2,340 lbs.), it may easily be calculated, using equation (7), that the engine speed at which fracture of the rod will occur will be about 4,000 R.P.M., a speed rather beyond the capacity even of the modern long-stroke engine.

It may be noted that the end thrust, P, has not usually a great effect on the bending stress, and, in many cases, may be safely omitted from the first term of equation (7), which thus becomes considerably simplified.

MACHINING PISTON RINGS.

A Description of a Cheap and Rapid Method.

By A. Davey.

OF all the parts of an automobile engine, no part has greater significance than the piston rings, for on them to a great extent depends the efficiency of the whole unit; for this reason too much care cannot be taken in their manufacture to ensure that the finished rings are as nearly perfect as possible. The method of manufacture here described represents the most modern practice, and the cost of the rings will be found to be extremely moderate.

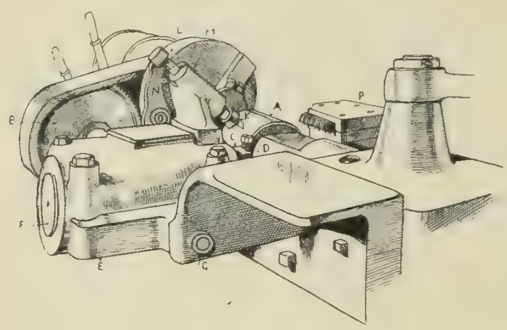


Fig. I.

The piston rings in general use by present-day manufacturers are what is known as the eccentric type. They are cast in the form of pots, each long enough for twelve rings. The first operation is performed on a 6A Potter and Johnson automatic lathe, equipped with the special piston ring attachment supplied by the makers of the machine and shown in Fig. I. The pot A is gripped in the standard chuck supplied with the machine, and is bored by a head carried on the bar, to which is secured the ring carrier D, which is a cast iron section screwed on to the boring bar, and, as the rings are cut off, they rest on the carrier, and remain there out of the way until removed by the operator. The eccentric turning is accomplished by means of the eccentric turning attachment mounted on one of the turret faces as shown. The body E has two bearings which are bored to suit the sliding shaft F to which is attached the tool-holder C. The eccentric motion is imparted to the tool-holder by the splined shaft G, which is geared to the spindle of the lathe. On this shaft is secured the eccentric which causes the sliding shaft F to travel in and out while the spindle makes one revolution.

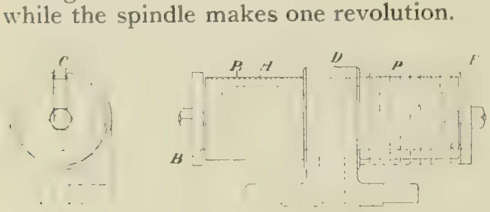


Fig. II.

While the boring and turning is going on the automatic marking attachment is marking the thin part of the ring where it is to be slotted. This attachment consists of the marking blade K, carried in the rod L, which slides in the bracket M. N is a lever pivoted at O, one end of which bears on a cam keyed to the shaft G, and the other end of which rests under the cap of the rod L; when the shaft G commences to turn the cam acting through the lever H pulls out the marker rod L until the high point is reached and then releases it. A spring housed in the bracket M forces the marker back to its first position with enough force to make a short line on the surface of the ring casting A. While this is taking place a gang parting tool P has been gradually feeding forward and cutting off the rings, so that, when the entire length of the pot has been machined, all the rings have been cut off. The only attention required from the operator during this operation is to remove the chucking piece and insert a fresh casting. Any diameter of ring from 4 in. to 12 in. can be machined with this attachment, the only changes necessary being the eccentric and probably the parting tool P. The time required to complete this operation on 12 rings $4\frac{1}{2}$ in. diameter is 15 minutes, including chucking, and this will compare favourably with other methods. The rings are next taken to a Pratt and Whitney surface grinder, where they are held on a Walker magnetic chuck and ground on both sides, to a limit of + or — .001. The best results have been obtained by grinding dry with an alundum wheel about 24 L. The operations take two minutes each.

The third operation is to mill a sufficient amount out of the circumference of the rings, so that after they have been ground on the periphery they will just spring inside the cylinder. For this operation the fixture shown in Fig. II. is employed, where A is a casting, cored out, as indicated by the dotted lines, for the sake of lightness, and turned on the portions P P to suit the bore of the rings. At each end is a "U" washer, which can be removed without removing the nuts. B is a dowell, which ensures the slot in the washer always being opposite the channel C, in which travels the slotting cutters. Ten rings are placed on each end of the jig, the "U" washers slipped in place, and the nuts screwed home. The cutters are fed in by hand at the end F, and the automatic feed put in.

After the cutters have travelled past the central part of the fixture D, against which the rings are clamped, the rings in the front part of the jig can be removed and a fresh batch inserted, and *vice-versa*, so that the operation is continuous. The time on this job is 120 per hour.

For the next operation of grinding the rings on the periphery, the fixtures shown in Fig. III. are used, which consist of a cast iron mandrel fitted with hardened steel centres, the body of which A is turned eccentric the same amount as the piston rings. It is of sufficient length to accommodate twenty rings, which are placed in position along the body A. The clamping plate B is then placed on the shoulder stud C, and the nut and washer D and E screwed on tight enough to prevent the rings and the plate B from working back. The split cast iron closing ring R is now placed in position as shown in the illustration, and the clamping bolts F, which have been slackened to allow the closing ring to pass over the tops of the rings, are screwed up tight. The nut D is then screwed home, forcing the plate B up against the side of the end piston ring which transmits the pressure to the rest of the rings, effectively preventing them from expanding when the clamping ring R is removed. This is now done, and the mandrel with the rings in position is placed between the centres of a Brown and Sharpe cylindrical grinder, and a driver attached to the end N. The outside diameter of the rings is then ground to the bore of the cylinder, the limit being + or — .002. The wheel used is a 24.J alundum, running at a peripheral speed of 6,000 feet per minute, while the piston rings are revolved at the rate of 40-45 ft. per minute, and lubricated by a copious supply of soda water. This operation takes 25 minutes for the 25 rings, including putting the rings on the mandrel, etc.

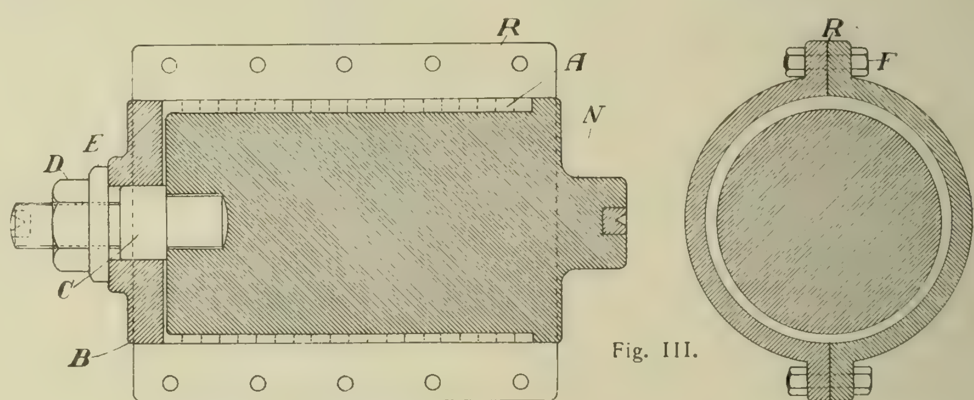


Fig. III.

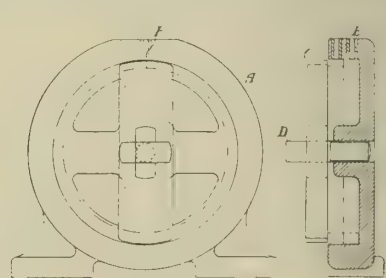


Fig. IV.

The last operation is to drill the hole for stop pin which prevents the piston ring from revolving when in place on the piston; for this the jig shown in Fig. IV.

is used. It consists of a casting A, standing on two feet, and is bored out the same diameter as that of the cylinder in which the rings will ultimately be used. Two rings are placed in the jig with the slots opposite the gauge mark F, and the slotted clamping plate C slipped over the "T"-headed bolt D, which is then given half a turn, securely clamping the rings in place.

The jig is then placed on the table of a drilling machine and the holes in the piston rings drilled through the hardened steel bushings E. The time taken on this operation is one minute each.

Summarising, the labour cost of the production of piston rings by the methods which have been described will be found extremely moderate. For example, taking the cost of 100 rings $4\frac{1}{2}$ in. diameter and 5/16 in. wide :—

	Hours	Mins.
1st operation, turn, bore, and part off	2	5
2nd operation, grind on both sides	3	20
3rd operation, slot	0	50
4th operation, grind on periphery	1	40
5th operation, drill for stop pin...	1	40
	9	35

Assuming that labour costs 8d. per hour then the labour cost for 100 rings = 6s. 8d., or for one ring about 4/5 of a penny. But the actual labour cost will be somewhat less, as the attendant's time for the first operation will be divided among several machines, and most of the other operations can be performed by comparatively unskilled labour. The times given allow for best workmanship and the setting up of the various machines, provided that not less than 200 are put through at one time, as, of course, the actual cost of the rings depends largely on the number put through at once.

THE GARDINER PATENT CRANK PIN TURNING MACHINE.

A crankshaft machine designed for greatly increased speed of output.

Now that the chief consideration of every machine shop superintendent is the rapidity with which his machines can turn out the jobs allotted to them, it is interesting to examine the particular machines which are evolved, in order to increase the speed of production, and to allow a greater degree of accuracy for each job by eliminating as far as possible the human element, and making the machines as near full automatic as the particular nature of the jobs that they are to handle will allow.

One of the longest jobs which faces the turning department is the machining of the ordinary four-cylinder crankshaft. The crank-pin machine here described has been introduced with a view to lessening the number of operations which take place before the solid forging becomes the finished article, and, at the same time, to reducing to a minimum the amount of time spent on each of these operations.

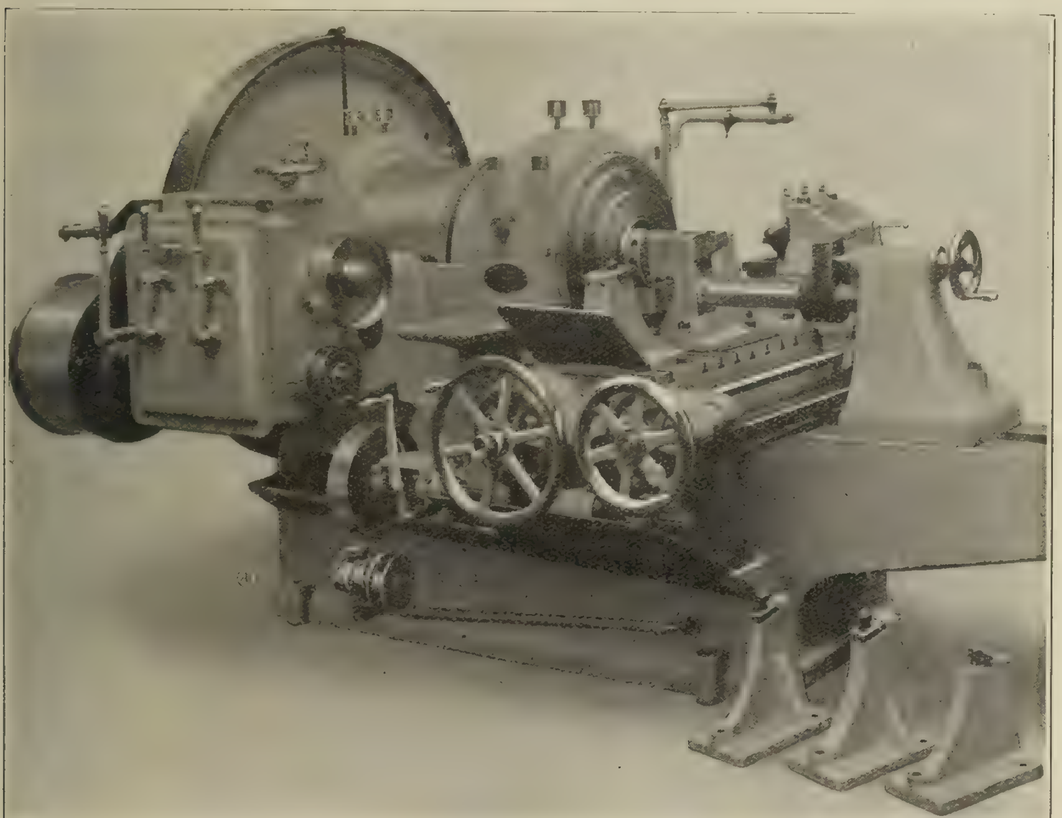


Fig. I.

Fig. I. shows more clearly the nature of the machine, and the crankshaft will be seen therein exactly as it is set up for the commencement of operations.

The headstock practically consists of a large diameter hollow drum, so designed that the entire crankshaft, with the exception of the particular throw which is being machined, can be held therein. This headstock is driven by a large helical gear wheel keyed to its outer end, and this gear wheel meshes with the pinion on the main spindle of the change speed gearbox, the primary drive being, of course, from the line shafting by means of the usual belt. In order that lubrication of the bearings may not be carelessly overlooked a pair of lubricators are fitted in a prominent position on the outer casing. Eight changes of speed can be obtained from the gear box by the usual sliding gears, while on each speed the gears are kept in mesh by locking the change lever, care being taken so to arrange the striking mechanism that it is impossible to mesh two different speeds at one and the same time.

There is a special form of chuck used in connection with this drum headstock, which has an adjustment in order to hold the crankshaft securely when it is set out of centre to the extent of the crank throw. A disc fastening is used inside the drum to steady the end of the crankshaft furthest from the crank-pin which is being machined, and additional rigidity is provided by the use of a tailstock which holds the opposite end of the crankshaft, and is adjustable, by means of the hand wheel shown, locking in the usual manner, whilst the amount of eccentricity necessary for crank-pin machining is obtained by a specially centred arm forming part of the holding-down clamp equipment.

The cross slide is provided with a couple of tool holders as seen near the crank throw in the illustration, while in each of these holders there is a cutter so designed that it will machine the whole of the crank-pin without lateral movement.

A centre bolt holds the cutters securely to the tool holder which is of massive, and consequently rigid dimensions. A slotted lid is provided for any altered position of the holders.

In order that the complete turning operation may be finished without the removal of any tools and without any form of adjustment, the second tool holder is provided with a cutter with which the finishing cut can be effected. The cutter seen on the near side of the illustration is used only for the rough cut—that on the further side being, of course, the finishing cutter. Beyond the finishing cutter can be seen a special tool holder, the cutter of which is secured by the usual three set screws. This is used to form the radius on the end of each of the crank webs. For this operation there is a special centre provided in the chuck giving the correct radius from which the webs must be machined, and provision has, of course, to be made within the cylindrical headstock to allow of the crank rotating on this particular centre.

The feed screws both for the finishing and roughing cutters are plainly seen in the front of illustration and are provided with the usual automatic feed, while the diameter to which the crank-pin has to be finished can be most easily set by means of a micrometer graduated disc employed in connection with the feed screw; thus it is claimed that there is scarcely any necessity for the use of gauges during the machining operations, a point giving evidence of the care used in designing the machine, and greatly enhancing the rapidity with which the job can be completed. Further, it is claimed that, once the chucking disc has been set, a number of crank pins can be machined to its setting without there being the smallest change in the finished size, a fact which is in itself an excellent testimonial to the rigidity and stiffness both of the two-holder and of the particular method used for securing the job, while the cutters themselves must stand up to their work and not require the constant grinding necessary with single point tools used for a similar operation.

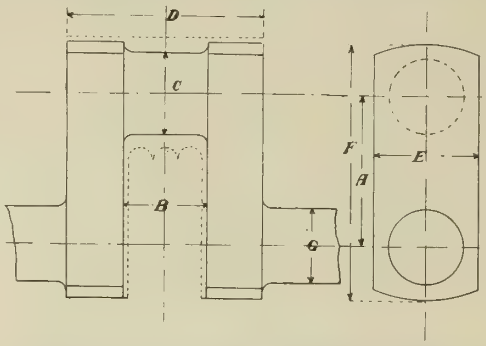
Lubricant is supplied to the cutters in a distinctly unusual manner. The headstock below its bearings is formed into a sump which contains the cutting lubricant; thence, by a pump, driven at a constant speed by a belt from a special shaft in the gearbox, the lubricant is forced to the operating cutter through the pipe seen in the illustration, and is then drained back to the pump by guides in the bed of the machine.

The makers claim that crank-pins can be finished within a maximum error limit of .0005 inches, which is certainly extremely good for this class of work. Undoubtedly the extremely massive construction of the whole machine is responsible for such accuracy of work, as there is not the slightest chance of spring in any part of the chuck or tool-holding mechanism, while the peculiar method which has been adopted for retaining the crankshaft renders such spring still less likely.

Fig. II. illustrates a typical job which can

most readily be dealt with in a machine tool of this type. It will be noticed there is a considerable amount of material to be removed from the rough forging before it is brought down to its finished size, and that part of this material has already been removed by the use of the twist drill and hack saw. It has been found considerably quicker to drill out the quantity shown than to allow the machine to make its way through the same mass of material, although it is perfectly capable of so doing. In Fig. 1 the forging which is in position on the chuck has been treated in this manner, and the amount of metal left normally for the cutter to operate upon is shown. The table shown beneath the drawing indicates the amount of time occupied in the various operations which go to make the finished crankshaft, and the figures, which certainly show quite remarkable speed of operation, are in themselves evidence of the machine's efficiency.

There is one point in connection with this method of machining which is not at first apparent, namely, that it is extremely likely to produce a sounder crankshaft than is possible by the use of the more common single point cutting tool, in that the steel of which the crankshaft is composed undergoes considerably less molecular stress than would be the case under ordinary circumstances. Although this may not be a very apparent defect, it is probably responsible for a great deal more wear and unsatisfactory behaviour in crankshafts than is gener-



A	B	C	D	E	F	G	Crowning time in minutes each	Necking time in minutes each	Total time
5½"	3"	2⅞"	7"	3¾"	9½"	2⅝"	8	25	33
3½"	3"	2½"	7"	3½"	7"	2½"	7	22	29
3½"	2¾"	2"	5½"	2¾"	6¾"	2"	5	16	21

Fig. II.

ally believed. The molecular stress in machine steel must be taken into account when the design of the crankshaft or any other part comes under consideration. That the work, to which the crankshaft is subjected, has a detrimental effect on the average steel is at once apparent to anyone familiar with the uses of the microscope in the examination of steel, and therefore any machine which may be evolved to lessen the amount of work put into a finished part is a point to the good.

The Gardiner Engine Co. are at present supplying this machine in only one type, provided with a barrel headstock of 15 in. internal diameter, which should be suitable for all but the most abnormally long stroke crankshafts. In conclusion, it may be mentioned that the times marked on the table in Fig. II. include both the time occupied in fixing the crankshaft truly in its chuck and that which is necessary to remove it finally from the machine when every operation has been successfully concluded.

The only item which is likely to interfere with the accuracy of the machine and its power of turning pins of a radius set by the feed screw disc, is wear on the edge of the cutter, since any trouble at this point, unless immediately allowed for, would alter the finished size from that shown on the disc. However, the design of cutter is such as to render this trouble as unlikely as possible.

CORRESPONDENCE.

No responsibility is accepted by the Editor for the opinions of correspondents, and the right is reserved to omit any portion of a letter if it is considered desirable to do so. Letters to be inserted with a nom de plume must be accompanied by the name and address of the writer. If publication in the next issue is desired letters should reach the Editor in no case later than the 20th of the preceding month.

ERECTING SHOP TROUBLES.

In a recent issue of *The Automobile Engineer* there appeared a letter on the subject of erecting shop troubles. It would appear that the writer of this letter either had experience of erecting methods of considerable antiquity, or was making a great deal of a very small thing.

In quite a number of erecting shops one is able to see methods which are all that could be desired, and in which there is scarcely any hand fitting. There is, to my knowledge, one firm at the very least whose engines are erected without an excessive quantity of fitting, and which are nearer to the drop-together principle than most.

In this case, with the exception of scraping engine bearings, valve grinding, and a certain amount of connecting rod setting, no further alterations are effected to any part, but each is placed in position, with the necessary amount of lubricant or boiled oil, and fits without trouble. It must have been a shop under the superintendence of a careless foreman, which allowed a fitter to adjust big ends with a hammer, and as such, can scarcely be taken as general practice.

As regards the absence of anyone capable of reporting such details as the writer mentions, it is my experience that most fitters will raise any amount of trouble over the smallest thing if they believe that it can be altered and their bonus times improved, while any really troublesome job is heard about throughout the entire works.

"Practics'" remark about the charge-hand of a gang having too much to do to efficiently overlook his men is obviously wide of the mark, since a charge-hand is not placed over a gang except for supervision, and he is generally the one whose attention is instantly called to any part which is giving unnecessary labour.

As to the difference between the average American fitting methods and those followed in this country, if anybody takes the trouble to thoroughly observe the smaller details on some of the American cars, and to compare the general class of work with that allowed here, he will find that it is considerably better to spend a certain amount on hand fitting than to rely entirely on the machine shop jigs.

Generally, it seems to me, that "Practics" is criticising the general fitting methods of this country from the standard he may have observed in one firm only, and that it would be better to carefully balance the whole before arriving at any decision concerning the general practice of this country. FITTER.

AEROPLANE FUSELAGES.

Sir,—In the June issue there was an article on the construction of aeroplane fuselages. The writer, in my opinion, does not lay enough weight on the design of the fuselage from the strength point of view. At the present moment there are some monoplane fuselages which are very inadequate in this respect; inadequate because they not only show decided signs of whip while travelling in the air, but also because they are almost invariably smashed if a rough landing is effected.

As an instance of the first, there was a machine at Brooklands whose tail whipped in a most visible manner, while as regards the second the manner in which the Antoinette tail seemed to fall off immediately the unfortunate Radley touched earth, although it was not directly affected by the impact, together with exactly the same breakage in much the same place when Latham came down on the sheds, seems to prove that there is a distinct weakness in the fuselage which has yet to be overcome. Again, when the Avro biplane had its smash some few weeks ago, several of the fuselage members had collapsed.

It would seem that the fuselage as a whole is designed for strength at the engine bed, but beyond this it is not as carefully considered as it should be.

Apart from the design, it is hard to believe that wood will be the future material for such construction; indeed, this might be said of the whole machine; it is to be hoped that something may be done with metal, either in sheet or tube form, as this is less liable to damage, while it can be made quite light enough. Damaged parts may be harder to repair, but on the other hand, they are less likely to occur, as a smash would be located round a smaller area.

One point about fuselage design which has not been touched on so far is the durability and the ability to withstand the action of the weather. In India and climates of a similar nature it would not be possible to use the same wooden construction for any great length of time, because the warping caused by the sun and the swelling due to the rains would speedily render every part of the machine inaccurate. This occurs even now fairly frequently under our own weather conditions and is largely responsible for the few machines seen about in the same condition as they were when originally issued by their respective firms. A. HOUGHTON.

WOOD IN AUTOMOBILE CONSTRUCTION.

Sir,—I read Mr. Maplethorpe's letter in your May number with interest, because at one time I felt very similarly to him about the use of wood in commercial car construction, but a more thorough study of the matter, and more modern developments, have now led me to think otherwise. I hope Mr. Maplethorpe will not misunderstand me. I do not claim to have made a more thorough study of the question than he, only my conclusions have been opposite to his.

I know, when I raise the objection of inflammability against wood, that Mr. Maplethorpe will point to the fact that wooden bodies are employed and considered good practice, but still I hold that wood for frame construction and body construction are two very different matters. Even for body construction wood is not an absolutely ideal material, and if some non-inflammable, unwarpable material could be found it would certainly have a fine future for body work. At present, however, wood is generally inflammable, and with hot exhaust pipes in close proximity and indirectly attached to it, I do not think it would prove a suitable material for chassis construcion. Then there are the joints, and nothing holds like hot rivetting, unless indeed it is a brazed tubular joint, but this class of joint must be arranged and made by men experienced in this branch of work, for the average engineer does not know as much as he ought to about brazing. Though I would like to suggest that the present pressed steel frame is largely a matter of fashion, the angle iron joints of a wooden frame can hardly be considered satisfactory, especially when they have to stand so much "working" as is inseparable from the long frame of a heavy road motor.

While one may fully agree with Mr. Maplethorpe that several woods may be as strong as steel in bending stress, weight for weight, it must be recollected that steel lends itself to being rolled, to give great stiffness for weight, in sections that would not be as effective in the case of wood, while the extra absorption of shock by the wood, would, I venture to say, be negligible compared with that which should, and can, be absorbed by the springs. A. R. HENDON.

A SUGGESTION FOR BODY CONSTRUCTION.

Sir,—Might I suggest that in your valuable journal you touch on a branch you have hitherto neglected; I mean the carriage building side of the business. At present the motor carriage building proposition is full of absorbing interest.

In this department there is one point about which I should like the opinion of others, and that is this. I feel that there is a considerable future for a cheap, light two-seated body, and am inclined to think that a stoutly-made frame covered over with canvas, similarly to canoe construction, might find a good many buyers. Most of those whom I have mentioned the matter to have opined that appearances would be against such a contraption. "Women," they say, "want something smart, and it is the woman who influences the buying." Obviously it is not this class of custom that I should be after with a frame and canvas body, but that there are plenty of others to whom nickel plate and fancy polished panels are quite secondary to utility, is proved to me by the number of cars—and not trade cars all of them by any means—going about in works grey. A body of the sort I have outlined would, I fancy, appeal to quite a distinct class of men, made up of those who want lightness, those who want cheapness, and those who want durability; for such a body with no highly finished panels to crack wherever a joint occurs, ought to look as good as new after a year or two of service, provided the canvas was not perforated or torn, and there is no reason why it should be. OCCIDENS.

THE ENGINEERING STANDARDS COMMITTEE.

The Engineering Standards Committee, which has been engaged in an enquiry concerning the number of sizes of bolts and nuts used by the motor car trade, have just issued their general report containing the Committee's recommendations and the proposed alterations in standard. It will be remembered that the *Automobile Engineer* has repeatedly advocated the adoption of some standard in bolts and nuts, since present practice entails great annoyance, in that it is not always possible to obtain a bolt or nut to replace one which may have been dropped on the road or damaged during the progress of some small repair.

Some firms have made a practice of adopting some strange thread in order that the car may less readily find its way to the small repairer; that this is a short-sighted policy is certain, since any annoyance or delay which may arise from such an item recoils invariably on the maker.

A sub-committee was entrusted with the drafting of the recommendations, and care was taken to render this committee a representative one by including members of the leading technical institutions and of any societies associated with the automobile trade. A full list of the whole committee will be found on the third page of the report together with the various bodies whose representatives they were.

The Committee have decided to recommend the use of the British Standard fine (B.S.F.) threads, but are of opinion that the same would be more suitable should the pitch of the $\frac{1}{4}$ inch size be altered from 25 to 26 threads per inch and a new standard size of 9/32 inch introduced, in which there were 26 threads to the inch. While recommending these threads for general use, the Committee fully realise that there are at present a great many parts in which a special thread must be used, and in this case have decided that such threads shall consist of a uniform pitch Whitworth of 16 threads per inch, irrespective of the diameter of the article in question.

Further, a wise step has been taken in making an effort to reduce the number of spanners which it is at present necessary to include in the kit supplied with a car. To accomplish this certain sizes of bolts are recommended with the same widths across the flats, which, consequently, can be manipulated with the same size of fixed spanner. Apart from this advantage, such a system results in great saving, both in weight and space. Any step which may be taken to provide fewer spanners is, undoubtedly, of great value, since there are few adjustable spanners which are so constructed that play does not render them inoperative, while a considerable saving in weight might conceivably be effected in the kit.

The Report itself occupies six pages, and is of great interest. Not only are the various sizes of bolts and nuts therein set forth, but a series of definitions locating the actual standard measurements are given for the guidance and instruction of possible users. All abreviations are set forth in a clear manner, and those sizes which it is proposed to abolish are also included. In each list every dimension is tabulated on a maximum and minimum limit basis.

All motor car manufacturers would do well to obtain immediately a copy of the report and to observe in what way their own products may be adapted to the recommendations, as there is everything to gain, both for manufacturer and user, in the adoption of such standards.

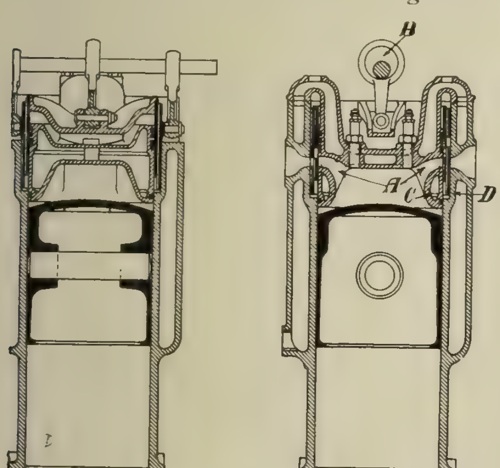
Published by Crosby, Lockwood & Son.

RECENT AUTOMOBILE PATENTS.

By Eric W. Walford, F.C.I.P.A.

A Double Sleeve Engine.

This engine is provided with two sleeves operating on the same principle as the Knight engine. That is to say, they have ports on both sides communicating with the inlet and exhaust, but it will be noticed that the sleeves are arranged at

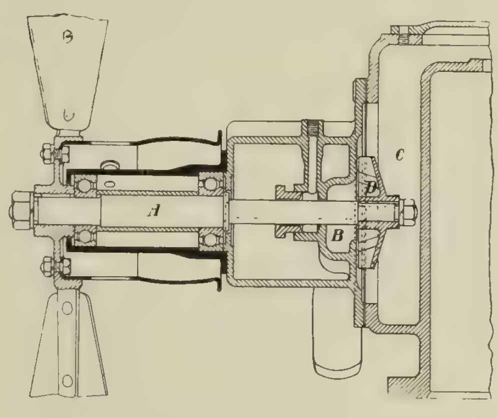


the upper ends of the cylinder only, the ports therein communicating with passages A. The sleeves are operated by an overhead shaft B, and they lie in a recess located between the cylinder head C and the main cylinder portion D, the recess not being in communication with the combustion chamber.

No. 12,610/10. Deutsche Automobil Constructions Gesellschaft M.B.H.

A Novel Pump Arrangement.

The vane spindle A is mounted in bearings in a sleeve which is bolted to the front cylinder, the sleeve being formed at B to form a water intake leading to the cylinder jacket C. The pump spindle is extended so as to project into the water jacket C, and at its inner end carries a



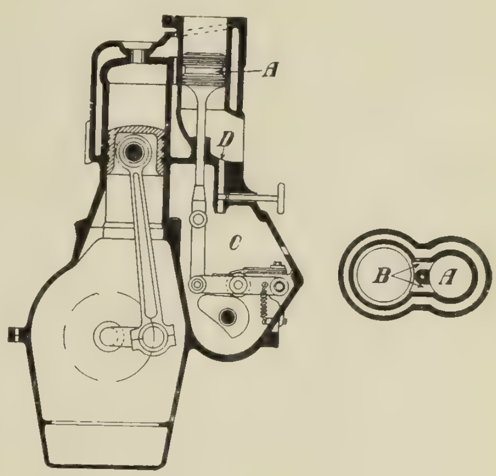
pump D, which takes its supply from the water inlet B and circulates the water through the cylinders. Further the spindle carries at the one end the pump and at the other end the fan, the pump consequently being located in the water jacket.

No. 20,277/10. Albion Motor Car Co., Ltd., and T. B. Murray.

A Piston Valve Engine.

A single piston A is used in a chamber communicating with the cylinder by passages B. The passages B are provided with a water-cooled bridge which prevents the rings on the piston A from being damaged, and the piston valve has a deep groove so that the pressure in the combustion chamber does not take effect on one side of it only. The inlet gas passes up past the bottom of the piston and the exhaust gas goes out at the top. For lubrication purposes some of the inlet gas is admitted to the chamber C, passing out by the outlet D. During its passage through the chamber C the gases take up some oily vapour and in this manner the valve piston A is lubricated. It will be noticed that the valve piston is actuated by a cam on the half-speed shaft.

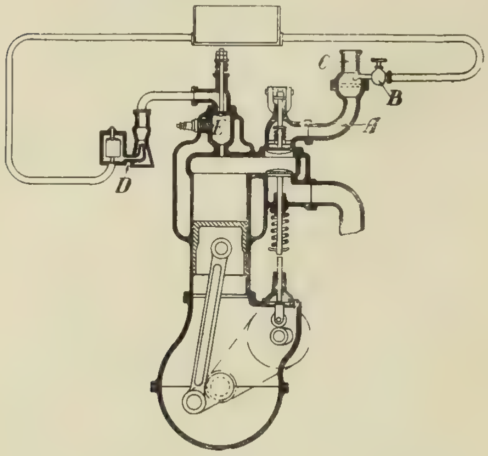
No. 22,408/10. S. Smith.



An Ingenious Ignition System.

This invention is intended to enable weak mixtures to be used without any fear of misfiring. The engine cylinder is supplied with the weak mixture by means of the induction pipe A, the fuel being admitted by a supply from the cock B, with the main air intake a constant opening. To vary the power, the cock B is opened or closed, with the result that the mixture is either enriched or weakened, the compression and volume of charge remaining the same, of course. It will be understood that difficulty would arise with regard to the ignition of weak mixtures, for which purpose a carburettor is arranged at D being adapted to provide a relatively

rich mixture to the chamber E, which communicates with the main cylinder. Thus on the suction stroke, in addition to

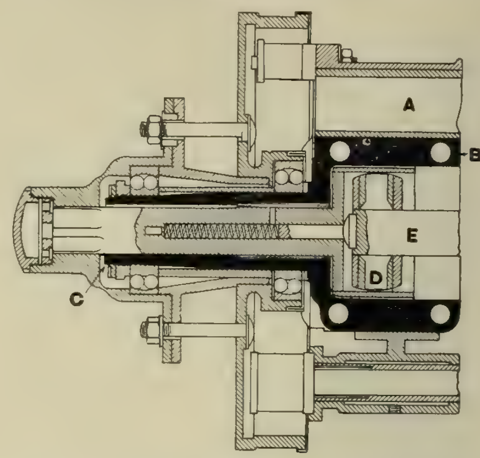


a weak charge being sucked in by the pipe A, the carburettor D operates and the chamber E is filled with a relatively rich mixture from the carburettor D. The sparking plug is located in this chamber and the rich mixture is consequently fired, this in turn igniting the weak mixture in the main cylinder.

No. 27,230/10. H. Pieper.

An Axle Arrangement.

This invention refers to the type of axle fitted to De Dion cars in which the wheels are connected to the differential gear by means of flexible shafts. The fixed axle bridge A carries at each end a tubular bracket B, the end of which is elongated to form a fixed wheel-supporting axle length C. The exterior of this axle length carries the wheel bearings, whilst passing through the interior is the short driving axle, which is connected by a universal joint at D to the flexible driving shaft E. The universal joint D is contained within the bracket B, which carries the rear springs. Hitherto this universal joint has been located outside the bracket, so that the shafts E have been correspondingly shorter and their



angle of inclination considerably greater.

No. 8,722/1911. De Dion Bouton, Ltd.

PROCEEDINGS OF THE SOCIETY OF AUTOMOBILE ENGINEERS.

The Society of Automobile Engineers have recently issued their volume of transactions during the year 1911. Beginning with a full list of all members of the Society, together with their commercial status, the volume goes on to deal with constitutional improvements which have been carried out, and all discussion which may be relative thereto. Then a complete report is made of all papers read before the Society, together with the usual illustrative diagrams.

The first paper concerns the manufacture of electro steels, and some considerations of the different forms of electric furnaces which have been in any way prominent, both of the type known as the electric arc, and those depending on induction and resistance for their results, beginning with a complete description of the early rotating Stassano furnace, and a short sketch of the career of such furnaces. Descriptions and criticisms are then applied to the Herault-Lindenburg arc furnace, and the Girod, mentioning the troubles peculiar to each type. Following on these come the Kjelin, Frick, and Röchling-Rodenhausen induction furnaces, and a comparison of the values of steels manufactured by the electric process, including some useful comparative tables. Finally the effect of the introduction of such a process to the automobile is considered, and what saving can be effected plainly set forth.

Immediately after this paper is the report of the iron and steel divisions of the Standards Committee, concerning which there is one rather interesting point, namely—that in America there is in some measure the same opposition to such standardisation which is constantly recurring in this country, in that manufacturers are afraid of it, holding the idea that it is an attempt to set out all cars in an exactly similar form.

A table is given where the desired compositions for each steel, whether of the carbon, nickel, or chrome nickel variety, are set out. Moreover, further added specifications are set up for valves and steel and iron castings. At the end of the tables there is an interesting series of notes giving the source of supply and particular process peculiar to each metal, together with its advised use, how it machines, whether it can be rolled or drawn ; in fact, all particulars of its use. The result of heat treatment is also tabulated, and particular forms of treatment recommended.

A further point of great interest is the paragraph dealing with material from which bolts and screws are machined. Here is laid down much that would be extremely useful to the designer in this country. Attention is called to the hap-hazard way in which such articles are, all too frequently, composed of any bar material which may be at hand, with the well known resultant breakage on the application of a spanner.

At the end of the paper a list sets forth all treatments allocated to the particular operations which are used during the building of a motor car, while a series of definitions and a short description of some testing machines employed closes an interesting paper.

Immediately after there follow the specifications recommended by the aluminium and copper alloys division, in which casting metals are fully dealt with, each in detail, from manganese bronze to aluminium alloys.

Ball bearing sizes then occupy several pages, a curious point being the adoption of comparative millimetre sizes when giving all details of the ball races. Sheet metal, broaches, and seamless steel tubing are also dealt with at some length.

A paper on multi-point ignition, containing several interesting diagrams of power curves with different timing variations, is next set forth, its conclusion following exactly on the lines of experiments conducted in this country concerning the value of a large spark as opposed to two or more separate sparks.

In the paper dealing with the construction of automobile roads, the problem was attacked from consideration of the road wear, and certain recommendations are set forth, among them not the least interesting is the unusual suggestion that the roads should be made cup shaped, instead of cambered, in section, to lessen the degree of disruptive effect exerted by an automobile when rounding a corner. Bituminous macadam is recommended for use in highway construction.

Immediately after this paper there is one on springs designed for automobile use, which has a degree of interest because it may be regarded as a guide for the designers whose springs must stand what are, probably, the worst civilised roads in the world. All the interest of the paper was actually centred in the discussion where, among other recommendations, there appeared one to add an additional leaf to the right hand front spring to deal with torque reaction and extra weight. Further discussion concerned the deletion of the centre hole, and a controversy as to the nature of the substance used between the leaves, that is to say, friction reducing or the reverse. A further noticeable statement is that springs having great flexibility on the front axle cause the car to become excessively unsteady to a dangerous extent. Illustrations appear of some useful slide rules manufactured for spring designers, and diagrams of total static load deflections.

The paper on hot rolled gears and that on valve systems have already appeared in the AUTOMOBILE ENGINEER. Further papers, mostly of use to American designers, are "The Franklin Motor Test," and "Crankshaft Grinding," together with that dealing with roller drives, garage fire risks, and automobile contest timing. With these later papers we shall hope to deal in future issues.

MISCELLANEOUS.

LATHES.—Thos. Ryder and Sons, of Bolton, have recently got out a catalogue of the machines which are manufactured by them. This includes types of screw-cutting lathes, full particulars of which are given, together with illustrations, and some examples of the cutting speed tables used for work of various diameters. A further catalogue deals with the forging and hot sawing machines manufactured by the same firm for the use of motor manufacturers.

CLUTCHES.—We have recently received the catalogue of the Saver Clutch Co., Ltd., of Manchester. Most of our readers will already know the clutch, which is a speciality of the above firm, and which has been used to a large extent for line shafting. The catalogue illustrates various forms of clutch suitable for machinery or automobiles, and gives a list of the requirements necessary to prevent errors when ordering the same.

AGENCY.—The Melbourne Motor Transit Company, of 115, Elizabeth Street, Melbourne, desire communications from any firms in this country who wish to be represented in Australia either for complete machines or for the accessories and spares necessary for the satisfactory running of those machines.

DYNAMOMETERS.—Heenan and Froude, of Manchester, have forwarded to us a booklet which deals with their turbine type water dynamometer in its new and altered state. It is claimed that the machine is more simple, more compact, and less costly than it was while retaining the accurate qualities which appertained to the older machine. In the booklet there are illustrations showing dynamometers suitable for motor car testing and full particulars as to the formula and methods employed.

CABLE.—W. T. Henley, of London Wall, have issued a new and complete list of the flexible electric cables suitable both for motor car wiring and for lighting. Practically all sizes and insulations are dealt with, and particulars are given of their respective resistances, prices and the extent to which the insulation is carried on each respective type.

CHANGE OF ADDRESS.—Vickers, Ltd., inform us that the address of their registered offices will in future be Vickers House, Broadway, Westminster, London, instead of 32, Victoria Street, S.W., and that in future all telephone enquiries should be on line 10110 Gerrard.

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CONTENTS.

ENGINEERING EDUCATION.

During the past year or two there has been much discussion concerning the proper training for an engineer, this doubtless being the effect of the changes which have taken place in educational methods during the last couple of decades. The matter of the education most likely to prove valuable to a would-be automobile engineer has been discussed at some length in these columns, and various schemes of training were suggested as being likely to lead to satisfactory results. However, it is confessed by practically all who have interested themselves in this wide subject, that it is impossible to point to any one system of training and say that it is the best for any particular branch of engineering. The case is perhaps made still more difficult for the parents of sons who wish to enter the automobile industry, because, with the proper education for the older branches of the science still a matter of dispute, it becomes nearly an impossibility to forecast the effect of any system with regard to the placing of a youth on the fair road to success in the youngest branch.

Most of those men who are now holding responsible positions in the motor car manufacturing business have come into it from other professions; few of them are young enough to have been trained entirely for automobile work, and thus the personal experience of the leading men of the trade is little or nothing to go by. It appears, therefore, that the Institution of Automobile Engineers are particularly bold in making a public offer of their willingness to assist parents in the determining of a course of instruction for boys who want to become members of the professional side of the industry. It is a step upon which the Institution are to be congratulated most heartily, and it ought to prove an immense asset both to that body, and to the trade as a whole eventually, because working this advisory system can only result in the gathering of a tremendous mass of information concerning the training and subsequent success of the students who are advised. Most of the graduates of the Institution, for example, have been destined for the automobile trade since they left school and commenced their collegiate training and, if it were possible to obtain a brief account of the education and commercial history of each graduate, it would certainly be easy to point to certain systems of training as being good.

In this country greater faith is still placed in works training than in theoretical instruction. In America, many quite successful designers have had a college training only, and no regular works training, as it is understood here. Both systems have obvious disadvantages, and no doubt the best all-round man would be produced by a combination yet to be found. Exactly where to draw the dividing line is the task which the Institution may be said to have set themselves.

We are glad to be able thus to record the activity of the Institution in taking up such difficult work, because in our last issue we felt obliged to criticise certain features of the Institution's procedure. We have received ample evidence that the opinions published last month are shared by very many of the membership of the Institution, but one or two items appear to have been misunderstood in some quarters, and we therefore take the opportunity of elaborating two of the chief criticisms. In the first instance, it was remarked that the Council might with advantage be re-organised, so as to be more representative of the industry. Of course, very many members of the Council have both a financial and professional interest in the manufacture of automobiles, but it would be easy to point out quite a large number of men whose status in the industry is entirely out of proportion to their power in the Council, while there are plenty of men amongst the rank and file of the membership whose knowledge undoubtedly entitles them to a place on the governing body. We believe that our criticisms have been taken in good part by the bulk of the membership of the Institution—that it is realised they were not made without good grounds and most careful consideration. We are glad that this should be so, not only because it is a vindication of our own action, but because when either a body of men or an individual reaches such a state that all criticism offends, then they are in a parlous state indeed. The Institution of Automobile Engineers has done much valuable work, and the motor industry is undeniably the better for the existence of the Institution. Representing as it does the newest and most active branch of engineering work, we believe the I.A.E. will take its proper place in the scheme of things when it becomes the most vigorous of the engineering societies. Let it not be forgotten that, whereas many of the older bodies have a vast membership, they cover also a vast field. That, whereas the members of the I.A.E. are practically all interested in the same subjects, the same detail matters, and the same commercial matters, the members of the Institution of Civil, Mechanical, or Electrical Engineers, are divisible into sections and sub-sections almost without end. At least seventy-five per cent. of the members ought to have been interested in each of the papers read during the last session, while any one paper read before the Mechanical Engineers probably interests vitally no more than twenty, or even ten per cent. of the membership. Automobile engineering is a very small thing now compared to what it will be in a few years' time. Let us hope that the same may be said with regard to the power, influence, and usefulness of the Institution of Automobile Engineers.

AMERICAN TOURING CAR DESIGN.

A consideration of the difference between American and European practice, and of the causes therefor.*

IT is an accepted maxim that a clever man is always willing to learn what he can even from those less gifted than himself. Many times in history Englishmen have not scorned to learn from other nations and, with engineering, have been particularly successful in gathering ideas from all the world and doing the best work with them. In the automobile industry it is idle to deny the value of early French and German influence, and though we need no longer be copyists, the British car would not occupy the leading position it does to-day did our engineers choose to close their eyes to all development outside the confines of these islands. So far it has been customary to neglect American practice as being devoid of desirable originality, but if this has been so in the past it will not be in the future, and it is even doubtful whether much has not been lost to us by lack of knowledge of transatlantic doings.

Many of our manufacturers know to their cost that the exports of cars from English ports to British colonies are not what they ought to be, that, despite even preferential tariffs, American competition has almost closed some British markets to British automobiles, and it has been customary to account for this by saying that American commercial activity is greater than British. A close study of American cars in service in their own country leads to the belief that this is not all : in some respects the average American car would be far better for service in—say South Africa, than the average British chassis. In order to realise why this should be it is necessary first to appreciate the wide difference in the working conditions for cars in Europe and America, and it is doubtful whether the untravelled Englishman can ever really grasp the nature of the roads on which American cars do nine-tenths of their running. Of course, many photographs of the roads traversed in the Glidden tour (America's greatest annual reliability run) have been published, but few people realise that such are not specially chosen bad stretches. It is not too much to say that the "dirt" roads, which are the only kind found outside urban areas in America, are no smoother and no harder than an ordinary ploughed field. Over such surfaces cars are forced at high speeds with heavy loads, and yet even some of the lighter and cheaper models contrive to hold together for several years.

Some Specific Differences.

To a British observer, sitting beside a rough American road and watching a number of cars passing to and fro, their most noticeable characteristics would be the noisy engines and the extremely silent chassis. The American designer has much to learn in engine work, but he is master of the world in the elimination of small noises.

Notwithstanding the constant and violent bumping neither squeaks nor rattles develop to the extent they would in a European car : in fact the few foreign cars met with occasionally in America are usually decidedly more noisy than the native product of like age. It is not too easy to account for this quality, especially when it is remembered that almost all American chassis have contracting band brakes on the rear hubs, but absence of squeaks certainly is due largely to the greater use of metal in body construction and the practice of mounting bodies either upon strips of leather or upon rubber studs, so insulating the metal chassis and wooden body frames.

Dashboards, too, are usually built very strongly, and add considerably to the rigidity of the middle portion of the frame.

Also there is no doubt that American leaf spring makers have been either particularly lucky or especially painstaking, for it is but seldom that a poorly sprung car is encountered—that is, for the long undulations of American roads. Springs, as a rule, have a narrow leaf and a large amplitude, with a distinctly low periodicity, while the use of shock absorbers or dashpots is common. Rattle is avoided by the elimination of many small parts and the large clearance allowed between such parts as brake-operating rods and levers and other chassis members. It is this, partly, which gives an impression of "stragglyness" to a British eye. Thus for American (or British Colonial) service the compactness and neatness of European design has undeniable drawbacks, because reduction of clearances calls for a greater rigidity if small sounds are to be avoided, and rigidity of frame, and of what might be called frame attachments, can only result in violent straining on roads where it is but seldom that even two of the four wheels are on a level.

Speaking broadly, it might be said that the European chassis is usually rigid from the front end of the engine to the rear of the gearbox, a distance of about half the wheelbase. In American cars there is a strong tendency to unify the crankcase and gearbox and to suspend it at three points, of which two almost coincide with the position of the dashboard on the frame. This reduces the rigid length to a matter of a few inches only, and whipping in the frame does no damage as long as it is within the limits of elasticity of the frame members.

Much controversy has raged from time to time concerning the advisability of combining gearbox and back axle, and as this is bound up with the question of rigidity it is not out of place to digress a little here in order to give consideration to a particular case. The Packard Company have confined their attention to this design for several years, but it has probably never been employed exclusively by any other firm of manufacturers either in America or elsewhere. Disadvantages are the increase of unsprung weight and the rather elaborate connections necessary between the change speed lever and the gears. Advantages are simplified manufacture by the combination of axle and gearbox castings, insulation of the body from the gears and decrease in the length of the necessarily rigid portion of the frame. Also, while large unsprung weight is a fault where the defects in roads take the shape of small hollows or obstructions, over which a car wheel must pass with more or less shock, it is not certain that a heavy axle does not assist a car to hold the road where the roughness consists of larger undulations over which a wheel can roll without breaking contact. The Packard car certainly does hold its native roads very well indeed, and it is a car with an exceptional reputation for durability. On the other hand, it is not conspicuously quiet, except with a closed body, when the insulation of the gears is an obvious gain. It appears that the Packard Company have adhered to their design more because they find no drawbacks to it than because they find advantages in it, and it seems to have been the simplification of manufacture which first brought about the adoption of the arrangement. Another and different effect of the rough roads is found in the steering gears on the cheaper cars, for these are undoubtedly much stouter and more generously proportioned than those on other than the better class of chassis here. Curiously, the ball thrust bearings in the front axle pivots, which are almost universal in Europe, are practically unknown in America, and the same applies to the bearings in the worm box, these also always being of the plain variety.

Axle Design.

In front axle design the American car possesses scarcely any peculiarities, though perhaps the tubular form finds more supporters there than here.

To again quote from Packard practice, it is claimed that the tube is better calculated to resist shocks applied in a more or less horizontal direction by road inequalities. Although this argument may be sound enough in theory, the H or I beam axle seems satisfactory enough for ordinary use and is the most common type by far. Usually it is the axle ends that are forked and not the stub pieces, which is a little surprising, because the other form of axle has a small advantage both on the scores of cheapness and of strength.

The matter of the attachment of the springs to both front and rear axles has had perforce more thought than this important detail is wont to receive, ∩ clips of very stout section being employed almost universally, on which it is not altogether unusual to find a castle nut and split pin used. Transmission of the driving effort and resistance of the rear axle torque by the chassis springs is not uncommon, but a torque rod is more often employed, and quite often radius rods are used as well—far more frequently than is the case outside America. Opinion generally is entirely in favour of the torque rod, and when it is omitted cost has always been the reason.

*This article is the first of a series written by a member of the staff of THE AUTOMOBILE ENGINEER who has lately made a tour of the North-Eastern United States, visiting all the principal automobile factories of America, and attending the summer meeting of the Society of Automobile Engineers held at Dayton, Ohio, during June.

Mention of torque rods suggests another brief digression which may prove of service to taxicab designers. The makers of one of the best known large cars in America were recently troubled by the tendency for the perpetual presence of a driver to cause the springs on the right-hand side to acquire a permanent, though slight set, and this they now claim to have cured by the simple expedient of transferring the torque rod to the opposite side of the chassis, so loading the left-hand springs.

To a British designer the American rear axle is an object of curious appearance, but this is due principally to the contracting band brakes on the hubs. Remove these and their attendant conglomeration of take-off springs, stops and operating levers and a more usual-looking axle will be found. Pressed steel casings are gaining rapidly in favour, but are usually welded in a vertical instead of a horizontal plane—a matter entirely of choice or convenience—and sometimes the centre case is pressed steel rivetted to tubular sleeves. A very common form has a cast steel centre rivetted to tubular sleeves, and malleable cast iron is also used occasionally, though not nearly so often as in Europe. The most favoured type here, that with conical sleeves, flanged and bolted to a centre case, is seen now and then, but not by any means frequently. Likewise it is scarcely usual to find the road wheels mounted outside the sleeves, the more favoured method being that of placing a single bearing outside the sleeve and relying upon the driving shaft to secure the wheel in place. Certain axle manufacturers are, however, giving all their attention to the newer pattern, and it will probably gain support for this reason.

The fact that a very large proportion of all the axles used are manufactured completely by specialists also results in the employment of practically identical axles by a number of car makers, and there is no doubt that this has militated against changes in design. None the less, it must not be forgotten that the axles of an American car have to withstand extremely violent shocks, and, as a car with a sagging rear axle is as rare a sight in America as elsewhere, it may be presumed that the axles of the parts makers are, at any rate strong enough.

Internally roller bearings are very much favoured, in fact they are used for rear axles and road wheel hubs to a much greater extent than ball bearings. Adjustment of the housings of the bearings is always provided for with all the better class of cars, and on many of the cheaper types as well, for it is recognised everywhere that adjustment of *both* crown wheel *and* bevel pinion is necessary if silent working axles are to be obtained with the minimum of trouble. It is not proposed to enter into details of these devices in the present article, but it may be said that they are always substantial, and often ingenious.

The axle arrangement whereby the differential may be removed without disturbance of the road wheels is encountered now and then, but not frequently. Likewise the expanding hub brakes, which are fitted as well as the contracting bands, are seldom so neat as they have become in Europe. Perhaps this can be accounted for by the fact that cleanliness of outline does not appeal very strongly to an American engineer—as witness the American locomotive—and that American automobile designers have had more important things than appearance to consider. Working under more arduous conditions, they have also given far more attention to manufacture than to finish.

Brake Arrangement.

Having mentioned brakes, they may fittingly receive consideration next. There seems to be no well-known car in America with a transmission or gear shaft brake, the arrangement of external and internal shoes on the same rear hub drums being practically in universal employment. Seeing that the bad roads of the country provide an overpowering argument against the external hub brake, this radical departure from current practice elsewhere is at first not easy to understand. Enquiry, however, has elicited the suggestion that it is due to the action of one of America's leading makers. This concern some years ago adopted a design of transmission which made the fitting of a propeller shaft brake practically impossible. It was therefore necessary to find every argument in favour of the double hub brakes, and this an advertising department did to such good purpose that the rest of the American makers were weak enough to copy the design—in some cases against their own better judgment.

Apart from arrangement, the only other peculiarity of brakework is that a metallic packing lining is almost always used, not only for the external bands, but for facing the shoes of the expanding brakes. The material certainly makes for smooth braking action and freedom from rattle or clatter, while the opinion of several users, who have also tried plain metal shoes, is that the durability of the woven material is greater.

One noticeable effect of the use of this composition is that a squeaking brake is very rarely to be heard, whether the type be either expanding or contracting. Of course, expanding brakes always have shoes similar to those in use here except for the facing, but the contracting brake is still most commonly a spring steel band. This latter, of course, requires a multiplicity of stops and small take-off springs, the need for these contraptions still further emphasising the great advantage of placing the foot brake somewhere else. Front wheel brakes seem to be entirely unknown, for the writer did not encounter a single individual who had tried them, and very few who had even considered them. Naturally, the compensation of a double set of brakes on the rear hubs calls for considerable rod work, and it may be remarked that stranded wire cable is very seldom used for this purpose, the ordinary compensating link mechanism being by far the commonest type. As has already been remarked, the brake connections are so disposed that they are most unlikely to cause rattle.

Hand adjusting devices are very rare compared with European practice, but are to be found now and then, wherefore their popularity is likely to increase very rapidly in the near future, for nobody is quicker to adopt a good feature than the American designer—once he is convinced that the new feature is really good and not an unnecessary extravagance which will increase the cost of production.

Transmission and Gear Design.

Most of the very large cars of America are provided with four speeds, of which the highest ratio is a direct drive, but three speeds only is almost universal practice on the small and moderate sized chassis, while there are a very large number of quite small cars with only two speeds. Our opinions on this matter of gear ratios have been expressed so often in these columns that there is no need to make repetition of them here, but the absence of the fourth speed is more easy to excuse in America because high ratios cannot often be employed owing to the difficulties of the road.

Taken as a whole, the spur gears used in American gearboxes run comparatively quietly, though the difference between the average American and European car in this respect is not very great. Of course, the problem of quietening transmissions is not quite so difficult in America because the engines in use there mostly give their power at much lower speeds, and the speed of revolution is undoubtedly a most important factor in gear noise. Really perhaps, the most striking thing in connection with gears is the ability of the American maker to turn out an enormous number of cars in which no *one* is conspicuously less noisy or more noisy than the rest of its series. The writer believes that this is due principally to the use of much larger pitches than are common here. A wheel with even such small teeth as 8 pitch inch diametral, which is a common size in Europe, is almost unknown in America, while some of the big cars use pitches as large as 4 inch diametral, on the sliding gears; 5 and 6 being the commonest pitches. The effect of using such large teeth is twofold, firstly by reducing the number of contacts made and broken per minute it lowers the "note" of the gear and, secondly, any inaccuracy either of tooth form (by warping in hardening) or in shaft spacing, is likely to be a very small matter by comparison with the size of the tooth, that is to say an error in position or form of a thousandth of an inch is only half as important on a wheel with 4 pitch teeth as it is on a wheel with 8 pitch teeth. The tooth forms used are much the same as ours, though a good many engineers are inclined favourably towards the long addendum tooth form, especially where the choice lies between the latter or a stub tooth.

The material employed for both sliding gears and axle bevels is almost invariably a mild casehardening steel, and it is not usual to find that there is any treatment after hardening except with respect to crown bevel wheels. For the latter a process which produces extremely good results is in use by many of the principal manufacturers, the idea being to harden only the faces of the tooth, leaving the whole of the rest of the blank quite soft so that it may easily be trued up afterwards. To attain this end the blank is turned and the teeth are rough cut. The half-made wheel is then electro-plated with copper, a fairly thick deposit being put on, and after this the final cut is made, trimming the teeth to their finished size and, of course, removing the plating in the process. This produces a wheel to finished dimensions, but with every part, except the

tooth faces, protected against the action of the carbon in the hardening furnace, for, although during the hardening most of the copper burns off, it is found that practically no carbonising takes place throughout the body of the wheel.

A variety of ingenious presses are used for returning warped wheels to their correct shape, most of them depending to some extent upon the deftness of the operator. In one of the largest works the straightening apparatus consists of a bed-plate in which is cut a circular groove corresponding in diameter to the pitch diameter of the wheel. In this groove slide five or six blocks with flat faces, and the crown wheel lies loosely on these. All the blocks are free to move, and the workman, having first tested the wheel on a special surface plate, arranges the blocks so as to bear on the high spots on the back of the wheel. Pressure can then be applied to the centre of the blank by an ordinary screw and flyweight, and after each "squeeze" the blocks are moved to different positions until sufficient accuracy is obtained to enable the wheel to be passed as finished. After this, adjustment is almost always made when the bevels are assembled in the axle case, this sometimes being done during the road test of the chassis, but more often during a bench test in the shops.

The Packhard Company have a most elaborate equipment for testing their combined back axles and gearboxes, it consisting of a number of small rooms built with thick brick walls and provided with double doors. There is a row of these chambers, and in each is a frame on which the assembled axle is bolted. Outside each chamber there is a 30 h.p. petrol engine, and a shaft runs through a stuffing box in the wall. This is coupled up to the transmission, and brake drums are placed on the ends of the driving shafts. When in the room with the door closed, it is impossible to hear any sound whatever from outside, so the noise made by the gears can be judged easily. Adjustments of the bevels is made very quickly, and a certain amount of interchanging is done with the spur gears. After this treatment it is found that there is never any necessity to interfere with the transmission during the road test and, although the gears are not exceptionally quiet, there is no doubt that they are *all* alike, while the bevels are very much above the average of European work.

In machine shops, scarcely so great a variety of gear cutting tools are used as here, especially with regard to bevel manufacture, the Gleason machine being almost universal for this branch, although the Bilgram is slowly gaining fresh adherents, and a few other types are found now and then. For spur gear work Brown and Sharp machines are found in the greatest numbers, but the Fellows is very much favoured, the Pierce Arrow Company using practically nothing else, and having something like thirty Fellows machines in daily use.

(To be continued).

THE ORGANIZATION AND EQUIPMENT OF A SMALL AUTOMOBILE FACTORY.

By Archibald T. H. Davey.

It will be easily understood that in laying out an up-to-date engineering works, great care and judgment has to be exercised in arranging the various departments, so that a minimum amount of time is spent in transferring the work from one department to another. For this reason it is usual to have all sections on the ground floor. Another important point to arrange is that as far as possible the work travels from one side of the works to the other, entering as raw material and leaving the far side as the finished product. A telephone system should be installed connecting the various departments of the works to the offices in order that a minimum amount of time shall be lost in giving orders, issuing drawings, etc.

The principal points to be considered when designing the shops are—

1. That they shall be strong enough to resist the heavy stresses brought about by rapidly revolving shafting and pulleys, heavy loads on floors, wind pressure, and the like.

2. Ample provisions shall be made for ventilation, heating and sanitary arrangements, so that the employees shall be working under as congenial conditions as possible, which tends to raise the individual output.

3. Numerous exits should be provided so that the employees may readily leave the buildings in case of fire.

4. The lighting should be arranged so that a sufficiency of illumination is provided, without the direct light from the sun or artificial sources shining in the eyes of the workmen, and so tending to distract their attention. With this object in view it is usual for the workshop windows to face the north so that a more constant light is obtained than if they were facing the sun, and therefore more affected by the various atmospheric changes. This is more marked in the case of foundry buildings where direct sunlight would play havoc with the delicate cores and moulds.

5. Facilities should be provided for handling the work so that a minimum amount of manual labour is required. On this account cranes and hoists are advisable wherever the heavier parts of the work will be handled. The various departments of the factory should be connected by truck rails as shown in the accompanying illustration. The gauge of these rails should be about twenty-four inches, and the trucks or trolleys which are propelled by hand should have ball bearings fitted to the axles, as it will be found that the increased ease of handling over the ordinary type will soon make up for the slight extra cost. In an automobile works, where the parts to be handled are comparatively light, cranes need only be provided in the foundry, heavy machine shops, chassis and erecting departments, and painting and despatching shops.

The actual design of the buildings will, of course, vary with individual ideas and requirements. In this article the case of a moderate-sized works, employing about five hundred or six hundred men, will be taken as an example.

The Power Plant.

In a works of this size the question of power supply and distribution is a matter of some importance. In practically all modern shops electric transmission is in vogue. The power producers should preferably be horizontal gas engines running on suction gas plants using anthracite coal, as this source of power is universally recognised to be the cheapest and most economical, the actual fuel cost being about .185d. per brake horse power hour. The plant in the present case should consist of two engines of 250 horse power each direct coupled to the electric generators. There being three gas producing plants, the spare one should be connected to both engines by means of a two-way cock to allow of cleaning out either of the producers without the necessity of "stopping down" the engines. The electrical power should be generated in the form of continuous current, as alternating is not to be recommended in a works of this class. The voltage of the supply should be fairly high in order to reduce the amount of copper in the leads. The Hopkinson three wire system of distribution is in general use in engineering works. With the two dynamos each giving 220, which is in accordance with usual practice, coupled in series, we shall have a voltage of 440 volts at the outer leads. The current for the motive power in the machine shops, etc., will then be taken from the outers and that for the lighting from one outer to the centre lead giving a 220 volts circuit. For the lighting of the shops flame arc lamps should be used for the general illumination, and metal filament lamps of moderate candle power for individual lighting. The candle power required for the various departments can be readily ascertained from the recent article on the "Illumination of Workshops," published in this journal.

The Pattern Shop.

This is isolated from the other buildings on account of fire risks. The equipment usually consists of small lathes, a large face-plate lathe, or a combination of the two, band saws, circular saws, planing machines, and the special pattern-making machines recently introduced. In view of the large amount of metal patterns now used, it is advisable to install a small metal-turning lathe, and perhaps a light drilling machine. These machines should be group-driven from a motor having a belt drive to the lineshaft. A good overload percentage should be allowed for the motor, as this class of work is apt to be deceptive as to the amount of power required, especially so in the case of circular saws. The line-

shafting here, as well as in all other shops in the works, should be fitted with roller bearings, as this type only absorb about half as much power as the ordinary type. Some authorities advocate ball bearings, but the writer has not found these satisfactory where heavy loads are frequent. A loft or cellar, preferably the former, should be provided, in which to store patterns which are not in use in the foundry, and in this an automatic fire alarm and water sprinkler is usually installed. A card index of the patterns is kept so that the first cost, material, subsequent alterations and the like may readily be ascertained. In the majority of establishments the pattern shop is under the joint control of the chief draughtsman and the foundry manager.

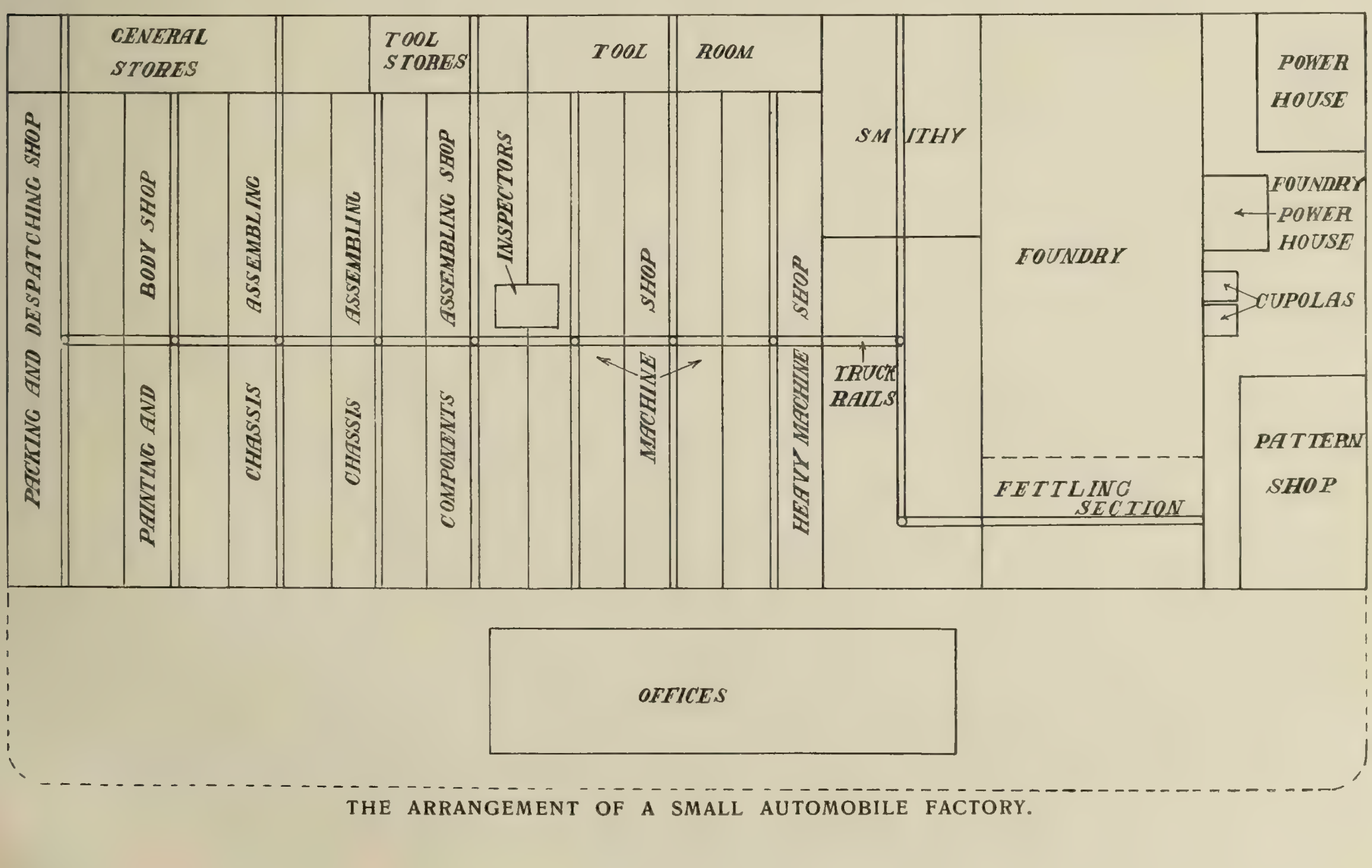
The Foundry.

This is one of the most important sections of the works, for good work in the machine shops will only be thrown away on faulty castings. An electric crane of about three tons capacity should traverse the main bay, and hand cranes capable of lifting from 15-20 cwts. should be provided in the smaller bay, the foundry building usually being in the form of a large central bay with a lean-to at one side. As before noted particular attention should be paid to the lighting, so that the direct sunlight shall not play on the floor. The cupolas employed for melting the metal should be situated as near to the centre of the building as is convenient, so as to minimise the risk of cooling the molten metal when transporting it from the cupola to the moulds. Roots' blowers should be used for the blast, these being preferable to fans, and they should be driven from a motor of ample power. A hydraulic lift should be provided for transferring the ore and scrap, etc., to the level of the cupola door, usually some fifteen to twenty feet above the ground level. The pump required by this installation should be motor driven from a lineshaft, which will also serve the air-compressor for the sand-blast plant. This motor, pump, and air-compressor should be contained in a substantial power-house, which may also contain the blowers for the cupolas. In the foundry, moulding machines are employed wherever the quantity of castings is sufficient to justify the cost of making the special patterns required by these machines. This class of machine is usually operated by hydraulic power, the same pump which operates the cupola hoist being used, the connection from the power-house to the moulding machines being made by suitable piping. A bench should be provided for the core-makers, whilst a machine for making standard diameters of round cores should also be installed. An oven for baking the cores preparatory to their being placed in the moulds will also be necessary. One end of the building is set apart for the "fettling" or cleaning off of the castings. This section should be equipped with "rumblers," which are revolving barrels in which small castings are placed, and by the rubbing action of one against the other, helps to remove the sand and scale. Pneumatic chipping hammers are used for removing the "runners," etc., from the larger castings, while the smaller are dressed by emery wheels, which may be driven from the same shafting which serves the rumblers. A sand-blasting equipment should be installed, as castings which have passed through this process may be more readily machined owing to the surfaces being free from sand and scale. The compressed air required by this plant is obtained from the compressor located in the foundry power-house as described above.

Small castings which have a lot of machining to be done on them should first be annealed in a furnace before going to the sand-blast, as with annealed castings the expensive formed cutters, reamers and the like, will last a great deal longer owing to the increased softness of the metal without a corresponding loss of strength. For this purpose a fairly large furnace should be installed, those burning coke with a natural draught being preferable, in which the castings are placed, packed in iron boxes with sand or lime, and kept at a red heat for several hours. The furnace with its contents is then allowed to cool down slowly, and the boxes containing the castings removed. The space between the foundry and the machine shops is utilised as a storage for castings and forgings. A travelling crane should span this yard, so that castings may be lifted on to the trolleys, running on the rails shown in the illustration, and quickly removed to any part of the works.

The Smithy.

In a works of the class we are at present considering, this need not be large owing to the extensive use of drop forgings, which are bought in from firms making a speciality of this work. The bulk of the work consists of tool dressing and the making of forgings for jigs and fixtures, and work of an experimental nature. Centrifugal fans should be used for the air blast to the hearths, as owing to the blast being constant, they are well adapted for small work. Steam has long been superseded by compressed air for working the power-hammers. The self-contained type are preferable, as these do not require a separate air compressor, but may be driven direct from the motor which serves the fan for the air blast. In this department will also be located the case-hardening furnaces and quenching baths for hardening the gears and other wearing parts of the engine and transmission. Special attention should be



THE ARRANGEMENT OF A SMALL AUTOMOBILE FACTORY.

given to the ventilation of this department, electric fans being installed and provision made for removing part of the roofing during the warmer period of the year. The flooring should be of brick on a thick layer of concrete, to prevent the damp from working upwards.

The Tool Room.

This is one of the most important departments of the works. The work carried on consists of making special tools, jigs and fixtures to the drawings prepared by the tool designers, who are usually accommodated in the main office building, and are generally under the direct control of the superintendent or works manager. The equipment should consist of small and medium sized centre lathes, milling machines, shapers, drilling machines, cylindrical and surface grinders and possibly a relieving lathe. The machines should be belt driven from a line of shafting by a separate motor so that this section may be worked overtime or at night without interfering with the rest of the works. This section is usually surrounded by wire netting, as is the tool stores adjoining, so that unauthorised persons may not enter without permission. Adjacent to the tool room is the tool stores in which are kept the tools, jigs, drawings and gauges used by the workmen in the assembling and machine shops. The tools are usually accounted for by the check system by which the whereabouts of any tool or jig may be readily ascertained. A workman requiring a tool of any description is required to first deposit a brass check (a supply of which is issued to each employee on the firm) bearing his works number. This check is placed on a hook underneath the space for the particular tool he requires; if any jig or tool has been taken to the tool room for repairs or alterations then a check engraved to suit is placed against it. Drawings are issued against an order signed by a department foreman, which is destroyed when the drawing is returned.

The Inspection Department.

This usually consists of benches enclosed by wire netting arranged with suitable entrances. The duty of the inspectors is to check every part that leaves the machine shops on its way to the stores or assembling shops to ensure that it conforms in all respects to the requirements indicated on the drawing, and also to examine castings and forgings for defects visible and otherwise, before they are operated on in the machine shops. The inspectors should be provided with a complete set of working drawings and gauges; in case of dispute the shop set of gauges and those of the inspectors are checked against a duplicate set stored under lock and key in the office of the works manager.

Machine Shops.

To enter fully into the equipment of a machine shop would require many volumes, so we must be content with a review of the chief features only. The first bay will contain all the heavy machines, such as cylinder boring machines, boring mills, vertical and horizontal, crankshaft lathes, planers, shapers, large drilling machines, and the heavier turret lathes, the various machines being belt-driven from line shafting running the length of the bay. In the next two bays will be situated the lighter machines, say the smaller lathes, shapers, milling machines, and the like, and the cylindrical and surface grinders will also find a place here. They should be as far as possible isolated from the rest of the machines on account of the emery dust which would play havoc with the bearing surfaces of machines not specially protected, and also on account of the impossibility of obtaining a good finish on the work when they are situated in the vicinity of large machines liable to "chatter." All line shafting should be geared to their respective motors by raw-hide pinions on the armature shaft engaging a cast iron wheel on the shafting, or else by means of a chain drive, the motors being fixed on suitable brackets attached to the stanchions or walls of the building, the starting boxes being placed directly underneath. The flooring in the machine shops, as well as in the tool room, erecting bays, etc., should consist of wood blocks resting on a layer of tar, under which is a bed of concrete from 18 in. to 24 in. thick.

The Assembling Bay.

The space required for this section is not large, as the extensive use of jigs, special tools, and the like tends to make the assembling of machined parts a simple and rapid operation. Light hand cranes should be installed capable of lifting from 30 cwts. to 2 tons. The work of assembling is usually divided between several gangs. One group of fitters will fit together nothing but change gear boxes, another gang back axles and differential gears, while yet another group confine their attentions to the actual engine, and so forth. The assembling of the engine will be facilitated if special stands are provided to which the engine crank-case is bolted, and the rest of the parts fitted to it. An exceedingly useful appliance in this section is a valve grinding machine; this is power-driven, and is used to "grind in" the inlet and exhaust valves used in the engine. The machine with which the writer is best acquainted is made by Alfred Herbert, Ltd., and has been used extensively, the saving of time over the old way of grinding being nearly 75 per cent.

Chassis Assembling Section.

It is convenient to utilise the whole of the next two bays for the assembling of the complete chassis. The frames usually arrive at the factory in a more or less finished condition, the only work to be done to them being the drilling of the holes for the engine, gearboxes, brackets, springs, and the like. These holes are jig drilled in the machine shops before being handed over to the erectors. Pneumatic riveters should be employed for securing the various fittings in place. The small parts required to complete the chassis will be handed out from the stores in boxes each containing the components for one car. As in other departments, cranes should be in use capable of lifting the completed chassis, and stands for holding the frames while the various parts are being secured to them. The finished chassis are now fitted with temporary bodies, and the cars given a thorough test on the road, in order to bring to light any weak points in their construction, those which fail to come up to the requirements being returned to the assembling department for reconstruction.

The Body and Painting Section may occupy the next bay, and be fitted with suitable ovens for baking the enamelled parts. The bodies, splash-guards, canopies, etc., are next fitted, after which the cars pass to the re-assembling and packing shop in the next bay, where the final assembling and adjusting takes place. The cars are then packed in suitable wrappings or crates for despatching to their future owners. If the works are situated conveniently to a railway it will be of advantage to arrange for a siding to run through to the packing shop, so that the cars may be loaded directly on the railway trucks.

The Store Room.

Here is stocked all parts which are usually bought out, such as studs, nuts and washers, as well as files, emery cloth, hammers and the like, and the finished parts from the machine shops, which the erectors do not require immediately. This department should be enclosed with wire netting and shelves, and bins, preferably of metal, should be provided for small parts. A space is usually reserved for parts which may be required by customers for renewals and replacements, such as piston rings, valve springs, etc.

It will be noticed that no provision has been made for the manufacture of sheet metal parts requiring the use of press tools, as these are usually purchased in a finished condition from firms making a speciality of this work. As indicated in the accompanying illustration, the offices for a plant of this class are usually built separately from the works on account of noise, etc., but near enough to be within easy reach of the various time clerks, draughtsmen, etc.

This applies also to the raw material stores and to the engine and chassis testing departments; the former may often conveniently take the form of a lean-to shed capable of accommodating a railway truck or a heavy waggon, and should preferably be placed adjacent to the machine shop. As regards the chassis testing and engine testing departments, these should always be out of the main building on account of the dirt, noise, smell and risk of fire, from all of which they are inseparable. The constitution and equipment of testing departments has already been considered at some length in these columns, so no attempt will be made to deal with them in detail now.

Editorial Note.

There is no doubt that the proper design and arrangement of a factory is a very potent factor in the profit-earning capacity of a business, and therefore the subjects discussed in this article are matters of very great importance. Many of the suggestions made by our contributor are of a highly controversial nature, and it would be easy to make an equally good case for various other schemes of arrangement. We should, therefore, be very glad to hear from any readers who can offer useful criticisms on Mr. Davey's scheme, or can suggest any desirable additions to it.

THE 40 H.P. NATIONAL CHASSIS

Which has been very successful in American stock car races.

THIS chassis is interesting to compare with European ideas, as well as with the 40 h.p. Velie design dealt with in our June issue.

The engine is clearly shown in Fig. I. and has a cylinder bore of 5 in., while its stroke is 5 11/16. At 700 r.p.m. it develops 35.64 h.p., which increases steadily until the maximum of 80 h.p. is reached at 1,800 r.p.m., and at 700 r.p.m. the mechanical efficiency is .852, while at 1,800 r.p.m. it is .8. The cylinders are cast in pairs and have the valves, which are $2\frac{1}{2}$ ins. in diameter, on opposite sides, ample jackets being provided for the cylinders and valve ports. In order to ensure proper cooling of these large valves—the exhaust in particular—the water inlet is placed on the exhaust side near the top, where the cool water can readily reach the exhaust valve pockets. It will be noticed that the water jackets are equal at every point, so that the metal can be run as evenly as possible, which greatly assists in cooling and ensures an equal expansion under all conditions. As, for an engine of this size, the compression is rather high, there must be some method of preventing leaks through the valve port plugs, and this, in the particular engine, is accomplished by using a cover with a bevel seat held down by a large hollow brass nut. This cover is ground to the seat so that it may be kept tight, and the spark plugs are carried in it. A feature of the valve guide construction is that it can be removed and replaced readily when necessary, which is accomplished by using a long rod with a pin at one end and a thread to receive a nut at the other end. The removal operation is performed by inserting the rod through the guide and cylinder head plug, and tightening up the nut at this end until the guide loosens sufficiently to permit its removal by hand. This guide practically reseats itself, as it has a slightly tapered seat, and a few light taps with a hammer will fix it in position. Another feature in connection with this valve guide is that both it and the boss in the cylinder are kept as light as possible, so that cooling may be rapid and heat prevented from affecting the valve springs. This is important, as it will be understood that good spring action is necessary in order properly to seat the large valves. The water jacket wall is also kept as thin as foundry conditions will permit, and there is a large opening in the top of the cylinder provided with an aluminium cover for cleaning purposes. It will also be noticed that the combustion chamber head is designed with an additional thickness of metal.

The pistons are 6 ins. long, are light, thoroughly ribbed and have four eccentric rings, while they are also provided with oil grooves at the bottom. The gudgeon pins are $1\frac{1}{4}$ ins. in diameter and are held by set screws, secured by the usual lock nuts. An ordinary I section is used for the connecting rods, the upper ends holding hard bronze bushings $2\frac{3}{4}$ ins. long as bearings for the gudgeon pins. The lower bearings are of die-cast white bronze 1 15/16 ins. in diameter and 3 ins. long, and the caps are held in position with four studs and two lock nuts for each stud, the rods having adjusting packing pieces properly doweled for taking up the wear. It will be noticed that the feature of using two lock nuts also applies to the main bearings, cylinders and other parts of importance, instead of spring washers or cotter pins.

The crankshaft is forged from chrome-vanadium steel and given the proper heat treatment. All the bearings have large fillets in order to strengthen them materially, and they are ground accurately to size: the front bearing is $2\frac{1}{8}$ ins. in diameter and $3\frac{1}{2}$ ins. long, the centre bearing $2\frac{1}{8}$ ins. × $3\frac{3}{4}$ in., while the rear is $2\frac{1}{8}$ ins. × $4\frac{3}{4}$ ins. The long crank webs between cylinders one and two, and three and four, are of rectangular section 1 11/16 ins. × $2\frac{1}{2}$ ins., while the shorter webs are 1 3/16 ins. × $2\frac{1}{2}$ ins. From these proportions it will be seen that this shaft is of ample strength, and it will also be noticed that the flywheel is located on a shoulder at the rear end of the shaft, while it is held to an integral flange by six $\frac{1}{2}$ in. nickel steel bolts. The crankcase is divided horizontally and all bearings are carried in the upper half, so that the lower half may be dropped and the bearings inspected for wear.

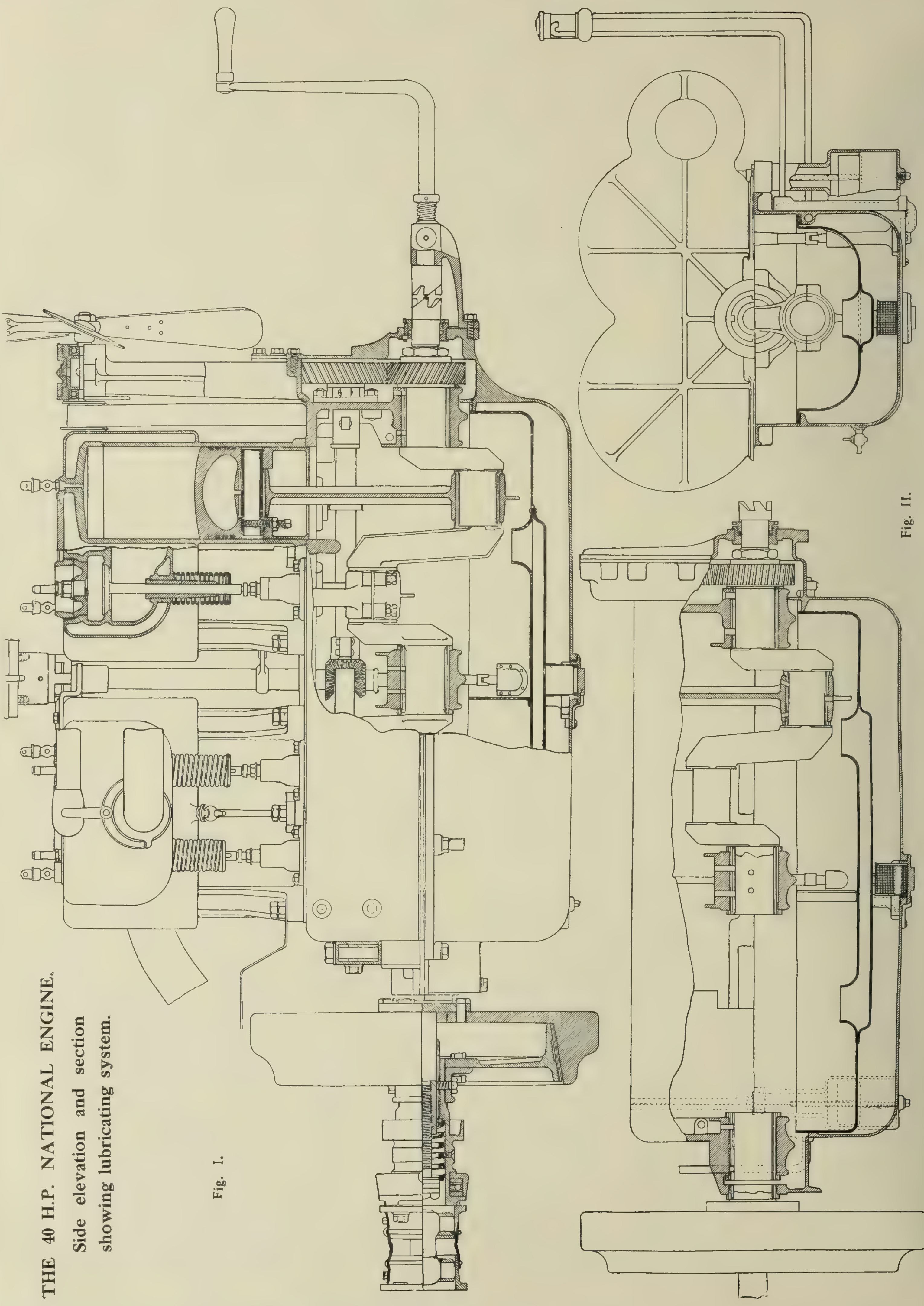
Turning to the valve operating mechanism there are bronze half-time gears with helical teeth meshing with steel pinions on the crankshaft, and on the pump and magneto drive shafts: it will be noticed that the thrust of these gears is taken by a collar on the front camshaft bearing. The valve tappets are of conventional design, and are provided with adjustment, while each also carries a pressed steel cup which acts as an oil retainer and is held in position by the adjusting screw lock nut. Steel forgings with the cams integral are used for the camshafts, and are 1 1/16 in. in diameter. A noteworthy feature of the camshaft bearings is that they are divided horizontally, so that they may be taken up when wear occurs. At the centre of the exhaust camshaft bevel gears are placed to drive the oil pump and the distributer, the latter being carried above the top of the cylinder, where it is accessible and kept away from heat and water.

Steel tubing is employed for the exhaust pipe, and the intake is made of copper tubing, while brass is used for the water pipes, of which the inlet is on the exhaust side near the top of the cylinder casting, while the outlet is placed on the intake side at the same height as the exhaust. A good feature in connection with this arrangement is that all pipes are held by four drop forged clamps and studs with lock washers and nuts, and all flanges are ground so that no washers are required, each pipe end projecting slightly into the cylinder to hold it in position while the clamp is being drawn up.

The water pump and magneto are placed on the exhaust side, the pump being driven through an Oldham coupling and having its shaft extending right through, for driving the magneto. This pump is of the centrifugal type, and is provided with a large stuffing box at each end of the shaft, while the Bosch magneto is driven from the pump shaft through two small universal joints. The latter—and the Oldham coupling also—are covered with leather boots packed with grease, which appears greatly to reduce the wear and noise usually present with these couplings. A platform integral with the upper half of the crankcase supports the magneto, which is located by two $\frac{1}{2}$ in. dowel pins, set diagonally in the base. It is held in position by a wide brass band and thumb nut, which makes a simple and accessible arrangement. The entire drive may be seen in Fig. VII., the side elevation of the chassis. It will be noticed that a large oil filler is situated at the centre of the crankcase, and all these parts are placed close to the level of the top of the frame, to make them accessible.

The fan is mounted upon two H.B. ball bearings which are carried in a brass cage, having its outside diameter turned eccentric to provide a means for adjusting the flat canvas belt, the bracket carrying these bearings and cage being divided, so that the cap may be loosened and the cage oscillated to the proper point, by a lever cast integral with the cage. A driving pulley is carried on the pump-driving shaft just at the back of the half time case. A cellular radiator is used, and the entire cooling system has a capacity of seven gallons of water, while drain cocks are placed in the radiator and pump. Drain cocks are also placed at the bottom of the water jackets on the cylinders, this being necessary because of the position of the water pipes.

A Schebler carburettor is used provided with a water jacket to heat the mixture properly before reaching the cylinders, and the carburettor is carried well to the front of the engine where it is accessible, while it is rigidly supported by a bracket bolted to the crankcase upper half. Petrol is supplied by gravity, and the tank has a capacity of twelve gallons. The starting handle is carried on the front end of the crankcase in an extension bracket, so that no stress is placed on the radiator, and a compression release is fitted, to facilitate easy starting, by connecting the four compression taps with a rod running through the dashboard. A neat stuffing box will be noticed at the front end of the crankshaft, where the starting handle is attached, while the rear bearing is carried well out from the splash trough as shown in Fig. II., a dividing bushing being used here to prevent leakage. Fig. II. is a sectional elevation of the crankcase for showing the lubrication system, both longitudinal and cross sections being given. Oil for the splash is carried in the pressed steel basin which is riveted to the lower half of the crankcase, there being a filter through which overflowing oil passes on its way back to the pump. This combination also acts as a compensating device for the oil level when climbing hills. The pump is located at the centre of the crankcase upon the left hand side, and is driven from the vertical shaft through an Oldham coupling. Oil



THE 40 H.P. NATIONAL ENGINE.

Side elevation and section showing lubricating system.

Fig. I.

Fig. II.

is pumped from the sump to a tell-tale on the dashboard, part of the lead being carried in the crankcase and having small holes drilled in it to permit some of the oil to reach the crankcase without going to the dashboard. The oil is returned from the tell-tale to another lead carried in the crankcase to the rear bearing and also to the half time case. Splash is caught in pockets and fed to the bearings as required, while the connecting rods are provided with scoops to distribute the oil. A float connected to a needle in the supply tube shows the position of the oil level, and in addition to this means of lubrication, oil and grease cups are placed at all points of vantage throughout the entire chassis.

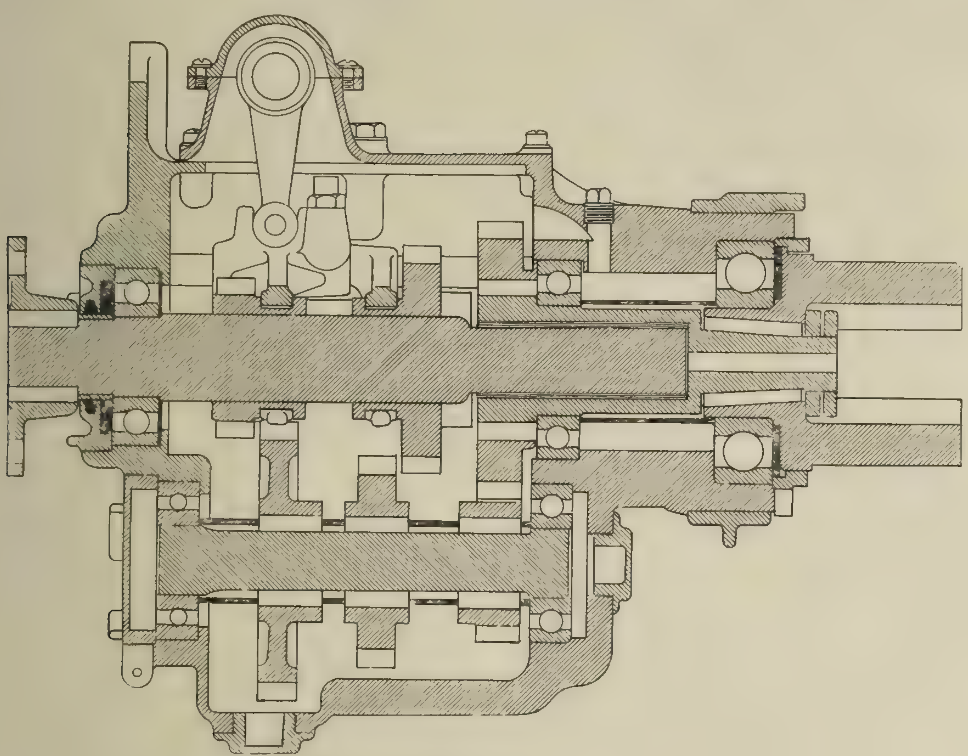


Fig. III.

Fig. I. depicts the cone clutch which is mounted upon the extension of the crankshaft. It is of conventional design, being provided with ball thrust bearings for the spring and disengagement. The universal joint is also shown in this view it having hardened steel blocks pivoting in drop forged flanges. A clutch brake is provided which is shown clearly in the view of the chassis; this has a spring to compensate for the pressure, so as to make the brake action good and not too sudden.

The gearbox shown in Fig. III., is a decidedly neat design, and perhaps the most noticeable point is the small space taken up by the gearbox countershaft, which is placed under the driving shaft, the length of this shaft being but $6\frac{1}{2}$ ins. between bearings. Only three forward speeds are given, the direct drive being through a clutch which is forged in one with the driving shaft that carries the universal joint. Two pinions, not shown in Fig. III., are carried on the left hand side of the case for reversing, which is obtained in the usual way the shifting fork having an extension which brings the gears into mesh. A heavy spring is used to throw the reverse out of gear.

Gears are made of special 20 ton carbon steel, are 6 pitch, and are given three heat treatments, after which they are casehardened. They are held to their shafts by two flat keys, and the shafts are made of chrome-vanadium steel. The bearings are H.B. ball bearings of liberal size mounted directly in the aluminium case, brass housings for carrying the bearings probably being omitted on account of the short length of the shafts employed, which tends greatly to reduce the reaction at each bearing. In any case it would seem best to use these housings as in time the bearings will probably loosen and housings will have to be inserted. The gear ratios are top speed direct, second speed 1.67 to 1, first speed 3.31 to 1 and reverse 4.68 to 1, while the rear axle reduction is three to one. There is also provision made for taking up the bearings from the front end of the case, and ample care is taken to prevent leakage occurring at either end. The gear striking forks work on stationary shafts, and are locked with the usual spring controlled plungers, which are carried in bosses integral with the striking forks, while the selector lever is carried inside the gearbox, where it is properly lubricated.

Fig. IV. shows the control levers which operate in the conventional H slot. By moving the control lever backward in the quadrant reverse is obtained, while first speed is in the inner slot in the forward position. In the outer slot, the forward position is second speed. As a special guard against entering reverse a lock is fitted which is controlled by the small spring and lever located at the handle of the control lever. The emergency or hand brake lever is placed inside the control lever, and operates on the usual ratchet tooth sector.

From the gearbox the drive is through a single sliding universal joint, which is made part of, and automatically lubricated from, the gearbox, the lubrication being accomplished through a small hole in the driving shaft. The heavy stress at this point is relieved by carrying the gearbox rear end in a swivel bearing, the torque tube being mounted in the same line as the universal joint and free to move in an endwise direction.

Fig. VI. shows the rear axle, which is of the full floating type, mounted upon Timken roller bearings. The housing proper is made of cast steel and is provided with a large cover for inspection, while it has specially drawn seamless steel tubing $2\frac{1}{2}$ ins. in diameter brazed into it, which extends to the outer edge of the wheel hub and carries the wheel bearings. Timken roller bearings are placed at each end of the differential and behind the driving pinion, also in the torsion tube, to support the universal joint and the drive shaft. The driving gears are four pitch and forged from $1\frac{1}{2}$ per cent. nickel steel, while the driving shafts are chrome-vanadium 1 11/32 ins. in diameter. The torsion tube is $2\frac{1}{4}$ ins. in diameter and threaded into the cast steel flange, which is bolted to the housing proper: it has a steel nut brazed on it for adjusting the bearing supporting the driving pinion, this adjustment being locked by a clamping bolt in the lower portion of the flange. The propeller shaft is also of chrome-vanadium steel and is $1\frac{3}{8}$ ins. in diameter. At the forward end of the torsion tube is a cast steel fork, which is free to revolve on the torsion tube and so mounted on the chassis frame as to permit a lengthwise movement, as is shown clearly in the chassis view. The axle is also mounted, so that it may swivel in the spring pads, which construction relieves the axle of any stress, while friction is reduced to a minimum, as all parts are thoroughly lubricated by grease and oil cups. Both sets of brakes are carried on the rear wheel drums, and are of the internal expanding and external contracting types, the inner one being used as a service brake and the outer for the emergency or hand brake. The internal brakes are metal to metal, and the outer are fabric lined on metal drums 15 ins. in diameter. All brake operating levers and shafts are carried inside the frame members, where they are thoroughly protected, and the entire axle is

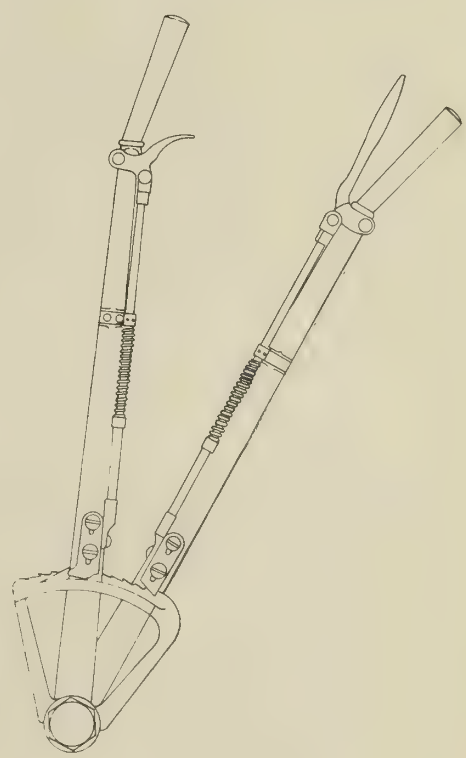


Fig. IV.

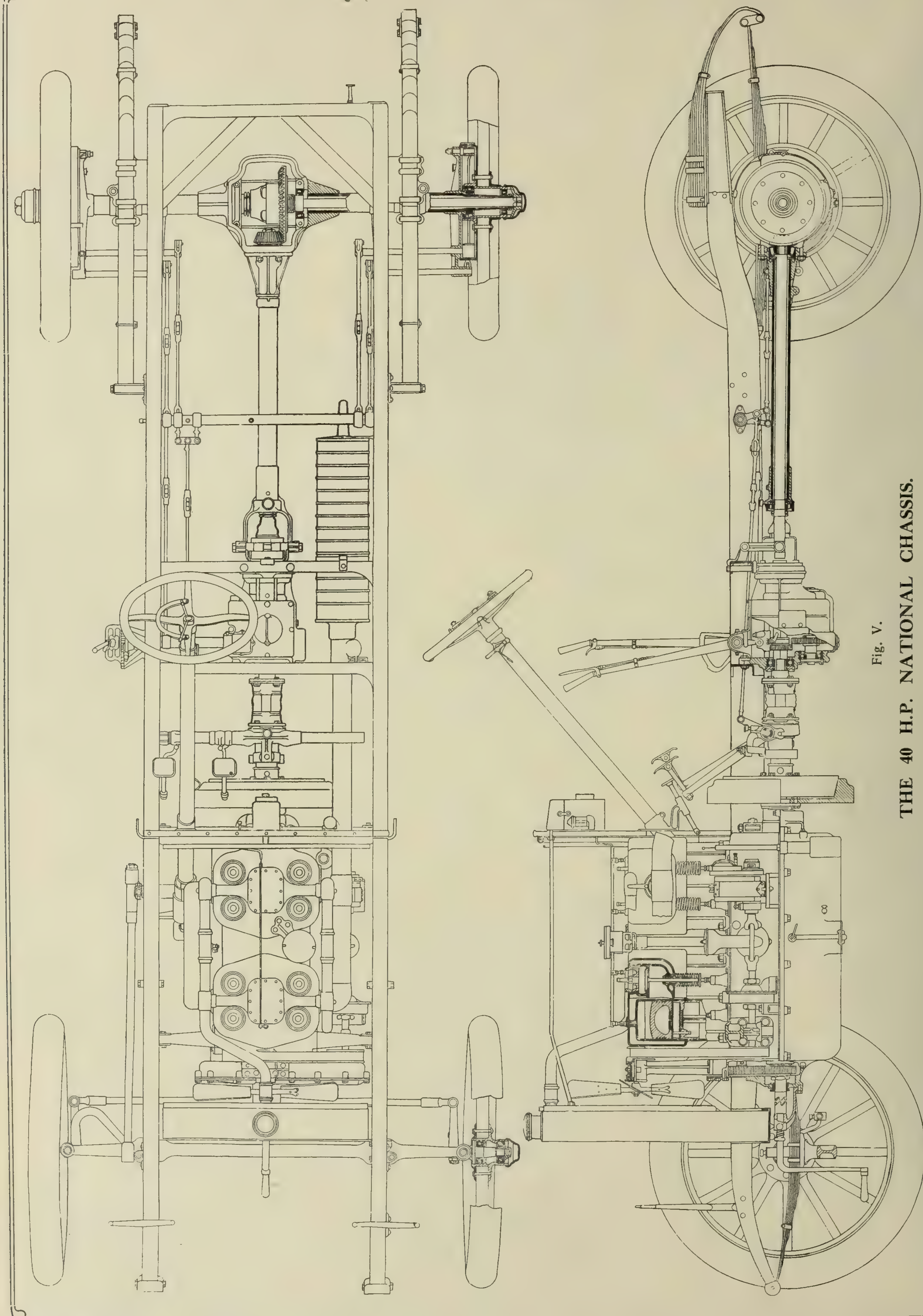


Fig. V.

THE 40 H.P. NATIONAL CHASSIS.

so constructed that the shafts may be removed and the differential taken down without dismounting the wheels or axle from the chassis. A heavy tie rod is placed under the axle to assist in resisting sudden shocks, and a drain plug is also placed at the extreme bottom of the housing.

Fig. VII. clearly shows both views of the chassis, the frame being made of special pressed steel having a channel section $4\frac{1}{2}$ ins. × 2 ins. × 11/64 ins. The main frame members are straight, but are arched over the rear axle to provide ample spring clearance, the frame width being 32 ins. throughout. It is mounted upon semi-elliptic springs 40 ins. × 2 ins. in front and three-quarter elliptic spring 48 ins. × 2 in. in the rear. Both sets of springs are nearly flat, and the entire mechanism is mounted well between the spring centres. The front axle is of conventional design running upon Timken roller bearings, the axle proper being a drop forging of carbon steel, while the steering arms are nickel steel forgings. All steering connections are provided with the usual leather boots packed with grease.

The steering column is of the worm and sector type provided with thrust bearings and adjustment. It is securely fastened to the frame side member, and the spark and throttle levers are mounted above the wheel, while below it is mounted a lever for the carburettor air control.

The radiator is carried in an inverted pressed steel channel, held by two studs in the radiator bottom and a stay rod running to the dashboard. A noteworthy feature of the radiator is the large filler provided with a rubber cap, which is cf such size that the radiator can be filled without the use of a funnel. The engine is mounted upon two aluminium alloy arms of I beam section in front and a hollow rectangular arm of cast steel at the rear. These arms rest in brackets having swivel joints and provide a flexible mounting for the engine, relieving it of any stresses due to road irregularities. This construction, combined with the gearbox mounting, makes a flexible driving unit, while the lining up is easily accomplished by packing up the gearbox in the customary manner.

The pedal arrangement is neat, each being adjustable for height, while the clutch pedal has an adjustment for the wear of the clutch. A foot throttle is also shown carried in a neat extension of the steering column bracket immediately between the clutch and brake pedals.

Ample provision is made for the adjustment of the brake rods and these parts are all steel forgings, but it seems as though it might be advantageous to place this adjustment where it would be more accessible.

The chassis has a wheelbase of 10 ft. 4 ins., and is provided with artillery wheels having 36 ins. × 4 ins. tyres mounted upon quick-detachable rims. In use the car is powerful and flexible. The clutch action is very good and smooth under all conditions, while the gears are exceptionally quiet, owing, no doubt, to the care taken in their manufacture and the short shafts employed. The clutch brake action is very good, and when changing speeds no noise can be heard except the click of the control lever striking the end of the slot. Taken altogether the car is very easy to handle, holds the road well, and very little skidding is noticeable when applying either set of brakes.

Criticism.

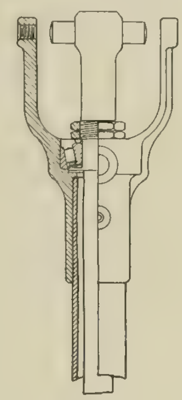
In pronouncing an opinion on any point in the design of the above car, it must be borne in mind that the conditions under which such a vehicle has to work are vastly different to those appertaining on this side of the Atlantic, and that the design has been evolved to a great extent by the extreme badness of the great majority of the roads in America.

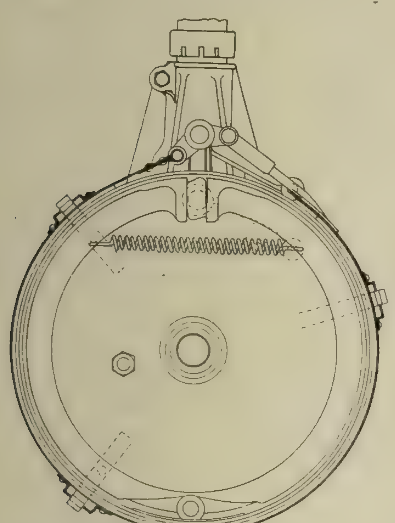
One point, however, which will immediately strike the European designer, is the absence of ribs on the combustion chamber, and the consequent thickening of that chamber's walls in a manner which is extremely likely to store heat to an injurious degree. Although in comparison with average practice the cast iron pistons might perfectly correctly be called light, to one conversant with design in this country, the great amount of totally unnecessary metal would call for some comment, together with the grub screw securing device seen in the sectional elevation of the engine. Fig. I. But it must be remembered that this is one of the points in which conditions have limited design, since the necessity for frequently cutting out the engine, and the consequent absence of prolonged periods entailing high engine speed have minimized the importance of light reciprocating parts for engines used in touring work.

As will be seen in Fig. I., the valves are on opposite sides, entailing the duplication of driving details and rendering it more difficult to obtain that degree of silence so strongly desired in this country, while the valves themselves seem strange, as they lack the sweeping radius common to the European design.

Care has been taken to provide the gearbox with ball bearings of extreme dimensions, but it is noteworthy that neither here nor in the engine has provision been made to prevent the escape of oil or grease in a trustworthily effectual manner.

As in the engine, the designer has obviously taken extreme care to eliminate the likelihood of external stresses affecting the gearbox and the attachment by which this has been safeguarded is worthy of study.





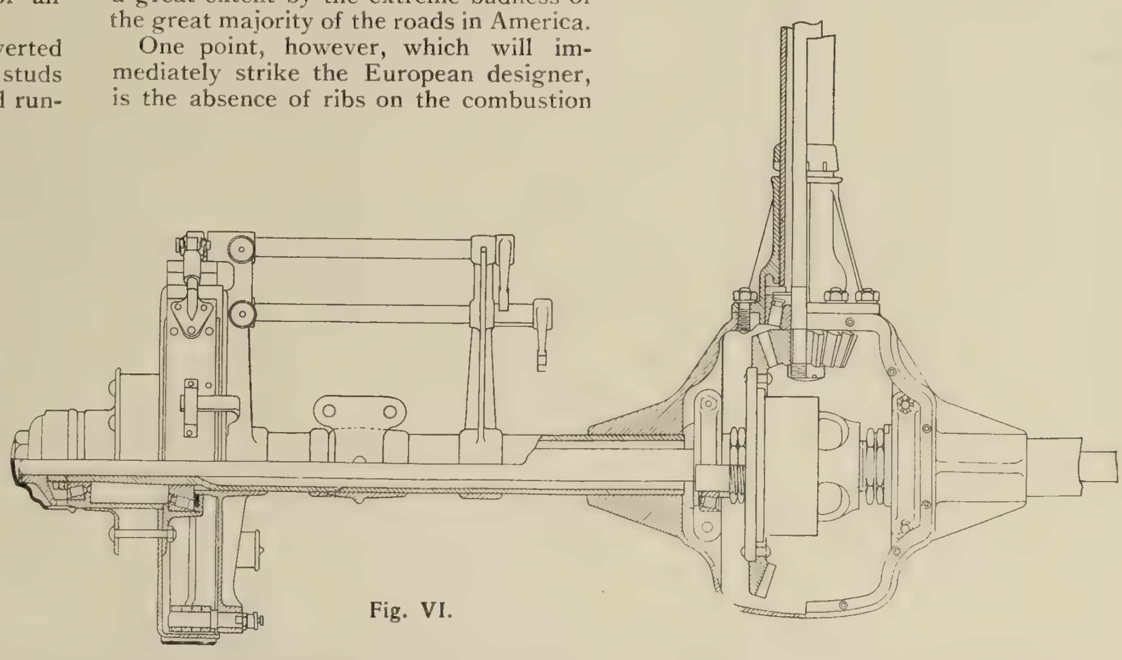


Fig. VI.

Taken as a whole there is much that could be improved in the engine design, but greater care has been bestowed on the chassis auxiliary parts with regard to frame distortion than has been noticeable in designs of similar types of cars in European practice.

Again, the peculiarities of the brakework are largely to be accounted for by reason of the fashion prevalent in America, and the decidedly clumsy appearance of the rear axle would not be noticeable if the external brakes were removed.

One good point in the design is also to be found in the back axle, this being the ready means for adjustment of both the bevel pinion and crown wheel provided by the mountings of the Timken bearings. The hub bearings, too, are well spaced, and should give rigid support to the road wheels.

STEEL CASTINGS IN AUTOMOBILE CONSTRUCTION.

The Probable Increase in their Use with the Improvements in Methods of Manufacture.

By R. W. A. Brewer, M.I.M.E., M.I.A.E., etc.

THE interesting and important paper read recently by Mr. L. A. Legros at the Institution of Automobile Engineers on the use of pressed steel in automobile construction cannot fail to cause engineers to consider the manufacture of many important details in steel by other methods than those described. In the following it will be necessary to take Mr. Legros' paper as a basis of this investigation and to draw certain inferences from his conclusions. The very last paragraph of his paper strikes rather at the root of the matter where it is shown that the employment of methods for pressing sheet steel to suitable shapes would prove a most important factor in the production of light and cheap motor vehicles, *provided always* "that the number of any given pattern required is sufficiently large to warrant the outlay on plant and tools for pressing." This means that in order that pressed steel work may be taken advantage of in the matter of cost, the expense involved in the initial outlays for dies, etc., must only bear a very small proportion to the value of the work.

Dealing now strictly with the English methods of construction, where it is quite usual for one firm to turn out several patterns of cars, and where no very great demand exists in any one year for pieces of the same size and pattern, it is evident that increased adoption of pressed steel work is not generally likely to prove beneficial.

In the past great difficulties have been experienced in casting steel of intricate formation on account of the viscosity of the molten metal and, for the same reason, it was practically impossible to run a steel casting less than a quarter of an inch thick. Even then unsound castings were met with in large proportion, and the presence of blow-holes was the rule rather than the exception. It is quite obvious that it is much cheaper to *cast* many of the parts used in automobile construction (and referred to by Mr. Legros) than to press them, particularly in comparatively small quantities. Difficulties occur in pressed steel work where the metal has to be of varying thickness, and Mr. Legros particularly refers to the fact that much yet remains to be done to perfect the methods of jointing thin stamped or pressed work to stouter flanges. If parts which are required to be treated in this way can suitably be cast, these difficulties disappear, and the present methods of welding electrically, or by the oxy-acetylene process, will not be required.

Steel castings are much more understood and appreciated nowadays than they were a few years ago, and many improved methods of producing them have been devised. Cast steel cylinders and pistons are well known to most of us, but considerable trouble and delay generally occurs in obtaining good, sound steel castings of this description. In the majority of processes annealing of the castings must be resorted to in order to render them suitable for tooling. Supposing the process about to be described were more generally adopted a large number of motor car parts, and particularly such as are not required in large quantities, could be more easily and cheaply made than they are at present. Referring again to Mr. Legros' paper, we at first see that for pressed work the material must be very ductile and have sufficient tensile strength to ensure that the metal shall not suffer fracture when drawn down between the plunger and the die : this very act of drawing subjects the material to severe stresses and creates a grain, leaving it naturally weaker across the grain, whereas a similar part cast in a mould is perfectly homogeneous throughout its mass. This is a very important consideration in connection with all pieces of material subjected to intermittent or alternating stresses, and in automobile construction practically every portion of the steelwork is subject to these conditions. Fatigue in the material takes place very rapidly, particularly in material which is not fine grained, and heat-treating processes are adopted in many steels to produce greater strength or to refine the grain. Such treatment naturally adds to the cost of the material, and only steels which will respond to the treatment in some proportion to the cost of the operation, can commercially be employed. Many commercial steels are delivered to the manufacturers of automobile parts in a coarsely crystalline state, due to the last hammering operation at the steel mill, and this can only be refined by further drop forging or heating operations. If the bar is to be forged, the crystalline condition does not signify much, but the drop forging should afterwards be annealed to rectify any crystallisation, which may not disappear under a single heat treatment. It will be seen, therefore, that steels which have been much worked previously are sometimes liable to lose their nature in the process, and for this reason alone a steel casting made by the latest methods is more likely to be reliable than a forging.

One must not lose sight of the fact that design plays almost as important a part as material in the ultimate result, and a suitable arrangement of material in the fillets will reduce the possibilities of fatigue enormously in any particular piece of steel. Any part which is of a fairly complex nature is much more easily constructed to a sound design when cast.

Chassis Parts.

We will now consider the application of steel castings to the construction of various parts of the chassis and engine, and thus obtain some idea of the very large number of details which can be conveniently constructed in this material.

It is unnecessary to point out the great improvements which have taken place in back axle design during the last few years and, whereas this portion was originally constructed of drawn tubes, screwed or pressed into a steel differential gearbox and steel spring seats, this construction has in many cases been discarded in favour of a complete system of steel work cast or pressed, so that taper tubular construction can be employed and rigidity gained thereby. Taper or bell-mouth axle tubes form a very rigid construction, but the cost of manufacturing them in dies is, of course, very considerable.

In order to avoid the expense and difficulty of pressing complete halves of axles they are sometimes built up from castings, but several makers in recent times are now using cast steel axles of somewhat large and heavy dimensions. The differential casing lends itself very well to construction by casting, and it does not signify very much whether the two halves are similar or not, as it is only a matter of using a different pattern, which is very cheap as compared with a die. Some of the English cars are fitted with cast steel axles of satisfactorily thin material, made by ordinary processes, but there is undoubtedly some difficulty in getting the metal to flow in a long thin casting such as an axle tube or a propeller shaft casing.

As axle weight is totally unsprung it is obviously of advantage to keep it down as much as possible, and the thinner the casting—provided that the material is disposed scientifically—the more satisfactory will be the finished article. Part of the weight is naturally contained in the differential box, and here again is a part which can advantageously be made in the form of a thin steel casting. Aluminium bronze, and cast iron have all been used to considerable extent for the construction of gearboxes and engine base chambers, and each one of these materials has some disadvantage for the particular purpose. Thus in the case of a fairly powerful engine, an aluminium base chamber or gearbox is always liable to distortion, or even fracture, by reason of the enormous stresses set up either by the engine itself or by road shocks transmitted to it. Again, there are stresses of considerable magnitude impressed upon such parts as the lugs of these castings, when a change of direction or momentum comes upon a heavy mass like the engine. These stresses are well illustrated in the case of a rail tractor, and are known as "buffer shocks," because if a tractor is suddenly bumped against trucks the stresses set up may easily be sufficient to cause the engine to carry away from the frame.

When we look for a stronger or stiffer material we have the choice of cast iron, which is heavy and brittle, or gunmetal, which is expensive, and the obvious alternative to these is, of course, cast steel. The latter material has an advantage over aluminium in that stud holes can satisfactorily be tapped into it, and it also resists tension far better than any of the materials more generally in use.

Within the last few years a good deal of use has been made of steel pistons, and the enormous reduction in weight which has been effected by their use has made possible engine speeds which would have seemed quite incredible four or five years

ago. The steel pistons now generally in use are described very ably in Mr. Legros' paper, where he points out that owing to the presence of the gudgeon pin bosses a considerable amount of work is involved in the construction of such pistons. The pistons made by the pressing or drop forging processes must be constructed with the bosses carried up to the piston heads and the superfluous metal bored out afterwards: when further lightening is required a special tool must be employed, with its bar passed through the hole for the gudgeon pin and a cutter of an overhung type employed to machine the bosses circular. Even then considerable metal is left in the fillets on account of the cylindrical formation of the piston.

Obviously cast steel pistons would be much cheaper to construct, as all this machine work would be avoided, though some makers have endeavoured to get round the difficulty of the gudgeon pin bosses either by building them into the pistons or making the pistons themselves in two parts screwed together. In the writer's opinion such a form of construction is undesirable for a light moving part subjected to very rapid reversals of stresses of considerable magnitude.

Passing now to other details which are either cut from the solid or could possibly be made of malleable castings, we find that wheel hubs and brake drums are very suitable for construction in cast steel, as when these parts have flanges made in them the flanges can be of any desired thickness, and any coring or filleting can conveniently be formed in the mould. The automobile engineer has also use for a number of articles such as dumb irons, pedal levers, shackles and step brackets, which can be cast more easily and cheaply than they can be forged. When lightness is required, as in small car construction, advantage can be taken of sections which give the maximum strength combined with a minimum weight, and in a steel casting it is no more expensive to produce these sections than any of the plainer and heavier ones.

Before leaving the consideration of cast steel details there is one very large field of usefulness open in the construction of motor omnibus wheels. There are several types now in daily use, but at least one pattern has proved unsatisfactory by reason of the wheels cracking. It is quite feasible that in a steel wheel cast by an ordinary process and in which the section and mass of material varies very much from point to point, brittleness may be set up in cooling, and this cannot always be rectified by annealing. Wheels of the Sankey type, consisting of two pressed halves welded together, could be made in hollow steel castings, and in ordinary quantities would no doubt be cheaper than pressed wheels, while they would also have the adantage of requiring no welding.

Fractured.				Extension.		Appearance of Fracture.
Dia.	Area.	Difference.		In 2 inches.	In 3 inches.	
		Area.	Pt. bent.			
inch.	sq. inch.	...	...	per cent.	per cent.	...
.645	.327	.166	33.7	31.0	27.7	Silky

Table I.

As Mr. Legros points out, the present methods employed for welding either by electricity or by the oxy-acetylene process are not such as the engineer usually favours in constructional work, particularly for material subjected to repeated loading. The use of castings in very many instances avoids the necessity for welding which exists where pressed work is employed.

Description of the Stock Steel Process.

As an example of one of the most modern methods of constructing thin steel castings, which when finished show a considerable tensile strength and ductility and which do not require annealing, the "Stock" process is of considerable interest. In a manner this process somewhat resembles the well-known Bessemer system, since a converter is used to transform the pig-iron direct into steel by driving off the excess of carbon in the form of CO and CO_2, and then adding to the molten metal such ingredients as are necessary to convert the iron into any desired quality of steel. The Stock converter is lined with silica firebrick and the pig-iron is melted by means of liquid fuel and a blast of hot air, while the waste gases pass through a regenerator. Thus the air blast is heated to the required temperature of about 800 to 850 degrees Fahrenheit.

As distinct from the Bessemer system, in which cold air is blown through the molten metal, in the Stock system the hot air blast is blown on to the top of the metal, causing the surface to form into ripples. The blast continues for about six minutes with a half-ton charge, or eighteen minutes with a three-ton charge, during which time the impurities pass out of the mouth of the converter in the form of waste gases. When the charge is thus reduced an addition of ferro-silica, ferro-manganese, or other material, such as aluminium is made, according to the quality of the steel required, and the metal is then ready for pouring. On account of the high temperature of conversion the molten metal is very fluid, and castings can be run one-sixteenth of an inch thick in ordinary sand. By this method a very great advantage is obtained as no annealing is required, and the loss of time which this process occasions in ordinary steel castings is avoided. The accompanying tables give the results of analyses of castings made by this process, and also breaking stresses and percentage of contraction on a number of test specimens. These results are most satisfactory as indications of what can be done in the milder qualities of steel.

This particular process is not confined to the production of mild or low carbon steel, but any of the higher grades can be produced, even in small quantities if desired, by adding the necessary alloys to the metal in the ladle. It is obvious that steel such as this, which can be cast thin and can be tooled immediately it comes out of the mould and from which the moulds can be knocked off when the metal is still red hot has distinct advantages to the engineer.

Test No.	Description.	Original.		Ultimate Stress.		
		Dia.	Area.	Total.	Per square inch of original area.	
TT.	...	inch.	sq. inch.	lbs.	lbs.	tons.
2508	Piece 33 one end: other end 1002. Turned to ½ sq. inch area.	.792	.493	32,620	56,166 =	29.5

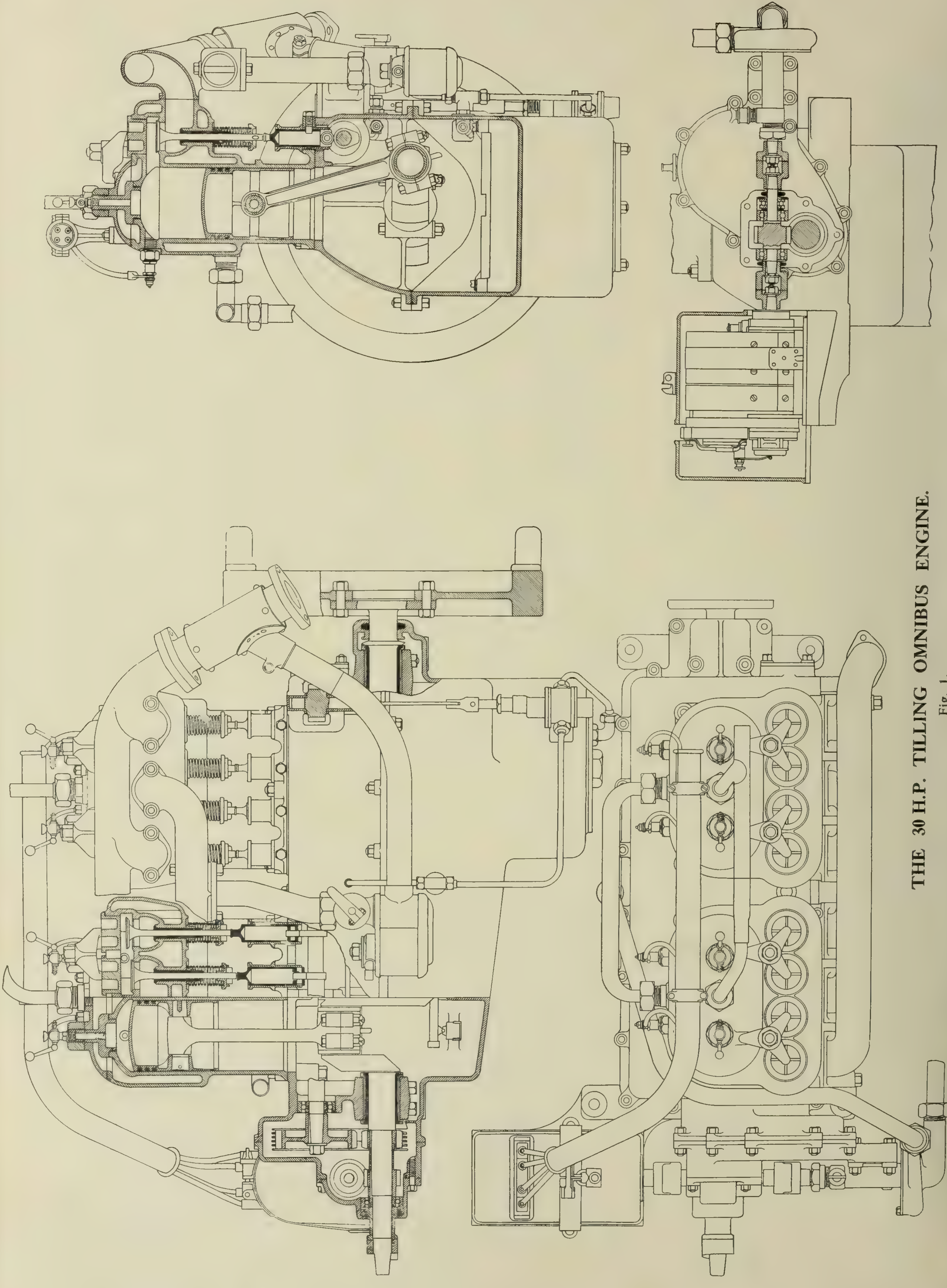
Table II.

THE STEVENS TILLING MOTOR OMNIBUS.

A heavy vehicle with an electric transmission and an exceptionally easy control mechanism.

Up to the present the motor omnibuses which have been running in the London streets have been propelled almost entirely by steam or petrol. It is interesting therefore to examine carefully the design of a petrol-electric vehicle which is intended for work under the same conditions as its rivals, and which is likely to be placed on the road in quantities that will make a trial of a really conclusive nature.

Before going into constructional details, however, it would be well to run over the particular system of control of which this vehicle is an example.



THE 30 H.P. TILLING OMNIBUS ENGINE.

Fig. 1.

The petrol engine is not under any circumstances coupled directly to the rear wheels other than electrically through the generator leads, and all driving effort is transmitted by the generator to the motor, which is coupled to the propeller shaft. No buffer battery is used, so that the engine is started in the usual manner with a crank, and, when the road wheels are at rest, is allowed to run at a speed below the "exciting" speed of the generator. A shunt resistance, which is operated by hand, governs the generator field, so that a strong field may be weakened to allow a species of slip by which the petrol engine can regain its normal revolutions. The torque of the series wound motor varies almost in accordance with the ampêres, and the speed with the voltage, so that the whole system relies on a generator so designed that it will deliver in accordance with the driving requirements.

Fig. I. shows four views of the engine, which has four cylinders of 105 mm. by 125 mm., cast in pairs and provided with ample water space, particularly in the neighbourhood of the valve chambers, where no less than 20 mm. width is allowed. Each cylinder is closed at the top by a screw-down plug, so designed that it forms both a water-jacket cap stud and a pipe into which the compression tap may be fitted, the whole arrangement being locked by a 20 mm. nut on the outside. A hole drilled horizontally in the upper part of the combustion chamber accommodates the plug. Push-in valve caps are used, secured in place by a dog which is adjusted by the usual stud and nut, while taking a bearing on a specially provided boss formed in the jacket cap. All the valve caps are carefully recessed in order that undue heat may not be stored therein, a point indicating some careful designing.

Fifty mm. valves are fitted, and they are unlikely to give trouble in use as a good sweeping radius is provided at the neck. A longer valve guide bearing would probably have been of some advantage, since considerable wear frequently takes place at such a point: however, it will be noticed that the guides are detachable when the lock nut at the bottom has been slacked off, a feature which is considerably in advance of many touring cars. Each tappet guide is housed in a boss provided on the cylinder foot and locked, when in position, by a 10 mm. set screw. Curiously, because it is practically essential for commercial engines, there is no adjustment for tappet wear, but it should be pointed out that the tappets have a large bearing surface in their guides. Rolling contact is provided for the cam, and the roller pin seems well up to its work without likelihood of undue shake developing. The camshaft, which is machined from the solid, runs on three ball bearings, and is driven by a chain wheel keyed to a taper at the forward end of the engine, while at the rear are attached the skew gears which drive the oil pump. A 44 mm. internal diameter cast iron pipe conducts the exhaust gases to a silencer, slung transversely across the frame, while a pipe of similar dimensions is used for the inlet from the Solex carburettor, whose attachment is made perfectly clear in the illustration. From the carburettor a long pipe is taken to a jacket on the exhaust pipe, so that warm air can be supplied to the jet and can be diluted with cold air drawn through slots on another sleeve registering with those on the pipe.

A three-bearing crank shaft is used running in plain white metalled bearings of ample length. In front of the forward bearing is the pinion which drives the camshaft through a silent chain, for which no adjustment is provided, though in this respect it is no worse than most, and the design might conceivably be greatly complicated by such an addition.

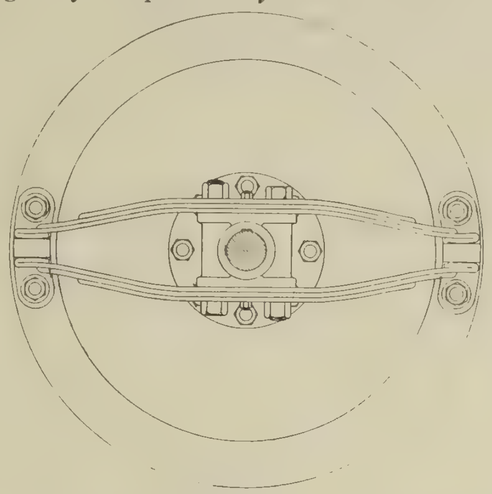


Fig. II.

Just beyond this pinion the cross shaft skew gear can be seen, while a small ball bearing steadies the crankshaft behind the jaw for the starting handle. One is pleased to note that a thrower ring effectually removes any oil which may attempt to escape from the tail bearing, additional security being provided by the felt washer seen. Possibly the inverted tooth chain is relied on to act in a similar manner at the forward end since there is no thrower ring. A stamped H-section connecting rod encloses the gun-metal and white metal big ends, for whose security and adjustment four 8 mm. bolts are used. The sectional and elevation gives a good

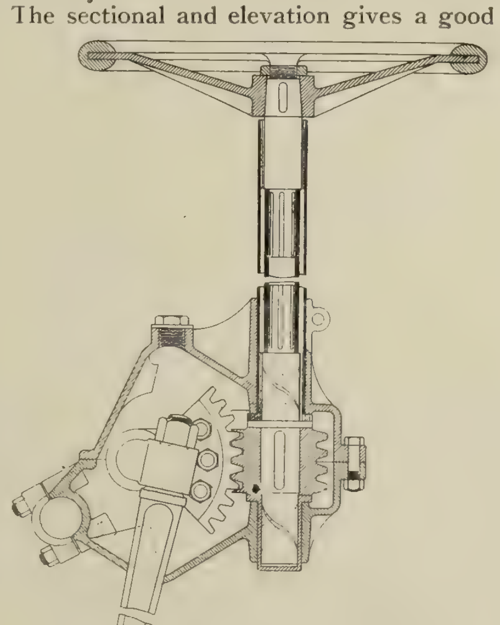


Fig. III.

view of the oil scoop, which is cast on the bottom half bearing. Of the gudgeon pin there is little to say, save that it is fixed by a taper on one end and a grub screw on the other, which should be certain security, albeit not easy to remove after some miles of use. The cast iron piston has three rings, and seem distinctly and admirably light for a commercial vehicle engine. A point worth noticing is in the ribs which strengthen the gudgeon pin bosses, as these latter appear somewhat light at first glance. A small oil retaining groove will be noticed machined on the piston skirt in order to aid in cylinder wall lubrication. On the plan view, at the forward end of the engine, a good idea of the cross shaft external arrangement can be obtained, it being only necessary to explain that both magneto and water pump are coupled by dog clutches. The view of the cross shaft in section reveals the ball thrust and journal bearings for the skew gear, the former of which are provided with washers for adjusting any wear which may take place, and small felt pads are relied upon for preventing waste of oil. An aluminium box encloses the whole magneto in order that as little dust as possible shall penetrate to the contact maker and winding, only a small slot being provided for the attachment of the high tension leads. Of course, in keeping with modern practice, the magneto can be removed by unscrewing one wing nut and removing the wiring. Wisely, a long bearing has been provided for the centrifugal water pump spindle, as seen on the right in the illustration, but the whole water system would be improved by the enlarging of the copper water pipes, as both outlet and inlet are small, while, as the radiator is in the position usually associated with the Renault, natural circulation might assist, or, even on some occasions, supplant the pump, which, however, is most desirable for traffic work.

Lubrication is on the constant level trough system, each big end having its own trough, which is supplied with oil by the rotary Albany pump seen bolted to the outside of the crankcase in Fig. 1, and, therefore, more accessible than usual. From the pump the external pipe conveying oil to the trough feed pipe can be seen, together with one end of the feed pipe and its nozzle, directly above the trough it supplies. Above each main bearing there is another trough, formed with holes drilled down to the bearing surfaces. These are at present filled by additional pipes and nozzles, but in the future it is proposed to do without these additional pipes, and to rely on the oil which is thrown up by the scoop. This is, perhaps, a debatable form of lubrication, inasmuch as true forced would probably be more economical and leave less to chance, while the pump and much of the piping could remain unaltered, but it must be remembered that the adopted form of lubrication has proved itself satisfactory on touring car chassis, and that high engine speeds are unlikely. Before leaving the lubrication, the extra large size of the sump which supplies oil through a filter to the pump should be noticed. The flywheel is remarkable only because of the peculiar form of coupling used between it and the dynamo. This is seen more clearly in the separate view, Fig. II., in which it will be observed that two laminated springs of considerable thickness and length are bolted to a bracket keyed to the dynamo shaft. These springs are then slid into place at either side of a pair of stout pegs bolted to the flywheel rim, care being taken to

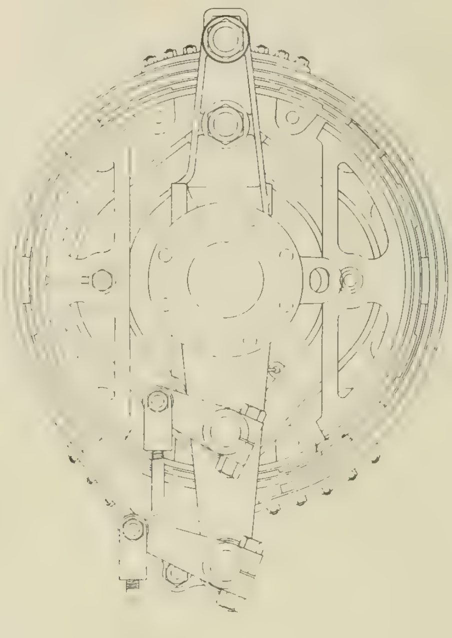
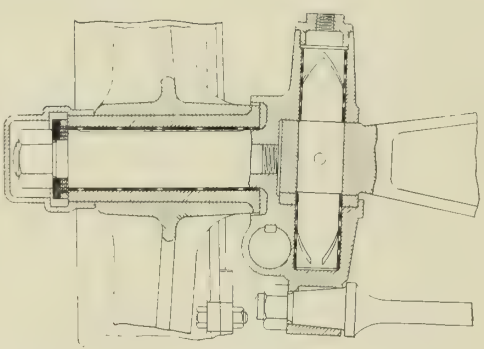


Fig. VI.



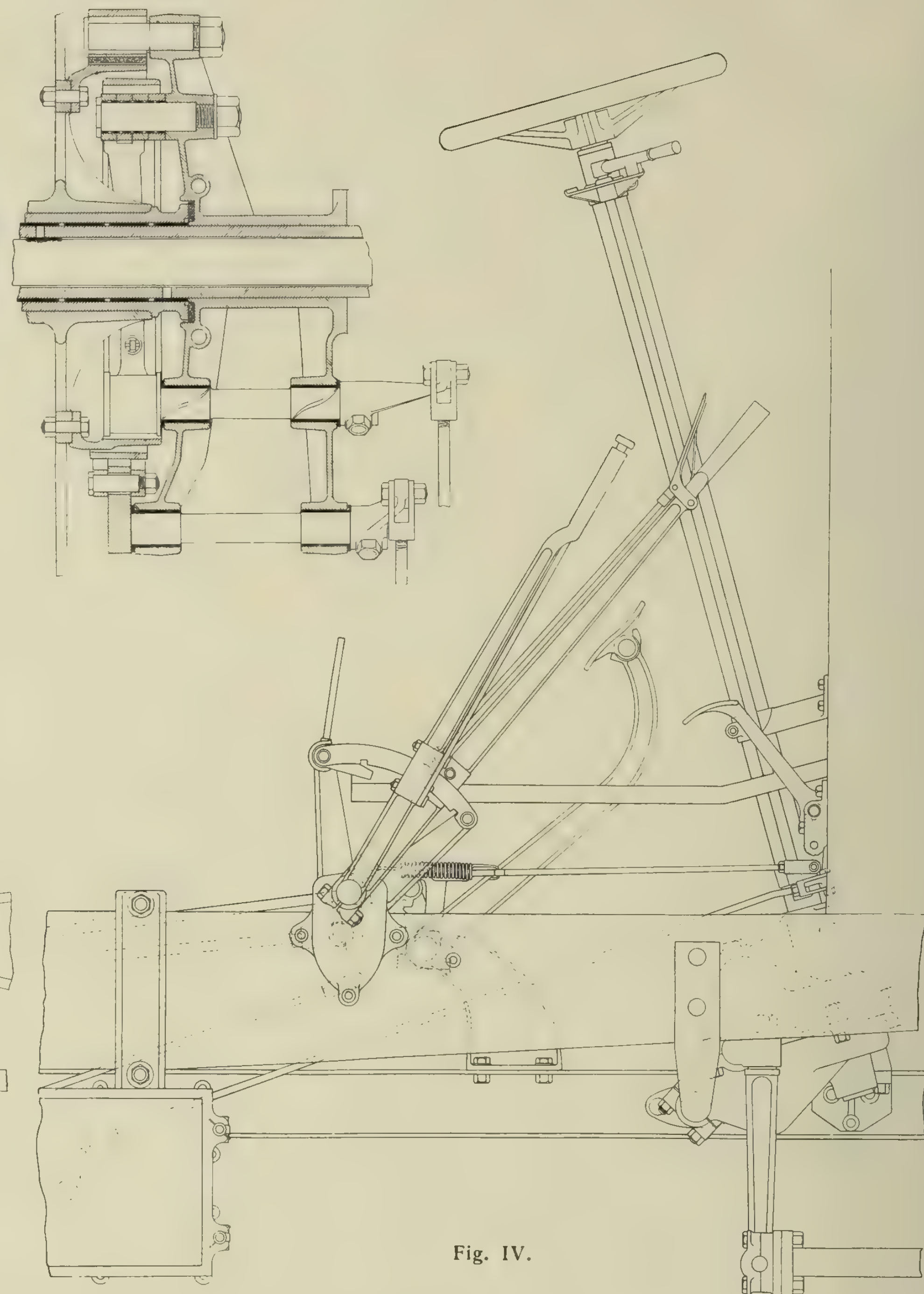


Fig. IV.

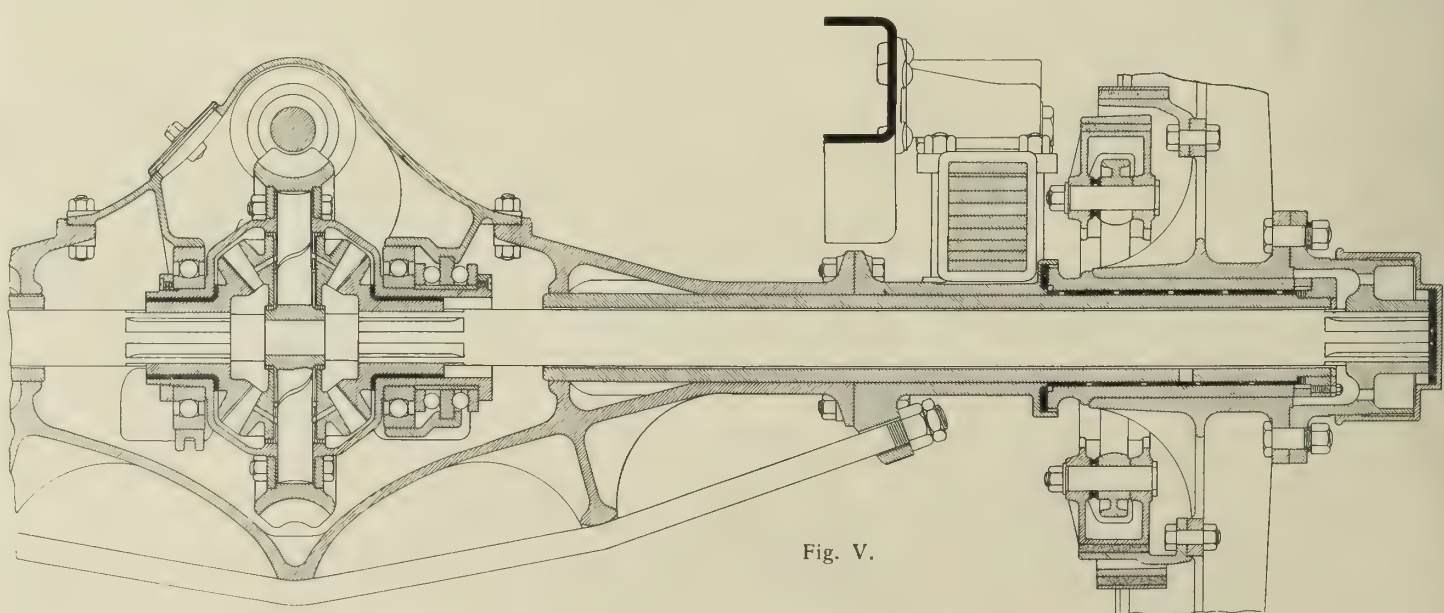
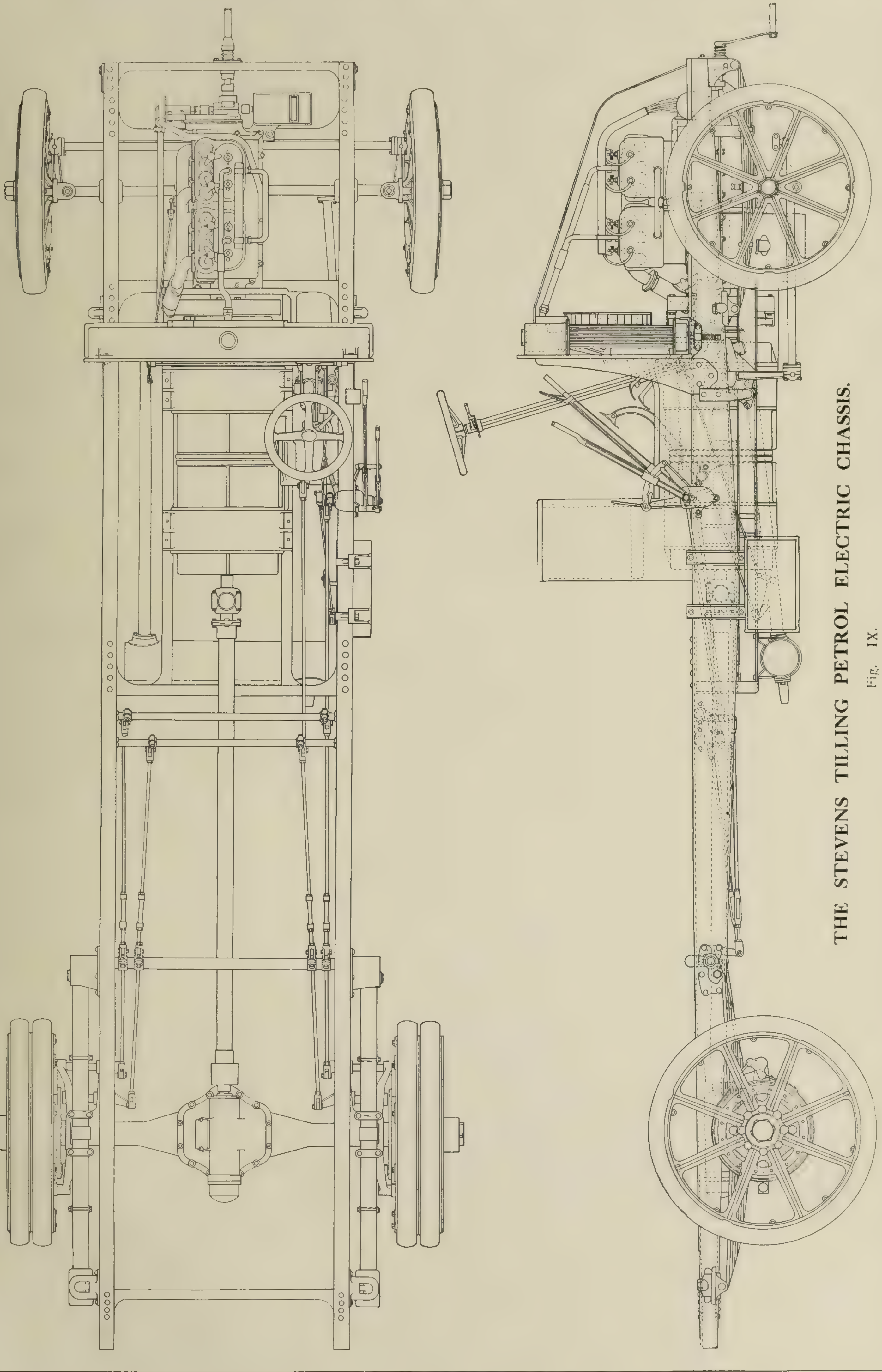


Fig. V.

TILLING OMNIBUS AXLE BRAKES AND CONTROL.



THE STEVENS TILLING PETROL ELECTRIC CHASSIS.

Fig. IX.

balance the whole arrangement so that an even running movement may be assured. The reason for such a remarkable coupling is to avoid jar or jerk on the dynamo armature, and to take up any stress due to the flexing of the side frame members when passing over bad surfaces.

Fig. IX. gives an idea of the positioning and arrangement of the electrical gear. The armature, which is constructed by W. A. Stevens and Co., runs on a nickel steel shaft, carried in ball bearings housed in the aluminium of the generator casing, and is a 25 kilowatt shunt wound machine designed to run up to 1,500 r.p.m. By means of a resistance placed in the box seen on the side of the frame behind the side levers, the field strength can be altered so that, on bad gradients, an increased armature speed is allowed through a weakened field. The commutator and brushes are enclosed in covers which however, are easy to remove, in order to clean and adjust the brush gear. Between generator and motor a large Sirocco type fan is fitted, so that there is little chance of the temperature rise of either amounting to a dangerous degree, and it is claimed that the actual running temperature is lower than in stationary work. Like the generator, the series wound motor runs on ball bearings which, together with the armature and its shaft, are exactly similar in size to those employed in the generator, while the brushes are protected in the same manner, special precautions being taken to prevent all avoidable dust entering the machine due to the fan action.

For the motor an efficiency of 90 per cent. is claimed, that for the generator being 88 per cent., while the insulation of each has been tested at 1,000 volts continuous, giving considerably above 50 megohms.

Connecting the tail of the motor shaft to the propeller shaft is a ball bearing universal joint. This is distinctly unusual, and is probably renewed with greater ease than a plain bush bearing, while in practice it should wear considerably better than the latter, if well protected from dust and water. At the rear there is a splined joint allowing a certain amount of sliding motion.

Fig. V. gives a sectional view of the rear axle in the vertical plane. It will be observed that the worm is placed above the worm wheel, a position which gives admirable results, while allowing a moderately straight through drive without an inclined engine. A point of interest concerning the casing is that it is a steel casting in one piece ended of course, at the flanges seen holding the spring pad. Cast steel is probably not the most sure material for an axle with the many bosses and sectional variations provided for in this design, and the danger of blow holes is always present in a comparatively large and complicated casting. As evidence of neat design the manner in which the worm and worm wheel, complete with all their bearings, can be withdrawn with the top cover is worthy of study. Of course, prior to such a removal the axle driving shafts must be removed, and this can be done through the hubs somewhat similarly to a touring car. Another noticeable point is the manner in which the differential spider steady pin is carried in recesses bored at the end of the axle driving shafts. At the outer ends the driving shafts rotate the wheels through a separate disc bolted to a flange on the wheel boss by a number of 15 mm. bolts. The wheel itself has a steel centre piece, which runs on a very long gun-metal bush over the axle tube, egress of oil at the rear being prevented by a felt ring held up against the flange shown, and lubrication provided for by holes which allow oil from the worm box to enter the bearing. As an additional

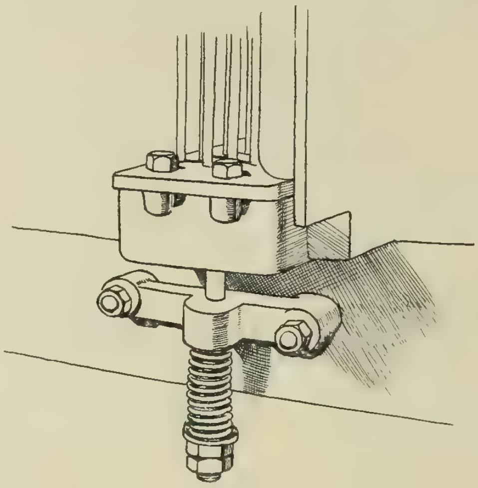


Fig. VII.

safeguard against shock, a couple of extra posts have been introduced, which the tie rod bears against. The whole axle appears quite robust enough to withstand the violent shocks inseparable from heavy industrial work. Both brakes are situated in the rear wheels, that actuated by the pedal being of the internal expanding type, with D shoes slung somewhat similarly to locomotive practice; the other, which is operated by pulling the inner of the two side levers, is an external contracting band brake, bell crank operated as shown in Fig. VI. Both have their own brake drums bolted to the wheel spokes, and each can be adjusted by twin buckles placed in an accessible position as well as by screwing up the tension rods.

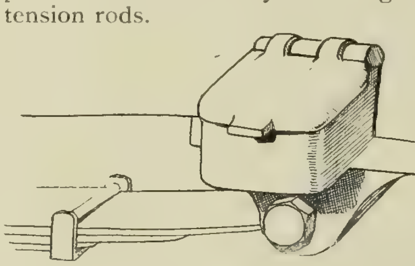


Fig. VIII.

Fig. IV. makes the general arrangement of the control gear clear. The small pedal seen near the foot of the dashboard is for throttle only, while the larger pedal is, of course, the foot brake. Of the side levers, the nearest is connected to the electrical switch gear, and is brought into the back notch for reversing, centre for cut off, and forward notch for normal running. In this latter position it remains for such a time as the vehicle progresses in a forward direction, the "off" position not being used, as the generator will not excite until 300 r.p.m. at which speed it can be kept throughout the duration of a halt.

Steering, Fig. III., is by worm and sector in the usual manner, and beyond extraordinary solidity is fully explained by the drawing. Perhaps it would have been well to provide a worm wheel instead of the sector shown, thereby obtaining at least four good adjusments, as wear is likely to be great at such a point. Absence of thrust ball bearing is at once noticeable, and possibly does not pay in work of this description, albeit one would imagine that free steering would be a great advantage on a heavy solid tyre machine, which, at its best, is stiff to steer. Like the rear wheels, those in front are carried on a plain bush which is lubricated with grease forced down the hollow spindle, as shown in Fig. V., by the screwing up of the hub cap. A very solid, well grooved pin, mounted in plain bushes, takes the steering swivel, a large greaser supplying the necessary lubricant, the whole showing signs of the care which has been taken to allow easy replacement of parts which are but too likely to damage. An excellent point is the gradual radius of the swivel arm into the main fork.

A radiator of the "Renault" type is used, mounted on the dash in a particularly accessible manner, which allows for slight movement, see Fig. VII. An aluminium top tank is used with plain vertical copper piping. Speaking generally, the whole radiator would be better were it larger, as traffic conditions are a very severe test of the efficiency in any radiator. Among the radiator tubes there is a belt-driven fan running on a ball bearing spindle, the whole fitting plainly shown in Fig. IX. The frame is] section girder stiffened by five cross members, the third and fourth of which are further braced by the dynamo-motor underframe. Long springs of 1,370 mm. carry the rear axle, and are clipped to the latter by a box bracket shown in Fig. V., accompanied by the U clips usually fitted, while both torque and radial stress is borne on the springs. Each spring is anchored at one end, while a slide and pad is formed for the other. A shorter spring, 1,010 mm., is used in front. Fig. VIII. illustrates the neat and effectual grease box used for the rear slide. It is impossible for this box to come loose or drop off, while its breakage is unlikely.

On the road the vehicle is quiet, and has a great advantage in the ease and simplicity of control, there being but a foot throttle and a small hand lever which need alteration, while it is impossible for the driver to do anything injurious to the machine or its passengers with the exception of pulling up too quickly, and using considerably more petrol than would otherwise be necessary. Undoubtedly the vehicle is much easier to handle than the more ordinary forms of transmission, save, perhaps — if other things were equal—electricity. Absence of a jerky start is a very great advantage, since it forms the one drawback to otherwise excellent machines at present on the road, while it is obtained without the use of doubtful batteries. Again, the instructions to a driver are of the simplest nature, as he has to be told to keep his engine at the same speed except downhill, where, as there is only the motor and rear axle gear to rotate, the machine runs with remarkable freedom.

It will be interesting to observe the progress of this particular type of vehicle under the stringent conditions of London traffic, and in the hands of comparatively unskilled men.

THE 80-H.P. ALL BRITISH ENGINE.

A most Interesting Design for Aeronautical Work.

The All British Engine Co. are putting on the market three separate sizes of engine, the smallest being of 40 h.p., with four cylinders of the vertical type, the second an 80 h.p. eight cylinder V type, and the third a 120 h.p. 12 cylinder V type. The bore and stroke of the cylinders is in each case $4\frac{5}{8}$ in. x $4\frac{3}{4}$ in., and it is the 80 h.p. that is described hereunder.

The crankshaft is made of vanadium steel cut from the solid, and has one throw to each pair of cylinders, with a bearing between each throw, while the webs are not cut away. Both the crankshaft and the crank pins are bored out to save weight, and light steel tubes of a diameter lesser than the bore are placed inside each pin where they are expanded at the ends so as to fit tightly. Thus an annular space is left between the inside of the hollow shaft and the outside of the tubes, which construction is used partly to provide oil leads. The oil finds its way by the space in the main crankshaft through the bearings, and holes are drilled up the webs to form a connection with the spaces in the crank pins. Phosphor bronze shells with white metal linings are used for the bearings throughout, and the forward main bearing is 7 ins. long.

As may be observed in the general arrangement in Fig. I., there is provision for a double ball thrust, the annular ring forming the central portion of the races being attached to the shaft by fitting on an increased diameter of the shaft adjacent to a collar. On the other side a groove is cut round the shaft and a split ring sprung into position to form the other abutment.

The cylinders are not staggered on the crank case, but are set opposite one another, while the difficulties of connecting big end bearings and clearances are overcome as shown in Figs. I. and II. The illustration of the connecting rod in the centre of Fig. II. shows that a master connecting rod and big end bearing is used for one cylinder, and the upper half of this big end bearing is provided with two lugs through which is passed a second short hollow shaft which forms an auxiliary bearing for the second connecting rod. The cap of the main big end bearing is also provided with two lugs which fit into the first-mentioned lugs, so forming a hinge, the secondary shaft of which is the pin. Thus it is necessary to use one big end bolt to secure the whole. Since the photograph from which Fig. II. was made was taken, a new pattern connecting rod has been adopted, which is similar in principle, but has a flat section, and is drilled through the central web for lightness.

There are some interesting points to be found in the piston, the attachment of the gudgeon pin being somewhat unusual for this type of engine. On the piston head there is a ring into which is screwed a second steel ring carrying the gudgeon pin. The latter is hollow, and is locked in place by means of a wire ring which can be seen on the lower piston in the side elevation of the general arrangement. The chief reasons advanced as explaining this attachment are that the piston shell can be machined all over without trouble, while the lugs carrying the gudgeon pin can be easily machined likewise. Cast iron is used for the piston, and the usual piston ring slot is turned on the piston near the head, but it is of somewhat larger size than usual. Into this fits a split phosphor bronze ring of channel section, and holes are drilled through the piston head to connect directly with the inside of the channel. The upper flange of the piston ring is formed so that these holes are left clear, and during the compression or the expansion strokes the gases force their way through the holes and hold the rings against the walls of the cylinder, while, at the same time, the upper and lower flanges of the rings are squeezed against the sides of the slot.

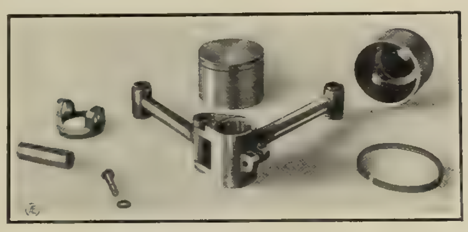


Fig. II.

The cylinders are of steel, and are turned from the solid, the general design being shown in Fig. I. The diameter is increased at the combustion chamber to permit the use of the large valves which are set in the head. Looking at the central cylinder in Fig. III., it will be seen that two tubular projections at the top receive the valve guides, and two further projections accommodate the sparking plugs. A copper water jacket is placed over the top of the cylinder so that the screwed end of the various lugs just project through corresponding orifices, and a water-tight joint is then made by means of washers and screwed rings, while the lower end of the jacket fits on a cup-shaped flange solid with the cylinder. This is made water-tight by means of a steel ring shrunk on, and the lower water connection is made directly into the steel flange supporting the jacket. A special bracket is provided and fixed to the top of the steel combustion head to take the flange of the outlet, and this bracket takes any stress on the water pipe, the copper jacket merely being pinched to it by the unions. The lower ends of the cylinders register into the top of the crank case, and are held in place by bolts.

The valves are interchangeable, and are 2 5/16 ins. in diameter, the cage being made of cast iron, into which a steel guide is screwed. The guide is increased in diameter at the top of the flat section and the helical springs have a seating inside. On the outside of this increased diameter a thread is provided to accommodate a ring which holds the inlet or exhaust pipes into position. The steel overhead rocking arms are given their movement from cams by tubular rods, with ends spiggoted into steel tappet rods, which work in guides of a special shape to prevent them from turning, and are attached to the crank case in pairs by light dogs. The camshaft is a hollow steel shaft on which the cams are mounted in sets of four, with a bearing between each set, and Fig. IV. shows the setting of the valves.

The cast aluminium crank case is made in two halves, the bottom half being merely a cover. Details can be seen clearly in Fig. I. The timing gear itself calls for no particular notice, but the centrifugal water pump is driven direct from a timing wheel, while the magneto is driven from an intermediate wheel. A square end of the intermediate shaft projects from the crank case and over this fits a square tube, while a Vernier adjustment is attached to the magneto, squared also to fit into the other end of the square tube. Thus, as the magneto is held to its bracket by means of a clip and two wing nuts, by

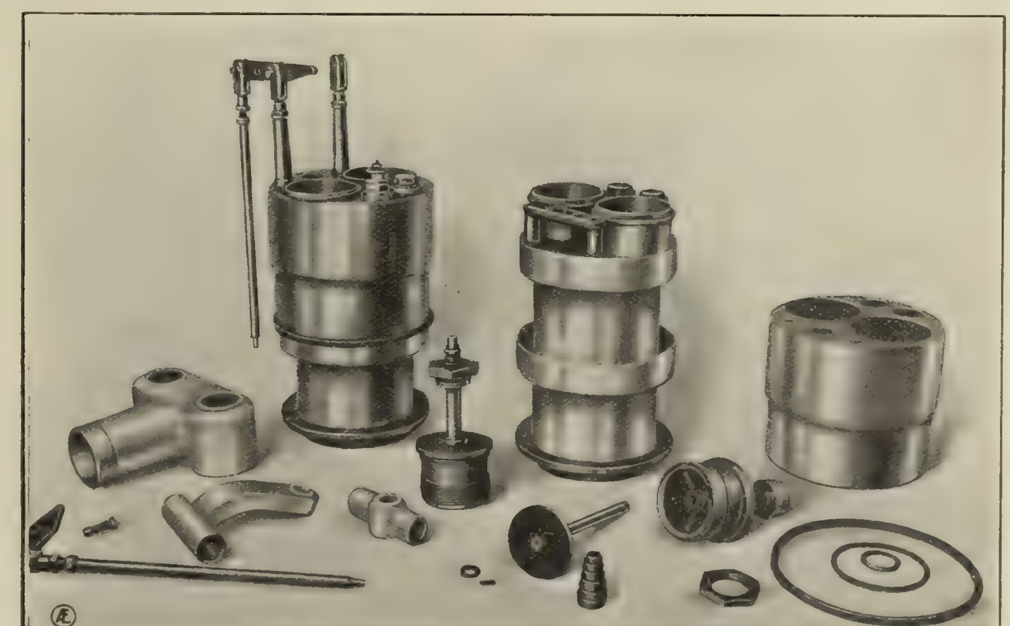


Fig. III.

THE ALL BRITISH AERONAUTICAL 80 H.P. ENGINE.

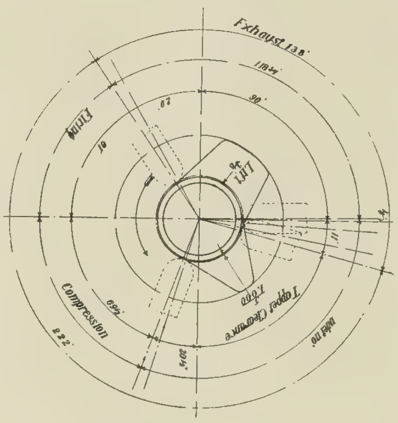


Fig. IV.

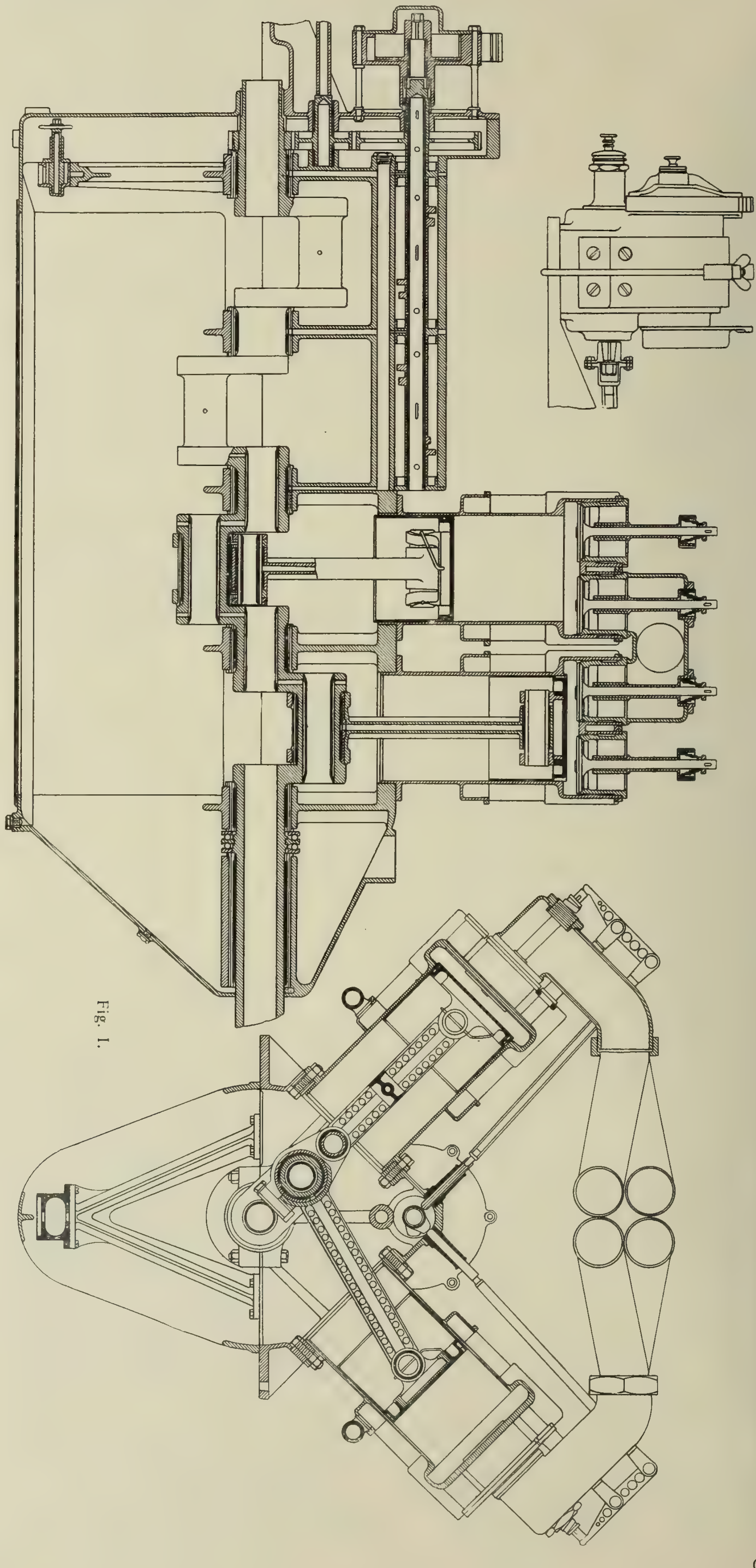


Fig. I.

undoing these it can be removed, and, owing to the arrangement of the square tube, the magneto can be taken off and put on without in any way upsetting its adjustment, as small pegs are provided on the two squares to ensure that the square tube comes back into exact position. The bottom half of the crankcase is made with two cast ends and a sheet aluminium centre, and is provided with a very large sump containing some four gallons of lubricating oil.

The lubrication system of the engine appears to be good, the outstanding feature being that a single copper pipe is used, all the oil leads being drilled through the various parts. The circulation of the oil is effected by a small gear pump situated at the bottom of the sump and driven at an increased speed by a chain from the crankshaft. It delivers about two gallons of oil per minute, and creates a pressure of 50 lbs. per square inch. Leads are drilled up the walls of the crankcase until they reach another lead drilled through from end to end of the engine, parallel to and just below the two-to-one shaft. It will be seen from the side elevation and general arrangement how subsidiary leads are drilled through to the various main bearings, and also the camshaft bearings. From the main crankshaft bearings the oil finds its way to the big ends, through the hollow crank pins, and then by the connecting rods to the gudgeon pins, while the timing gear itself is kept flooded with oil by the chain driven pump.

Perhaps the most interesting point in the design of this ingenious engine is the fact that the air intake for the carburettor passes right through the crank case, the outlet being situated at the forward end, and the inlet being through the hollow portion of the rear end of the crankshaft. It is intended that the cool air should pass right through the interior of all the bearings and thereby cool the lubricating oil as it is being used. At the same time a spray of lubricating oil is easily carried right through the carburettor into the combustion chambers, and so on to the walls of the cylinders, thereby providing extra lubrication for the pistons.

CORRESPONDENCE.

No responsibility is accepted by the Editor for the opinions of correspondents, and the right is reserved to omit any portion of a letter if it is considered desirable to do so. Letters to be inserted with a nom de plume must be accompanied by the name and address of the writer. If publication in the next issue is desired letters should reach the Editor in no case later than the 20th of the preceding month.

BODY BUILDING MATERIALS.

Sir,—Recently several correspondents have addressed you on the subject of wood in the construction of automobile frames and bodies.

Very little appears to be known by motor manufacturers, and even coach builders, of the most suitable materials for body work, with which this letter is chiefly concerned. It is my purpose to show that sheet steel, properly employed, fulfills all the needs of those who desire, with "Occidens," lightness, cheapness, and durability.

I may preface my remarks by observing that practically all motor carriage builders use four qualities of mild Siemens-Martin sheet steel, varying in thickness from 18 B.G. (.05 in.) to 24 B.G. (.025 in). These are, in descending order of cost:—

(1) *Tinned Sheets.*

Steel sheets coated with pure tin, and having a very smooth surface.

Used chiefly for the best class of body work, especially suitable for petrol tanks, as the distillates of petroleum have no action on tin.

(2) *Terne Sheets.*

Steel sheets coated with a mixture of tin and lead.

Very largely used for bodies, bonnets, mudguards, etc.

The surface is not quite so smooth as that obtainable on the tinned sheets.

(3) *Leadcoated Sheets.*

Steel sheets coated with pure lead. Perhaps the most widely used of all. Tinned and Terne sheets have a smoother surface than the leadcoated, but the last-named takes the paint very well.

(4) *Black Sheets.*

These are mild steel sheets uncoated with any protecting and non-oxidising metal. They are commonly employed either P.CR/CA (Pickled, cold rolled and close annealed) or CR/CA (Cold rolled and close annealed). The former are pickled to remove the scale or iron oxide which forms a thin skin on the sheets when they are rolled red-hot in the mill, thus securing a sound foundation for the paint. They possess the defect of rusting very quickly unless painted on both sides. The CR/CA sheets are usually employed in less exposed places, where appearances do not so much matter; as, for instance, for under shields and silencers. The scale is liable to work off in a short time, but this is of little consequence in the positions indicated.

By a special process this scale can be polished and more firmly affixed to the steel base. Steel sheets treated thus are known as "Planished." They are often used for bonnets and mudguards and afterwards painted, but I do not consider them as satisfactory as the coated sheets, although they cost more than leadcoated steel. The protecting skin is liable to crack and peel off, especially when the steel is bent.

Aluminium panels have the merit of lightness, and fair durability, but the tensile strength is inferior to that of steel; and even when painted this metal has a tendency to disintegrate into a powdery substance, especially when exposed to the sea air. On the other hand, tin, terne, or lead coated steel is practically rustless, and need only be painted on the exposed side.

In order that steel sheets used for carriage work, whether coated with a protecting metal or not, may stand hollowing to the curve of the body under hammer or press, it is necessary for the steel to be annealed most carefully. But if only a right-angled bend or simple curve is necessary, as in the case of most forms of mudguards or bonnets, and some kinds of bodies, the steel can easily be left stiffer, thus allowing a thinner gauge to be used.

For "Occidens'" cheap and light two-seater I believe there is nothing more suitable than P.CR/CA or CR/CA steel with a coat or two of paint. Two sheets 72 in. x 30 in. x 26 B.G. (.02 in.) or 28 B.G. (.015 in.) unannealed to render them hard, and used only for simple bends, should be sufficient to form the body and scuttle dash. Their combined weight, exclusive of their wooden framework, would be about twenty pounds; not a very ponderous matter even on a car weighing only 1,120 lbs. The cost of these twenty pounds of uncoated sheet steel would be about 2s. 6d. or 3s. not a very expensive item, even on a car costing little more than £100.

In view of these facts it can be asserted with some show of confidence that sheet steel fulfills the three conditions of lightness, durability, and cheapness. I am afraid the defects of a canvas covered body would outweigh its merits. As an oarsman I recollect how our matchboarded and canvas-covered racing boat had to be handled with the care of a new-born babe. A shock which would not put a tyre out of action would doubtless cause a canvas framework to collapse. Sheet steel will dent under a blow of a certain intensity, but it can easily be straightened again.

A. S. H. DORE.

THE INSTITUTION OF AUTOMOBILE ENGINEERS.

Sir,—That your remarks and strictures upon the Institution of Automobile Engineers are well meant and to the point, there is no doubt the majority of the membership will agree, and whilst one may not be in accordance with all your remarks, there are some on the question of general management which are well worth discussion in your columns.

Personally, I feel that you are very largely correct in your strictures as to the constitution of the Council; in fact, there are members on it who do not even profess to be automobile engineers, if the last ballot sheet for the Council be any criterion, and *apropos* of this there appears reasonable ground for the opinion which is prevalent in some quarters, that the Council is a mutual admiration society, from the fact that the retiring members at the last election of Council were proposed for re-election with, I believe, one exception, by two members of Council only.

The degree of success likely to result from a procedure such as this is, of course, dependent upon the lack of interest—a common thing, I presume, in most societies—which the majority of the membership take in an election, and the thanks of all who have the success of the institution at heart are due to you for pointing out the necessity for live and practical men.

There is no doubt a tendency for some discussions to degenerate into wearisome platitudes of an after-dinner kind, which are irritating to author and audience alike, and it is worth consideration whether, with a young institution such as ours, the practice of calling speakers to the dais if dropped would not mitigate this. There are plenty of good men who could contribute towards an interesting discussion, but whose powers of delivery are not proof against what is to them an ordeal.

The increase in membership is, of course, subject for congratulation, but there are many who wonder by what method candidates are elected, when they find good men accepted as associate members, whilst others whom they know are not so good are accepted as full members, and occasionally candidates are accepted who do not even appear to "fill the bill."

A subject of importance which has recently been discussed by other learned institutions, namely the training of engineers, I find from reference to a semi-technical weekly, appears to have been settled off hand by a committee appointed by the Council. That this matter, particularly in relation to automobile engineering, is one worthy of good consideration and discussion, will be admitted. There is certainly a difference of opinion as to what constitutes the proper training for an automobile engineer, as may be gathered from the experience of a friend of the writer, who, on applying for an appointment for which he considered himself particularly suitable to a gentleman who is a member of Council, found that one of his disqualifications appeared to be that he had spent some time gaining experience as a draughtsman.

MEMBER.

CATALOGUES RECEIVED.

CHAINS.—Hans Renold, Ltd., have recently issued a further catalogue dealing with the firm's output of the well-known silent and roller chains. The catalogue is in the form of a booklet, and begins with the reasons which weighed with the firm when deciding what alterations should be made in the latest type of chain. There follows a description of the type of chain which is used by the firm, with a number of excellent half-tones illustrating the constructional details of the wares in question, and explaining some questions of policy in connection with the fitments now adopted. Illustrations then show characteristic examples of each particular type, with a view to proving the governing principles underlying the design.

THE CATALOGUE of the Lynton Wheel and Tyre Syndicate, of Warrington, is interesting, because the device advertised therein is one of the few of this nature which have remained for any length of time on the market. The forward part of the booklet is occupied by a comparison between ideal and real methods, interspersed with photographs of the various types of car fitted with Lynton wheels, and showing one or two examples of the wheel in pieces. The principles of the whole device are clearly set forth with sectional illustrative drawings, and care is taken to make this understandable even by people whose technical knowledge may be excessively small.

CLUTCH BRAKE DESIGNS.

A description of some types in successful employment at the present time.

UNTIL a few years ago it was almost impossible to discover a car which had a clutch stop permanently fitted as part of its standard design. One or two makers it is true, were, at an early date, using some type of stop, and have continued to use practically the same design from that time forwards. Now, however, practically all makers are using some form of brake to prevent the spinning of the clutch shaft and to facilitate speed changing.

Nearly every other refinement on the modern car has had considerable thought and attention from the designing staffs, but in many cases one is forced into the belief that a clutch stop has been fitted rather to satisfy the demand of the moment than as an engineering job necessary for the car's well-being. Thus cars with clutches of large diameter are found almost without a clutch stop, while others with a light clutch of small diameter are sometimes fitted with a form of stop which most readily adapts to itself the duties usually allocated to a car brake.

It is not suggested that the designer is faced with an extremely simple or easy task when planning this small accessory; on the contrary, although a small matter in size, it is an extremely hard problem which will take great thought and inter-discussion before the solution can be expected to present itself.

Accurately to overcome the difficulties, a clutch brake must be so designed that it will slow down the clutch shaft when changing to a higher gear, but must in no wise effect the same shaft when the change down is operating, while in addition, such a design must not consist of more moving parts than are absolutely necessary and, above all, those portions which form the actual frictional surfaces must be renewable in a cheap and easy manner even by an unskilled man. With the older type of gear striking mechanism working in a single slotted quadrant, such a device as set forth above was simpler, since the change speed lever moved in one direction for a higher gear, and another when used for the lower speeds, and an attachment could be used which might operate the clutch brake in accordance with the direction of lever movement. With the almost universal adoption of the H quadrant the movement of the change speed lever varies to such an extent that the problem is greatly complicated, despite the correlative reduction in clutch diameter.

So far the only solution which has been attempted involves careful manipulation of the clutch pedal, which is so arranged that full forward movement operates the clutch brake, while a reduced movement withdraws the clutch only. This design is exposed to the troubles which eventually overcame a similar device used to control the foot brake and clutch from the same pedal, namely, that few owners or drivers would take the trouble to handle the arrangement in a gentle manner and that none of them were prepared to keep such a combination adjusted so that its automatic action might be ensured. Anyone inspecting a quantity of clutch brakes in place on their respective chassis at the present day, would at once notice that quite a considerable number could be made extremely formidable car brakes, and are likely to be used as such even by a car owner of experience. Moreover, the diameter of these brakes is extremely striking, and varies apparently with the designer rather than with the clutch for which he intends it. It can be said, however, that the majority fully realize that accessibility and ease of replacement are points which add both to the salesman's arguments and to the customer's happiness.

In the main, there would seem to be three classes or designs under which the various clutch brakes can be classified. There is, firstly that design involving the use of coned, or double-coned, surfaces engaging exactly as the ordinary leather clutch, but usually having fibre on steel working surfaces.

Then follows that type in which a plain disc, usually of large diameter, is brought up against a stationary pad sometimes provided with recoil springs to magnify its soft engagement; and finally there is a class in which the operation of the clutch pedal, acting through certain suitably placed levers, moves a pad downwards on to a revolving disc bolted to the clutch.

Of course, such a classification must necessarily be somewhat broad, since there are no two clutch brakes of which the details are in any way the same, but they can roughly be formed into such classes. As examples of the first, or cone clutch principle, Fig. I., that supplied with the Austin cars, is shown. In this design the backward movement of the clutch-actuating fork brings a double cone clutch into action, the frictional surfaces being cast iron. As a whole it is an arrangement which suffers from complication, and is not as easy to remove as it should be, while, although good in the hands of a demonstrator, it might be somewhat too fierce for the average driver, whose actions are unfortunately seldom gentle.

By far the largest class can be placed under the second head, Figs. II. and III., presenting the Maudslay and Napier, both of which are plain discs of fibre. The latter is a particularly accessible design, as the removal of one split disc allows easy access to the other, and the soft copper rivets which attach the fibre are easy both to handle and to obtain. The Maudslay is more neat, but hardly quite as accessible, while both are gentle in action and not easy to mishandle.

Fig. IV. is an example of a clutch brake in which the pad has a recoil spring to soften the action, a feature shared with the well-known Daimler clutch brake. In this case a much softer action is obtained, although accessibility is affected rather adversely, when the renewal of the leather-covered disc becomes necessary. The clutch is considerably lighter and smaller than that used on the Daimler, for the latter, operating at a considerable radius from the clutch shaft centre, is rather harder to actuate softly, although it is considerably easier to replace, and about equally easy to adjust. As in the Straker Squire (which was described recently), the locknuts shown are perfectly easy to unscrew.

The Adams design, Fig. V., comes under the same classification, although differing in detail. The clutch brake shown is remarkably easy to replace or reline, as it can be removed from the car without in any degree affecting its complementary or surrounding parts, when the nut shown on its bracket, together with another situated immediately below it, is undone. In this case a forward movement of the ball-bearing striking fork disengages the clutch and engages the clutch brake, a fibre lining rivetted to the fork coming into contact with the steel flange. Beyond its accessibility such an arrangement has not too great a braking effect, and would seem to satisfy all conditions.

Fig. VI. illustrates the type relegated to the third class, namely, those actuated by the striking mechanism which controls the clutch, and is that fitted to the Arrol Johnson. Rearwards movement of the clutch fork brings the half-moon shaped fibre pad against a fibre faced disc secured to a split flange on the clutch shaft. Rivets are employed to secure each of the fibre portions to their steel flanges, and accessibility is increased both by bolting the half-moon piece to the clutch actuating levers, and by the aforementioned split coupling. As in Fig. V. this type should give a sufficient braking effort without allowing violent action. The sketches, Figs. VII., VIII., and IX. all show examples of this particular type, belonging respectively to the National, Thames and Valveless cars. The latter is probably the simplest, and offers smooth action combined with almost unique accessibility. Fig. VIII. might conceivably become rather too powerful, and is certainly somewhat on the complicated side of a difficult problem. Fig. VII. should have about the correct stopping power, and is slightly less complicated, although it might prove more expensive than the simple flat spring and fibre pad used in the arrangement illustrated in Fig. IX.

On heavy vehicles the clutch brake is frequently a miniature pattern of the ordinary car brake; two patterns of this type are set out in Figs. X. and XI., Fig. X. being contracted on the application of the clutch pedal, and Fig. XI. spring-controlled immediately the cam is released on the downward movement of the clutch pedal. Both are used with large diameter heavy clutches, for which they are efficient, if distinctly complicated and expensive, while both can be removed without the disturbance of the clutch.

It is not too much to hope that there will arise presently a suitable solution to this hard, if minor, problem. The annual general improvement in cars is largely obtained by attention to petty parts, and of these the clutch brake is one of the most important.

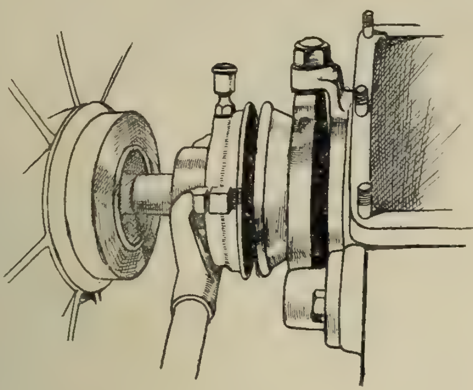


Fig. I.

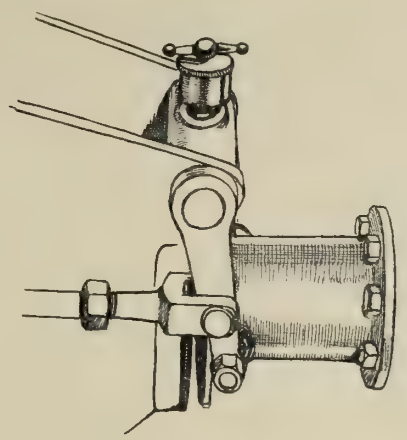


Fig. II.

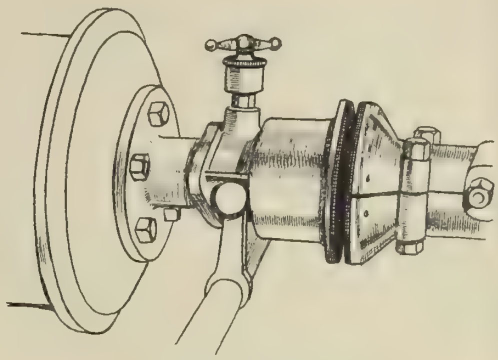


Fig. III.

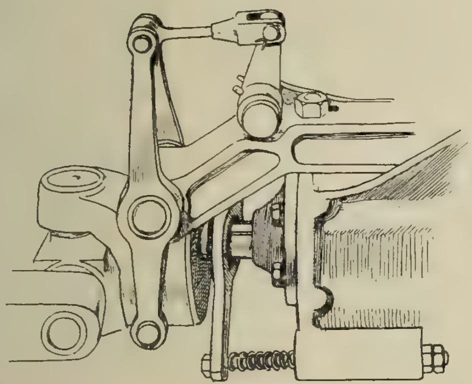


Fig. IV.

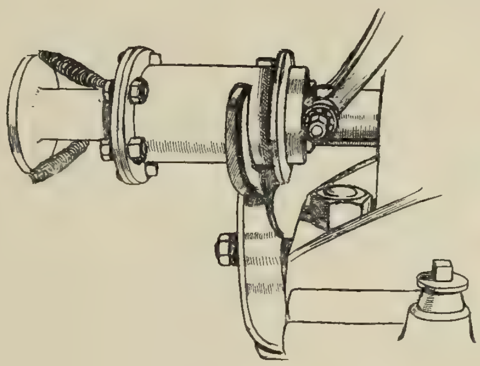


Fig. V.

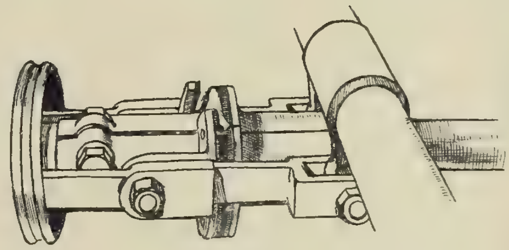


Fig. VI.

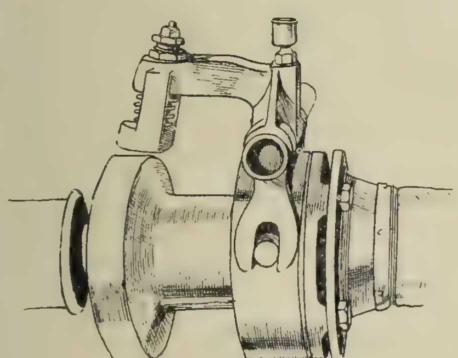


Fig. VII.

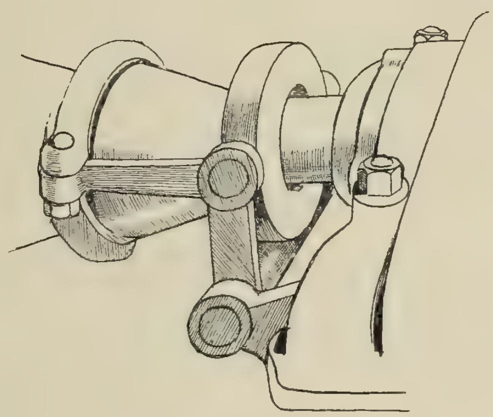


Fig. VIII.

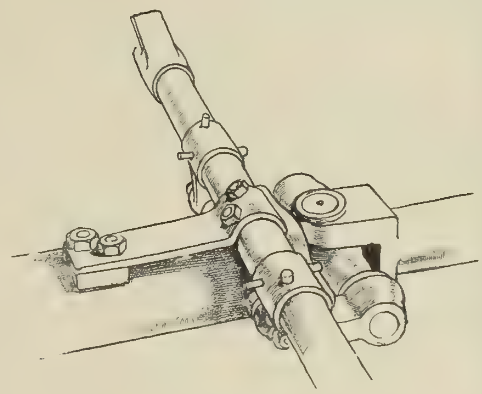


Fig. IX.

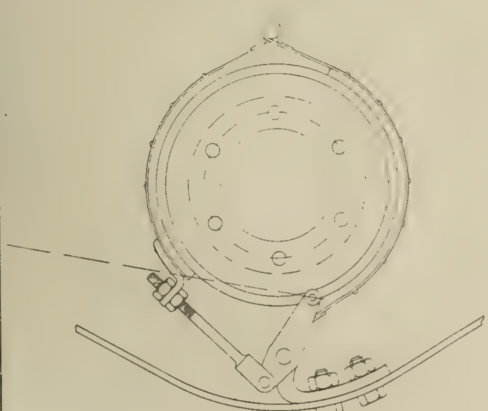


Fig. X.

TYPICAL CLUTCH BRAKE DESIGNS.

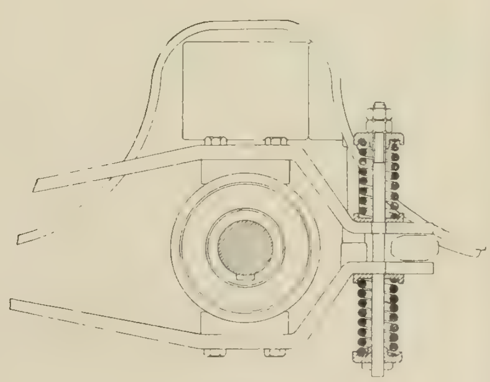


Fig. XI.

THE MATERIALS OF AUTOMOBILE CONSTRUCTION.

A collection of material specifications in actual use by various manufacturers, together with some suggested specifications. *

Specifications for Steel.

These steels may be of open hearth, crucible or electric manufacture, and must be sound and free from physical defects, such as seams, heavy scale or scabs and surface defects.

These steels will be purchased on the basis of chemical analysis. The specifications indicate the desired chemical composition. Any shipments not conforming to these specifications after careful check analysis may be rejected.

SPECIFICATION No. 1.

.15 Carbon Steel.

Carbon	.08% to .18%	(.15% desired)
Manganese	.40% to .60%	(.50% desired)
Silicon, not over	.20%	
Phosphorus, not over...	.04%	
Sulphur, not over	.04%	

SPECIFICATION No. 2.

.20 Carbon Steel.

Carbon	.15% to .25%	(.20% desired)
Manganese	.50% to .80%	(.65% desired)
Silicon, not over	.20%	
Phosphorus, not over	.04%	
Sulphur, not over	.04%	

SPECIFICATION No. 3.

.30 Carbon Steel.

Carbon	.25% to .35%	(.30% desired)
Manganese	.50% to .80%	(.65% desired)
Silicon, not over	.20%	
Phosphorus, not over	.04%	
Sulphur, not over	.04%	

SPECIFICATION No. 4.

.45 Carbon Steel.

Carbon	.40% to .50%	(.45% desired)
Manganese	.50% to .80%	(.65% desired)
Silicon, not over	.20%	
Phosphorus, not over	.04%	
Sulphur, not over	.04%	

SPECIFICATION No. 5.

.80 Carbon Steel.

Carbon	.75% to .90%	(.80% desired)
Manganese	.25% to .50%	(.35% desired)
Silicon	.10% to .30%	
Phosphorus, not over	.035%	
Sulphur, not over	.035%	

(Primarily for Springs.)

SPECIFICATION No. 6.

.95 Carbon Steel.

Carbon	.90% to 1.05%	(.95% desired)
Manganese	.25% to .50%	(.35% desired)
Silicon	.10% to .30%	
Phosphorus, not over	.035%	
Sulphur, not over	.035%	

(Primarily for Springs.)

SPECIFICATION No. 7.

20 Carbon, 3½ Per Cent. Nickel Steel.

Carbon	.15% to .25%	(.20% desired)
Manganese	.50% to .80%	(.65% desired)
Silicon, not over	.20%	
Phosphorus, not over	.04%	
Sulphur, not over	.04%	
Nickel	3.25% to 3.75%	(3.50% desired)

(Primarily for Case-Hardening.)

SPECIFICATION No 8.

.30 Carbon, 3½ Per Cent. Nickel Steel.

Carbon	.25% to .35%	(.30% desired)
Manganese	.50% to .80%	(.65% desired)
Silicon, not over	.20%	
Phosphorus, not over	.04%	
Sulphur, not over	.04%	
Nickel	3.25% to 3.75%	(3.50% desired)

(Primarily for Heat Treatment.)

SPECIFICATION No. 9.

.15 Carbon, Chrome Nickel Steel.

Carbon	.10% to .20%	(.15% desired)
Manganese	.40% to .60%	(.50% desired)
Silicon	.10% to .30%	
Phosphorus, not over	.04%	
Sulphur, not over	.04%	
Nickel	1.50% to 2.00%	(1.75% desired)
Chromium	.50% to 1.00%	(.75% desired)

(Primarily for Frames.)

SPECIFICATION No. 10.

.20 Carbon, Chrome Nickel Steel.

Carbon	.15% to .25%	(.20% desired)
Manganese	.30% to .50%	(.40% desired)
Silicon	.10% to .30%	
Phosphorus, not over	.04%	
Sulphur, not over	.04%	
Nickel	3.25% to 3.75%	(3.50% desired)
Chromium	1.25% to 1.75%	(1.50% desired)

SPECIFICATION No. 11.

.30 Carbon, Chrome Nickel Steel.

Carbon	.25% to .35%	(.30% desired)
Manganese	.30% to .50%	(.40% desired)
Silicon	.10% to .30%	
Phosphorus, not over	.04%	
Sulphur, not over	.04%	
Nickel	3.25% to 3.75%	(3.50% desired)
Chromium	1.25% to 1.75%	(1.50% desired)

SPECIFICATION No. 12.

.45 Carbon, Chrome Nickel Steel.

Carbon	.40% to .50%	(.45% desired)
Manganese	.40% to .60%	(.50% desired)
Silicon	.10% to .30%	
Phosphorus, not over	.04%	
Sulphur, not over	.04%	
Nickel	1.50% to 2.00%	(1.75% desired)
Chromium	.80% to 1.20%	(1.00% desired)

SPECIFICATION No. 13.

.20 Carbon, Chrome Vanadium Steel.

Carbon	.15% to .25%	(.20% desired)
Manganese	.40% to .70%	(.50% desired)
Silicon	.10% to .30%	
Phosphorus, not over	.04%	
Sulphur, not over	.04%	
Chromium	.80% to 1.10%	(.90% desired)
Vanadium, not less than	.10%	(.18% desired)

(Primarily for Case-Hardening.)

SPECIFICATION No. 14.

.30 Carbon, Chrome Vanadium Steel.

Carbon	.25% to .35%	(.30% desired)
Manganese	.40% to .70%	(.50% desired)
Silicon	.10% to .30%	
Phosphorus, not over	.04%	
Sulphur, not over	.04%	
Chromium	.80% to 1.10%	(.90% desired)
Vanadium, not less than	.10%	(.18% desired)

(Primarily for Heat Treatment.)

SPECIFICATION No. 15.

.45 Carbon, Chrome Vanadium Steel.

Carbon	.40% to .50%	(.45% desired)
Manganese	.60% to .90%	(.75% desired)
Silicon	.10% to .30%	
Phosphorus, not over	.035%	
Sulphur, not over	.035%	
Chromium	1.00% to 1.30%	(1.20% desired)
Vanadium, not less than	.10%	(.18% desired)

SPECIFICATION No. 16.

Silico-Manganese Steel.

Carbon	.45% to .55%	(.50% desired)
Manganese	.60% to .80%	(.70% desired)
Silicon	1.90% to 2.20%	(2.00% desired)
Phosphorus, not over	.04%	
Sulphur, not over	.04%	

Note.

The foregoing specifications (1 16) are drawn to meet ordinary manufacturing conditions. Steel of greater purity can be obtained, that is, with lower limits of phosphorus and sulphur, say .025%. Also narrower limits of carbon, manganese and other elements can be obtained under special terms.

SPECIFICATION No. 17.

Common Screw Stock.

The following composition covers two types of screw stock :

Carbon	.08% to .25%
Manganese	.30% to .80%
Phosphorus, not over	.16%
Sulphur	.05% to .15%

SPECIFICATION No. 18.

Low Alloy Steel.

There has lately come on the market a type of steel which sells at a slight advance over the price of open hearth and which is superior to it in a great many properties. These steels are of any carbon desired, and of the following approximate composition :

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* Report of the Standards Committee of the Society of Automobile Engineers.

SPECIFICATION No. 18.

Carbon	As desired
Manganese	.50% to .80%
Silicon, not over	.20%
Phosphorus, not over	.04%
Sulphur, not over	.04%
Nickel	.90% to 1.20%
Chromium	.15% to .40%

Valve Metals.

SPECIFICATION No. 19.

Valve Metal No. 1.

This metal shall contain not less than 96% of nickel.
This material must be malleable.

SPECIFICATION No. 20.

Valve Metal No. 2.

This metal shall contain:

Carbon, not over	.50%
Manganese, not over	1.50%
Phosphorus, not over	.04%
Sulphur, not over	.06%
Nickel	28.00% to 35.00%

The remainder to be iron.

Steel Castings.

SPECIFICATION No. 21.

The following composition is desired:

Carbon	.30% to .40%	(.35% desired)
Manganese	.60% to .80%	(.70% desired)
Silicon	.10% to .30%	
Phosphorus, not over	.06%	
Sulphur, not over	.06%	

Gray Iron Castings.

SPECIFICATION No. 22.

The following composition is desired:

Total Carbon	3.25% to 3.50%	
Manganese	.40% to .70%	(.50% desired)
Silicon	1.90% to 2.20%	(2.00% desired)
Phosphorus	.60% to 1.00%	
Sulphur, not to exceed	.10%	

Malleable Iron.

SPECIFICATION No. 23.

The following composition is desired:

Manganese	.30% to .70%	(.50% desired)
Silicon not over	1.00%	(.60% desired)
Phosphorus, not over	.20%	(.17% desired)
Sulphur, not over	.06%	

Steels.

The materials specified in detail hereinbefore include those most important to the builder of automobiles. It is obvious that there are more kinds of material specified than are likely to be used by any one manufacturer. The number given presents a choice sufficient to cover many designs. A few grades of steel properly selected and properly treated are enough to put into the parts of any given car.

SPECIFICATION No. 1.

.15 Carbon Steel.

The source of this material is always the basic open hearth furnace. It is a material commonly used for seamless tubing, pressed steel frames, pressed steel brake-drums, sheet steel brake-bands and pressed steel parts of many varieties. It is soft and ductile and will stand much deformation without cracking.

This steel is weak, and in an annealed condition will have an elastic limit of about 30,000 lbs. per square inch, with very high reduction of area and elongation.

In the cold drawn or cold rolled condition the elastic limit may be as high as 60,000 lbs. per square inch, with but little reduction of area and elongation. This relatively high elastic limit can be obtained only in light or small sections, either sheet or rod form.

This material should not be used for such parts as are to be machined. It will tear badly in the turning, threading and broaching operations.

Heat treatment produces but little benefit and that not in strength but in toughness. It is possible to quench this grade of steel and put it in a condition to machine a little better than in the annealed state; but this is an expedient that should be used only in an emergency and not as a regular practice.

Forgings may be and are made of this steel, but it is not desirable material for forgings for the above stated reasons.

The heat treatment which will produce a little stiffness is to quench at 1500° F. in oil or water. No drawing is required.

This steel will case-harden but should not be chosen for that purpose.

SPECIFICATION No. 2.

.20 Carbon Steel.

The natural source of supply for this material is the basic open hearth process.

This quality of steel is intended primarily for case-hardening purposes. It forges well and machines well. It may therefore be used for a very large variety of forged, machined and case-hardened parts of an automobile where strength is not paramount.

This steel is a soft, ductile material having an elastic limit in the annealed condition of about 35,000 lbs. per square inch.

Steel of this quality may be also drawn into tubes and rolled into cold rolled forms, and, as a matter of fact, makes a better frame because of the slightly higher carbon and resulting stiffness. The increased carbon has no detrimental effect as far as usage is concerned, and it is only the most difficult cold forming operations that cause it to crack during the forming. For automobile parts it may be safely used interchangeably with Specification No. 1 as far as cold pressed shapes are concerned.

The elastic limit in a cold rolled or cold drawn condition is about 70,000 lbs. per square inch; with the same degree of working as will give 60,000 lbs. per square inch with Specification No. 1.

Heat treatment of this steel produces but little change as far as strength is concerned, but does cause a desirable refinement of grain after forging, and the toughness is materially increased. A simple quenching operation from about 1500° F. in oil is all that is necessary. The treatment will often help the machining qualities.

The case-hardening treatment is the most important for this quality of steel. The character of the treatment must depend upon the importance of the part to be treated and upon the shape and size. There is a certain group of parts in an automobile which are not called upon to carry much load or withstand any shock. The only requirement is hardness. Such parts are fairly illustrated by screws and by rod end pins. The simplest form of case-hardening will suffice, viz:

Heat Treatment No. 1.

After forging or machining—

1. Carbonize at a temperature between 1600° F. and 1750° F. (1650°-1700° F. desired).
2. Cool slowly or quench.
3. Re-heat to 1450°-1500° F. and quench.

Operation 2 may be either a slow cooling or quenching, depending entirely upon the convenience and shape of the parts. If the parts are a large number of small pieces, it is very convenient to cool them quickly by quenching at a temperature somewhat below the case-hardening temperature. The subsequent heating operation (No. 3) refines the grain of the outside, carbonized steel and produces good enough results for the class of parts in question.

Another class of parts demands the best treatment, such as gears, steering-wheel pivot-pins, cam-rollers, push-rods and many similar details of an automobile which the manufacturer learns by experience must be not only hard on the exterior surface but possess strength as well. The desired treatment is one which first refines and strengthens the interior and uncarbonized metal. This is then followed by a treatment which refines the exterior, carbonized, or high carbon, metal.

Heat Treatment No. 2.

After forging or machining—

1. Carbonize at temperature between 1600° F. and 1750° F. (1650°-1700° F. desired).
2. Cool slowly in the carbonizing mixture.
3. Re-heat to 1500°-1550° F.
4. Quench.
5. Re-heat to 1400°-1450° F.
6. Quench.
7. Draw in hot oil at a temperature which may vary from 300°-450° F., depending upon the degree of hardness desired.

In the case of very important parts, the last drawing operation should be continued from one to three hours, to insure the full benefit of the operation.

The objects of drawing are two-fold: First, and not least important, is the relieving of all internal strains produced by quenching; second, a decrease in hardness, which is sometimes desirable. The hardness begins to decrease very materially from 350° F. up, and the operation must be controlled as dictated by experience with any given part.

There are certain very important pieces that demand all of these operations, but the last drawing operation may be omitted with a large number. Experience teaches what degree of hardness and toughness combined is necessary for any given part. It is impossible to lay down a general rule covering all different uses. If the fundamental principle is well understood, there should be no trouble in developing the treatment to a proper degree.

Following the foregoing treatment a fractured part should show a fine grain on the exterior, without any appearance of shiny crystals. The smaller the crystals the better. The interior may show a silky, fibrous condition or a fine crystalline condition; but it must not show a coarse, shiny, crystalline condition.

SPECIFICATION No. 3.

.30 Carbon Steel.

The natural source of supply for this material is the basic open hearth.

This steel is primarily for use as a structural steel. It forges well, machines well and responds to heat treatment in the matter of strength as well as toughness; that is to say, intelligent heat treatment will produce marked increase in the elastic limit. It may be used for all forgings such as axles, driving-shafts, steering pivots and other structural parts. It is the best all-round structural steel for such use as its strength warrants.

In an annealed condition the elastic limit is about 45,000 lbs. per square inch, with ample elongation and reduction of area.

Heat treatment for toughening and strength is of importance with this steel. The heat treatment must be modified in accordance with the experience of the individual user, to suit the size of the part treated and the combination of strength and toughness desired. The steel should be heat treated in all cases where reliability is important.

Machining may precede the following heat treatment, depending somewhat upon convenience and the character of the treatment. If the highest strength is demanded, then strong quenching methods must be employed; for example, brine. In such case, the elastic limit will be correspondingly high and the steel correspondingly hard and difficult to machine. On the other hand, if a moderately high elastic limit is all that is desired, an oil quench will suffice and machining may follow without any difficulty whatever.

Heat Treatment No. 3.

After forging or machining—

1. Heat to 1500° F.
2. Quench.
3. Re-heat to 600°-1300° F. and cool slowly.

This is the simplest form of heat treatment. The drawing operation (No. 3) must be varied to suit each individual case. If great toughness and little increased strength are desired, the higher drawing temperatures may be used, that is in the neighbourhood of 1100° F., 1200° F. or 1300° F. If much strength is desired and little toughness, the lower temperatures are available. Even the lowest of the temperatures given will produce a quality of steel, after oil quenching, that is very tough—suf-

ficiently tough for many uses. In fact, with some parts the drawing operation (No. 3) may be entirely omitted.

Results better than obtainable with the above sequence of operations may be obtained by a so-called double treatment, viz. :

Heat Treatment No. 4.

After forging or machining—
1. Heat to 1500° F.
2. Quench.
3. Re-heat to 1400°-1450° F.
4. Quench.
5. Re-heat to 600°-1200° F. and cool slowly.

This produces a refinement of grain not possible with one treatment and is resorted to in parts where extreme good qualities are desired.

Elastic limits varying from 45,000 lbs. per square inch in the annealed condition, up to nearly double this amount, are obtainable by proper combinations of quenching and drawing operations. A few practical tests will teach the best method of control. High elastic limits are safe so long as they are accompanied by generous reduction of area, say, 45 per cent. or better. Such steel will withstand shock and alternate stress.

This quality of steel is not intended for case-hardening, but by careful treatment it may be safely case-hardened. This should be in emergencies only, rather than as a regular practice ; and, if at all, only with the double treatment followed by the drawing operation ; that is, the most painstaking form of case-hardening.

SPECIFICATION No. 4.

.45 Carbon Steel.

The natural sources of supply for this steel are various—basic or acid open hearth, crucible or electric, the most natural source being the basic open hearth.

This quality represents a structural steel of greater strength than Specification No. 3. Its uses are more limited and are confined in a general way to such parts as demand a high degree of strength and a relatively low degree of toughness. At the same time with proper heat treatment the fatigue-resisting qualities are very high—higher than with any of the foregoing specifications.

This steel is commonly used for crank-shafts, driving-shafts and propeller-shafts. It has also been used for transmission gears, but it is not quite hard enough without case-hardening and is not tough enough with case-hardening to make safe transmission gears. It should not be used for case-hardened parts, except in an emergency. Other specifications are decidedly better for this purpose.

In an annealed condition this steel should have an elastic limit of about 50,000 lbs. per square inch, and in a heat-treated condition this figure may be nearly doubled.

The best heat treatment for this quality of steel for crank-shafts and similar uses is as follows :

Heat Treatment No. 5.

After forging or machining—
1. Heat to 1550° F.
2. Quench.
3. Anneal by heating to 1450° F.
4. Cool slowly in furnace, in lime or in soft coal.
5. Re-heat to 1400°-1500° F.
6. Quench.
7. Heat to 800°-1000° F. and cool slowly.

SPECIFICATION No. 5.

.80 Carbon Steel.

The source of this steel may be open hearth, crucible or electric furnace.

As stated under the specifications, this quality of steel is primarily for springs, and, generally speaking, for springs of light section.

The tensile test figures for this steel are unimportant inasmuch as it is intended for spring use.

The hardening and drawing of springs, that is, the heat treatment of them, is, as a rule, in the hands of the spring maker, but in case it is desired to treat, as for small coil springs, the following treatment is recommended :

Heat Treatment No. 6.

1. Coil.
2. Heat to 1400°-1450° F.
3. Quench in oil.
4. Re-heat to 400°, 500° or 600° F., in accordance with degree of temper desired, and cool slowly.

It must be understood that the higher the drawing temperature (Operation 4) the lower will be the elastic limit of the material. On the other hand, if the material be drawn at too low a temperature it will be brittle. A few practical trials will locate the best temper for any given use.

SPECIFICATION No. 6.

.95 Carbon Steel.

The natural source for this steel will be the same as for Specification No. 5—it may be made in the basic open hearth, crucible or electric furnace.

This grade of spring steel is suited for the most important springs. Properly heat treated, extremely good results are possible. Substantially the same remarks apply to this quality of steel as to Specification No. 5. It is possible that the quenching temperature (Operation 2, Heat Treatment 6) of the heat treatment may be lowered slightly because of the increase in carbon, and it is also probable that the drawing temperature (Operation 4) will be at a different temperature.

Sufficient data to specify the exact physical tests of hardness or other characteristics are not yet available in the spring industry, and it is therefore impossible to draw a hard and fast set of specifications. Such data are now being accumulated.

SPECIFICATION No. 7.

.20Carbon, 3½ Per Cent. Nickel Steel.

The source of this steel will be the open hearth, crucible or electric furnace.

As stated under the specification, this quality of nickel steel is primarily for case-hardening. The carbon is relatively low, which means that the interior metal of a case-hardened piece will be tough.

The elastic limit of this material in an annealed condition is 45,000 lbs. per square inch, with good reduction and elongation. It will respond well to heat treatment, with an increase of elastic limit up to 60,000 lbs. or 70,000 lbs. per square inch and with better reduction of area than in the annealed state. As before stated, this steel is not primarily for structural purposes,, but for case-hardening ; consequently the physical characteristics either annealed or heat treated are of minor importance. It is intended for case-hardened gears, both the bevel driving and transmission systems, and for such other case-hardened parts as demands a very tough, strong steel with a hardened exterior.

The case-hardening sequence may be varied considerably, as with Specification No. 2, those parts of relatively small importance requiring a simpler form of treatment. As a rule, however, those parts which require the use of nickel steel generally require the best type of case-hardening, viz. :

Heat Treatment No. 7.

After forging or machining—
1. Carbonize at a temperature between 1600° F. and 1750° F. (1650°-1700° F. desired).
2. Cool slowly in the carbonizing material.
2. Re-heat to 1450°-1525° F.
4. Quench.
5. Re-heat to 1400°-1450° F.
6. Quench.
7. Re-heat to a temperature from 250°-500° F. (in accordance with the necessities of the case) and cool slowly.

The second quench (Operation 6) must be conducted at the lowest possible temperature at which the material will harden. It will be found that this is sometimes as low as 1300° F.

In connection with certain uses it will be possible to omit the final drawing (Operation 7) entirely, but for parts of the highest importance this operation should be followed as a safeguard. Parts of intricate shape, comprising sudden changes of thickness, sharp corners and the like, should always be drawn, in order to relieve the internal strains.

SPECIFICATION No. 8.

.30 Carbon, 3½ Per Cent. Nickel Steel.

The natural source for this steel will be basic open hearth, crucible or electric furnace.

This quality of steel is primarily for heat treatment for structural parts where much strength and toughness are sought ; such parts as axles, spindles, crank-shafts, driving-shafts and transmission shafts.

In an annealed condition this steel has an elastic limit of about 55,000 lbs. per square inch. Under heat treatment this may be increased anywhere up to 160,000 lbs. per square inch, the ductility at this latter figure being satisfactory, reduction of area of at least 45 per cent. being obtainable. This wide variation of elastic limit is obtainable by the use of different quenching mediums—brine and oil—and the difference in drawing temperatures—from 500° F. up to 1200° F.

Heat Treatment No. 8.

After forging or machining—
1. Heat to 1450°-1500° F.
2. Quench.
3. Heat to 600°-1200° F. and cool slowly.

A higher refinement of this treatment is :

Heat Treatment No. 9.

After forging or machining—
1. Heat to 1450°-1500° F.
2. Quench.
3. Re-heat to 1350°-1400° F.
4. Quench.
5. Heat to 600°-1200° F. and cool slowly.

By proper regulation and changes of quenching and drawing temperatures, a wide range of physical characteristics may be obtained. The thickness of the mass treated, the volume and temperature of the quenching medium and other details peculiar to most hardening plants, must be recognized in order to get intelligent or desirable results.

This material may be case-hardened, but it is rather high in carbon for the practice of the average hardening department. The lower ranges of carbon—in the neighbourhood of .25—are satisfactory, but the upper ranges—in the neighbourhood of .35—approach the danger point, and steel of this carbon must be correspondingly carefully handled.

SPECIFICATION No. 9.

.15 Carbon, Chrome Nickel Steel.

The source of this steel will be open hearth, crucible or electric furnace.

As stated in the specification, this quality of steel is primarily intended for alloy steel frames and for heat treatment when so used. It may be used for other purposes, such as structural parts or for case-hardened parts, by suitably modified heat treatments, but as such use is not the intent of the specification, no further discussion is given along that line.

Heat Treatment No. 10

Heat treatment for frames after forming :
1. Heat to 1400°-1450° F.
2. Quench.
3. Heat to 1000°-1200° F. and cool slowly.

The exact temperatures for operations 2 and 3 must be determined experimentally, being somewhat dependent upon the thickness of the stock treated, the temperature of the quenching medium and the design. Generally speaking, thin sections, such as frames, respond very sharply to quenching operations and must be handled with corresponding care.

SPECIFICATION No. 10.

.20 Carbon, Chrome Nickel Steel.

The source of this steel will be open hearth, crucible or electric furnace.

This quality of steel is intended primarily for case-hardened parts of chrome nickel steel. The treatment may be so varied as to render it possible to use this quality of steel for many structural parts.

The strength of this steel in an annealed condition is not of much importance, as this alloy, as well as others, offers no material advantage over carbon steel unless it be heat treated. The heat treatment is substantially

the same sequence of operations as apply to other steels already dealt with, with suitable modification to be determined by practical experiment. An elastic limit of 120,000 lbs. per square inch is possible, with a large measure of reduction of area and elongation.

The proper quenching temperature for this grade of steel is from 1400° F. to 1500° F. The drawing temperature depends upon the elastic limit desired.

Case-hardened parts demanding this high grade of steel also demand the most careful treatment, viz. :

Heat Treatment No. 11.

After forging or machining—

1. Carbonize at a temperature between 1600° F. and 1750° F. (1650°-1700° F. desired).
2. Cool slowly in the carbonizing mixture.
3. Re-heat to 1400°-1500° F.
4. Quench.
5. Re-heat to 1300°-1400° F.
6. Quench.
7. Heat to 250°-500° F. and cool slowly.

The heating for second quench (Operation 5) should be carried out at the lowest possible temperature at which sufficient hardness may be obtained.

SPECIFICATION No. 11.

.30 *Carbon, Chrome Nickel Steel.*

The source of this steel will be the same as in the case of Specification No. 10.

This grade of chrome nickel steel is intended primarily for structural parts of the most important character. Parts requiring this grade of steel must be heat treated; otherwise there is no gain commensurate with the increased cost of the steel.

This quality is suitable for crank-shafts, axles, spindles, drive-shafts, transmission shafts and, in fact, the most important structural parts of the highest-priced cars.

The elastic limit of the annealed material is of no importance, as this steel should not be used in an annealed state. The elastic limit of the heat treated material may be carried as high as 175,000 lbs. per square inch, with generous reduction of area and elongation.

Heat treatment recommended :

Heat Treatment No. 12.

After forging or machining—

1. Heat to 1450°-1500° F.
2. Quench.
3. Re-heat to a temperature between 500° and 1250° F. and cool slowly.

A higher refinement of this same treatment is :

Heat Treatment No. 13.

After forging or machining—

1. Heat to 1450°-1500° F.
2. Quench.
3. Re-heat to 1400° F.
4. Quench.
5. Re-heat to a temperature between 500° F. and 1250° F. and coo slowly.

This grade of steel is sensitive and must be handled with a great deal of care. The temperatures should be controlled by pyrometer. The lower the temperature at which the proper response to treatment is obtained, the better will be the results. At the same time, if sufficient temperature is not used, there will be an incomplete or unsatisfactory response.

This steel is not intended for case-hardening, but may be so treated in an emergency. If case-hardening is attempted, the highest degree of care must be exercised.

SPECIFICATION No. 12.

.45 *Carbon, Chrome Nickel Steel.*

The source of this steel will be the basic open hearth, crucible or electric furnace, probably one of the last two.

The use of this steel is mostly for gears where extreme strength is necessary. The carbon is sufficiently high to cause the material, in the presence of chromium and nickel, to become sufficiently hard to make a good gear when quenched, without case-hardening (carbonizing).

The characteristics of this steel in an annealed condition are unimportant, as it should not be used in that condition. Heat treatment produces an elastic limit that may be carried to over 200,000 lbs. per square inch, with good reduction of area and elongation.

This steel is difficult to forge. During the forging operation it should be kept at a thoroughly plastic heat and not hammered or worked after dropping to ordinary forging temperatures, as cracking is liable to follow. The steel also becomes so very hard as to forge with great difficulty. On the other hand, too high a temperature is not advisable, as the steel becomes red-short and breaks. In brief, the forging temperature limits are narrow, and this steel must be re-heated more frequently than any of the other qualities dealt with. Generally speaking, the higher the carbon of chrome nickel steel, the more is this characteristic found to be true.

To heat treat for gears :

Heat Treatment No. 14.

After forging—

1. Heat to 1500° F. (plus or minus 25° F.).
2. Quench.
3. Re-heat to 1400°-1450° F. (Hold at this temperature one-half hour, to insure thorough heating).
4. Cool slowly.
5. Re-heat to 1450°-1500° F.
6. Quench.
7. Re-heat to 250°-550° F. and cool slowly.

This steel cannot be machined unless thoroughly annealed (Operations 3 and 4).

The final drawing operation must be conducted at a heat which will produce the proper degree of hardness. The desired Brinell hardness for a gear is between 430 and 470, the corresponding Shore hardness being from 75 to 85. This quality of steel should not be case-hardened under any circumstances.

SPECIFICATION No. 13.

.20 *Carbon, Chrome Vanadium Steel.*

The source of this steel will be open hearth, crucible or electric furnace.

As noted in the specification, it is primarily for case-hardening. The uses are for the best case-hardened parts of high-priced cars; that is, case-hardened shafts, gears and like important parts.

The physical characteristics of this steel annealed are relatively unimportant. The material will respond somewhat to heat treatment and be considerably toughened thereby. Its proper use is for case-hardening, in accordance with the following treatment :

Heat Treatment No. 15.

After forging or machining—

1. Carbonize at a temperature between 1600° F. and 1750° F. (1650°-1700° F. desired).
2. Cool slowly in the carbonizing mixture.
3. Re-heat to 1600°-1700° F.
4. Quench.
5. Re-heat to 1300°-1400° F.
6. Quench.
7. Re-heat to 250°-500° F. and cool slowly.

The high initial quenching temperature of this steel is noteworthy, that is, something a little over 1600° F. This is different from the other steels referred to and characteristic of chrome vanadium steel.

The heating for second quench (Operation 5) should be conducted at the lowest possible heat that will harden the exterior, carbonized surface. Practical experiment will develop the best temperature for local conditions in any hardening room.

SPECIFICATION No. 14.

.30 *Carbon, Chrome Vanadium Steel*

The source of this material may be open hearth, crucible or electric furnace.

The uses of this steel are for structural purposes, very much as set forth for .30 carbon, nickel steel and for .30 carbon, chrome nickel steel; that is, the important structural portions of an automobile—the crank-shaft, drive-shafts, axles and the like.

The physical characteristics in an annealed condition are unimportant. The steel should not be used in that condition—not that it is unsafe, but because there will be no gain commensurate with increased cost of the material.

For heat treatment :

Heat Treatment No. 16

After forging or machining—

1. Heat to 1600°-1700° F.
2. Quench.
3. Re-heat to some temperature between 500° F. and 1300° F. and cool slowly.

The elastic limit obtainable after heat treatment may be between 60,000 lbs. per square inch, and 150,000 lbs. per square inch, with good toughness as represented by reduction of area and elongation.

This steel may be case-hardened, but if so treated it must be handled with care on account of the relatively high carbon.

SPECIFICATION No. 15.

.45 *Carbon, Chrome Vanadium Steel.*

The source of this steel may be open hearth, crucible or electric furnace.

This quality of steel contains sufficient carbon in combination with vanadium to harden when quenched at a proper temperature.

The elastic limit after suitable heat treatment may be carried up to the neighbourhood of 200,000 lbs. per square inch, with reduction of area great enough to indicate good toughness.

This steel may be used for structural parts where exceedingly great strength is required.

The heat treatment may be as described for the .30 carbon, chrome vanadium steel; and the drawing temperature must be suitably modified to produce proper toughness. For gears this steel should be annealed after forging, the treatment to be about as follows :

Heat Treatment No. 17.

1. Heat to 1600° F.
2. Quench.
3. Re-heat to 1450° F.
4. Cool slowly.
5. Re-heat to 1600°-1650° F.
6. Quench.
7. Re-heat to 350°-550° F. and cool slowly.

This last drawing operation must be modified to obtain any desired hardness.

SPECIFICATION No. 16.

Silico-Manganese Steel.

The source of this steel will be the open hearth, crucible or electric furnace.

This steel is intended for structural parts, springs and gears. It should not be used without heat treatment.

The physical characteristics in the heat treated condition are similar to those of other alloy steels specified, the elastic limit being under control and ranging from 60,000 lbs. per square inch to 175,000 lbs. per square inch, with good reduction of area and elongation.

For structural parts the treatment should be :

Heat Treatment No. 18.

After forging or machining—

1. Heat to 1650°-1750° F.
2. Quench.
3. Re-heat to a temperature between 600° F. and 1400° F, and cool slowly.

Suitable temperatures for any given thickness of piece and the character

of the quenching medium must be determined experimentally.

When used for springs, this material may be treated as above with proper modification as to drawing temperature, with the probability that 800° F. as a drawing temperature will give about the proper characteristics.

SPECIFICATION No. 17.
Common Screw Stock.

This steel may be made by any process.

There are two types of screw stock commonly found in the market. The controlling element in one type is phosphorus, which is commonly found between .10 and .20 per cent. The controlling element in the other type is sulphur, which is commonly found between .07 and .15 per cent.

Ordinary screw stock is a free machining and very cheap steel, lacking strength and toughness. It is an unsafe steel, and must be kept out of the vital parts of an automobile.

Screws of these materials from hot-rolled bars should be heat treated and not used in a rolled or annealed condition. Screws made from cold-rolled bars are much stronger than if in a rolled or annealed condition. But the best results from both types of steels may be obtained if heat treated.

Heat Treatment No. 3 is suitable for screws. Heat treatment after machining produces the strongest screw If machined after treatment the grain of the material is, in effect, nicked by the thread and thereby weakened.

SPECIFICATION No. 18.
Low Alloy Steel.

This specification represents a class of steels that are made from ores containing chromium and nickel. Steels made from such ores may be referred to as natural alloy steels. At the same time other steels closely duplicating them in analysis are made by the addition of chromium and nickel-containing materials to an otherwise simple steel.

These natural alloy steels are low in cost as compared with chrome nickel steels in general. This is to be expected in view of the fact that the alloying elements are low in percentage and cost little in view of their origin. These steels are not intended to compete with the higher alloys of chromium or nickel, but are intended to furnish a better quality than carbon steel at a very slight increase in cost.

As found in the market, these steels contain about 1.00 per cent. of nickel and from .15 to .40 per cent. of chromium. Of such composition, there is a material gain in strength and toughness as the result of heat treatment.

These steels are obtainable in any desired carbon from .10 to 1.00 per cent. At a slight increase in cost, say, from 12s. to 20s. per ton in the billet form, these steels may be obtained containing from .30 to .60 per cent. chromium, which composition results in a very decided gain in physical characteristics.

The natural alloy steels respond to case-hardening treatments better than the plain carbon steels and the nickel steels of a similar carbon content. For example, a .20 carbon, natural alloy steel makes an excellent gear when properly carbonized and treated in accordance with Heat Treatment No. 7.

For structural purposes the .30 carbon or even the .40 carbon steel may be used, properly treated, in accordance with Heat Treatments Nos. 8 or 9.

SPECIFICATION Nos. 19 and 20.
Valve Metals.

These materials are high-nickel valve metals. They do not respond to heat treatment. The best that can be done with them is to treat for the purpose of securing uniformity of condition, by annealing at ordinary temperatures or by quenching from ordinary temperatures (1500° F. or thereabouts). Change of strength or ductility cannot be expected to any commercial degree.

SPECIFICATION No. 21.
Steel Castings.

The specification given for steel castings represents a quality commonly made by open hearth and crucible manufacturers.

Genuine steel castings, and not malleable iron and complex mixtures often found in the market masquerading under the name of steel, are referred to.

Genuine steel castings may be annealed or heat treated to great advantage for important parts. A steel casting of the composition given in the specification should be tough, so as to bend to a considerable angle before breaking. The elastic limit of such a casting in an annealed condition is in the neighbourhood of 35,000 lbs. per square inch.

The tensile strength of genuine malleable iron castings properly annealed is in the neighbourhood of 30,000 lbs. per square inch, with no elastic limit worth mentioning.

Like other castings, steel castings are subject to blow-holes. Consequently, they should not be used for the vital parts of an automobile. It is impossible to inspect against blow-holes. Steel castings for axles, crankshafts and steering-spindles are used only at great risk.

SPECIFICATION No. 22.
Gray Iron Castings.

The use of specifications for cast-iron in the present state of the foundry art is not very easy. The foundryman, if he is not accustomed to work to analysis, will object, although his iron may be within the specifications given 90 per cent. of the time. Moreover, if there are any defective cylinders he will be likely to lay it to the composition of the iron, whereas the fault may lie in his foundry methods, apart from composition.

Consequently these specifications should be used as indicating the ideal mixture—something for the foundryman to work to, even though he may not be willing to guarantee the analysis.

If trouble is experienced with cylinders, analysis of samples of the iron will show whether or not the composition is somewhere near what it should be. If the composition is very far from the specification here given, the purchaser will be justified in putting up strenuous objection.

Iron in accordance with this specification will be strong and reasonably close-grained in the thicknesses cast, and one that wears well.

SPECIFICATION No. 23.
Malleable Iron.

The remarks made in connection with the gray iron specification (No. 22) apply even more strongly to malleable iron. Iron of the composition given, properly annealed, will make a strong and tough casting; but improperly annealed it will not make a good casting.

Castings that are received brittle may be so from two causes: first, unsuitable mixture of iron; second, incomplete annealing. Consequently, if brittle castings are received, they should be analyzed, and if the analysis is correct, then it is certain that the annealing operation was not properly performed.

Castings varying seriously in composition from that given in the specification, especially as to phosphorus, are liable to be brittle, even if properly annealed, and should therefore be avoided.

List of Heat Treatments.

Heat Treatment No. 1.

After forging or machining—

1. Carbonize at a temperature between 1600° F. and 1750° F. (1650°-1700° F. desired).
2. Cool slowly or quench.
3. Re-heat to 1450°-1500° F. and quench.

Heat Treatment No. 2.

After forging or machining—

1. Carbonize at a temperature between 1600° F. and 1750° F. (1650°-1700° F. desired).
2. Cool slowly in the carbonizing mixture.
3. Re-heat to 1500°-1550° F.
4. Quench.
5. Re-heat to 1400°-1450° F.
6. Quench.
7. Draw in hot oil at a temperature which may vary from 300°-450° F., depending upon the degree of hardness desired.

Heat Treatment No. 3.

After forging or machining—

1. Heat to 1500° F.
2. Quench.
3. Re-heat to 600°-1300° F. and cool slowly.

Heat Treatment No. 4.

After forging or machining—

1. Heat to 1500° F.
2. Quench.
3. Re-heat to 1400°-1450° F.
4. Quench.
5. Re-heat to 600°-1200° F. and cool slowly.

Heat Treatment No. 5.

After forging or machining—

1. Heat to 1550° F.
2. Quench.
3. Anneal by heating to 1450° F.
4. Cool slowly in furnace, in lime or in soft coal.
5. Re-heat to 1400°-1500° F.
6. Quench.
7. Heat to 800°-1000° F. and cool slowly.

Heat Treatment No. 6.

1. Coil
2. Heat to 1400°-1450° F.
3. Quench in oil.
4. Re-heat to 400°, 500° or 600° F., in accordance with degree of temper desired, and cool slowly.

Heat Treatment No. 7.

After forging or machining—

1. Carbonize at a temperature between 1600° F. and 1750° F. (1650°-1700° F. desired).
2. Cool slowly in the carbonizing material.
3. Re-heat to 1450°-1525° F.
4. Quench.
5. Re-heat to 1400°-1450° F.
6. Quench.
7. Re-heat to a temperature from 250°-500° F. (in accordance with the necessities of the case) and cool slowly.

Heat Treatment No. 8.

After forging or machining—

1. Heat to 1450-°1500° F.
2. Quench.
3. Heat to 600°-1200° F. and cool slowly.

Heat Treatment No. 9.

After forging or machining—

1. Heat to 1450°-1500° F.
2. Quench.
3. Re-heat to 1350°-1400° F.
4. Quench.
5. Heat to 600°-1200° F. and cool slowly.

Heat Treatment No. 10.

1. Heat to 1400°-1450° F.
2. Quench.
3. Heat to 1000°-1200° F. and cool slowly.

Heat Treatment No. 11.

After forging or machining—

1. Carbonize at a temperature between 1600° F. and 1750° F. (1650°-1700° F. desired).
2. Cool slowly in the carbonizing mixture.
3. Re-heat to 1400°-1500° F.
4. Quench.
5. Re-heat to 1300°-1400° F.
6. Quench.
7. Heat to 250°-500° F. and cool slowly.

Heat Treatment No. 12.

After forging or machining—
1. Heat to 1450°-1500° F.
2. Quench.
3. Re-heat to a temperature between 500° F. and 1250° F. and cool slowly.

Heat Treatment No. 13.

After forging or machining—
1. Heat to 1450°-1500° F.
2. Quench.
3. Re-heat to 1400° F.
4. Quench.
5. Re-heat to a temperature between 500° F. and 1250° F. and cool slowly.

Heat Treatment No. 14.

After forging—
1. Heat to 1500° F. (plus or minus 25° F.).
2. Quench.
3. Re-heat to 1400°-1450° F. (Hold at this temperature one-half hour, to insure thorough heating.)
4. Cool slowly.
5. Re-heat to 1450°-1500° F.
6. Quench.
7. Re-heat to 250°-550° F. and cool slowly.

Heat Treatment No. 15.

After forging or machining—
1. Carbonize at a temperature between 1600° F. and 1750° F. (1650°-1700° F. desired).
2. Cool slowly in the carbonizing mixture.
3. Re-heat to 1600°-1700° F.
4. Quench.
5. Re-heat to 1300°-1400° F.
6. Quench.
7. Re-heat to 250°-550° F. and cool slowly.

Heat Treatment No. 16.

After forging or machining—
1. Heat to 1600°-1700° F.
2. Quench.
3. Re-heat to some temperature between 500° F. and 1300° F. and cool slowly.

Heat Treatment No. 17.

After forging—
1. Heat to 1600° F.
2. Quench.
3. Re-heat to 1450° F.
4. Cool slowly.
5. Re-heat to 1600°-1650° F.
6. Quench.
7. Re-heat to 350°-550° F. and cool slowly.

Heat Treatment No. 18.

After forging or machining—
1. Heat to 1650°-1750° F.
2. Quench.
3. Re-heat to a temperature between 600° F. and 1400° F. and cool slowly.

THE EFFICIENCY OF WORM GEARING.

By E. R. Whitney.*

There seems to be a popular impression, and, I am sorry to say, among some engineers as well, that the worm gear is a device that is all right for an elevator, a mechanical motion, or for a steering device where irreversibility is desired, but that it is not to be considered seriously for the driving gear of a motor car or waggon. The idea seems to be a single-thread worm having a spiral angle of 5 or 10 degrees and an efficiency of 25 to 50 per cent. We often hear questions like these: Will it coast? Will it drive backwards? Is the efficiency high enough for automobile work? Does it not wear out rapidly?

Contrary to the opinion expressed by writers

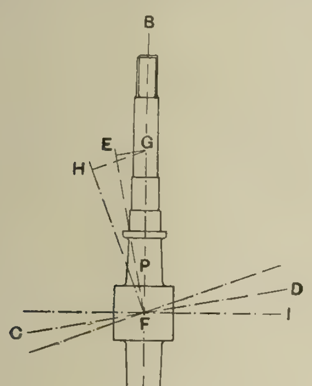

Fig. I.

from time to time, and especially by several writers recently, the worm gear, within its legitimate field, is superior in many respects to other forms of gearing. It has its limitations, and is not applicable to all conditions, but for certain classes of motor-driven vehicles it is superior to other devices on the score of efficiency, durability and simplicity.

The efficiency of a worm gear is a function of the thread angle and the coefficient of friction. The coefficient of friction is a function of lubricant and the nature of the bearing surfaces.

The relation of the thread angle to efficiency may be expressed as the amount of sliding between surfaces for the amount of useful work done.

Frederick A. Halsey, in a work on the subject, has expressed this relation very clearly, and I cannot do better than to quote his words:

"The reason why an increase of pitch, other things being equal, or, in other words, an increase of the angle of the thread, gives higher efficiency will be understood from Fig. 1. If *A B* be the axis of the worm and *C D* a line representing a thread, against which a tooth of the wheel bears, it will be seen that if the tooth bears upon the thread by a pressure P, that pressure may be resolved into two components, one of which, *E F*, is perpendicular, with the other, *E G*, is parallel to the thread surface. The perpendicular component produces friction between the tooth and thread. The useful work done during a revolution of the thread is the product of the load P, and the pitch of the worm, while the work lost in friction is the product of the perpendicular pressure *E F*, the coefficient of friction and the distance traversed in a revolution, which is the length of one turn of the thread. Now, if the angle of the thread be doubled, as indicated, the load P remaining the same, the new perpendicular component *F H* of P will be slightly reduced from the old value *E F*, while the length of a turn of the thread will be slightly increased. Consequently, their product and the lost work of friction per revolution will not be much changed. The useful work per revolution will, however, be doubled, because, the pitch being doubled, the distance travelled by P in one revolution will be doubled; for a given amount of useful work the amount of work lost is therefore reduced by the increase in the thread angle, and, since the tendency to heat and wear is the immediate result of the lost work, it follows that that tendency is reduced. For small angles of thread the change is very rapid, and continues, though in diminishing degree, until the angle reaches a value not far from 45 degrees, when the conditions change and the lost work increases faster than the useful work, an increase of the angle of the thread beyond that point reducing the efficiency.

"This general consideration of the subject shows the principles at the bottom of successful worm design, but a more exact examination is desirable. According to Professor Barr, the efficiency of a worm gear, the friction of the thrust bearing being neglected, is:

$$e = \frac{\tan a \,(1 - f \tan a)}{\tan a + f} \text{ (approximately)}$$

in which

e = efficiency.

a = angle of thread (being the angle *D F I* of Fig. 1).

f = coefficient of friction.

"This formula gives no clear indication of the manner in which the efficiency varies with the angle, and the diagram, Fig. II., has been constructed to show this to the eye. The scale at the bottom gives the angles of the thread from 0 to 90 degrees, while the vertical scale gives the calculated efficiencies, the values of which have been obtained from the equations and plotted on the diagram. In the calculations for the diagram it is necessary to assume a value for f, and this has been taken at .05 and .025. The calculated efficiencies from these values are shown in the lower and upper curves respectively. The experiments made by Mr. Wilfred Lewis for Wm. Sellers and Company showed an increase of efficiency with the speed. The present diagram may be considered as confined to a single speed, and at the same time is not to be understood as showing the exact efficiency to be expected from worms, but rather to exhibit to the eye the general law connecting the angle of the thread with the efficiency."

Fig. III. is the plotted result of an efficiency test on a set of Hindley worm gears, as used in the Commercial Truck Company's 1,000-pound electric delivery waggon. The data on these gears is as follows:—

Pitch	0.96 inches
Lead	3.84 inches
Ratio	9¾ : 1
Centre distance	6.796 inches
Angle of thread (average)	28 degrees
No. threads in worm	4
No. teeth in gear	39
Diameter of worm	2.8 inches
Diameter of gear	11.917 inches

The test was made on a stock rear construc-

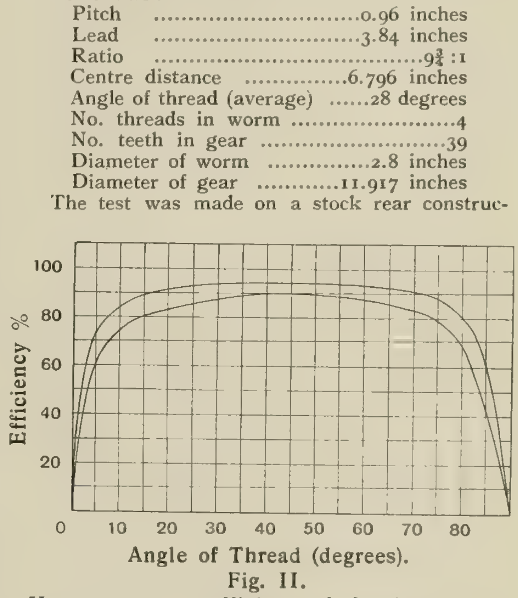

Fig. II.

Upper curve coefficient of friction ·025.
Lower curve coefficient of friction ·05.
43° 34'. angle of maximum efficiency for worm alone.
52° 49'. angle of maximum efficiency for worm and seat.

tion, the load being taken on a brake. The drum was mounted on a shaft with a square end passing through the square holes in both gears of the differential. The torque values were taken by a platform scale. The worm gear was driven by a standard series-wound automobile type motor, and which has a normal rating of 85 volts, 22 amperes, at 1,200 revolutions per minute. Brake tests were first made on the motor, and before starting the test for efficiency, observations were made at various loads to deter-

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* Paper read before the Society of Automobile Engineers, June, 1911.

mine the effect of varying the voltage, and consequently the speed; it was found that the torque values on the Prony brake measuring the output of the gears was practically constant for a given current, with a considerably wide range of voltage and speed.

It is interesting at this point to note this fact in view of the results obtained by other experimenters, they having found a decided change in the efficiency with changes in speed. This may possibly be accounted for by the much higher surface speed of gears in this test, ranging from 200 to 800 feet per minute.

The fact that the voltage and speed could be disregarded greatly simplified the test, as it was then necessary to observe only current and torque, the efficiencies being calculated directly from the torque tests on motor and gears.

The curve of gear torque, Fig. III., is plotted from torque values on the worm gear divided by the gear ratio. The speed curve shows the motor speed at 85 volts with varying loads.

The test corresponds to actual service conditions with the waggon and full load, working through a range of from a slight down-grade to about an 18 per cent. up-grade. The maximum efficiency of 93 per cent. corresponds to a co-efficient of friction of .032.

An interesting and valuable application of the principle of worm gear efficiency may be made in connection with steering gears. The efficiency of a steering gear for lorries, and especially heavy lorries, should be as high as possible, and yet with the proper degree of irreversibility, which means the proper amount of friction. I have found that these conditions can be met with worm thrusts of hardened steel, and a large spiral angle of worm thread, equally as well as with ball thrust bearings and a smaller spiral angle. The steel thrust washers are not so apt to give trouble in service, and are steadier.

The durability or life of a worm gear is a direct function of efficiency, and it follows that if a gear is produced that is high enough in efficiency throughout the full load range to be practical it will also be satisfactory for durability.

The Commercial Truck Company's first worm-driven waggons were put into service about two years ago. Some of the first wagons have now covered approximately 25,000 miles. Recent examination of the gears showed very little signs of wear. I would conservatively estimate the life of these gears at from 50,000 to 60,000 miles.

The worm-gear drive is not a cheap device, and the results as indicated above for efficiency and durability cannot be expected unless the gears are properly designed, constructed of the best materials and accurately mounted on high-grade anti-friction bearings. Unless facilities are at hand for doing accurate machine work, the use of the worm-gear drive had better not be attempted, but once properly constructed and mounted it is practically as free from trouble as a pair of spur gears.

The Hindley worm gear is much more expensive than the straight type, but the superior bearing between the worm threads and gear teeth and the greater durability warrant the additional expense. The commercial Hindley gear is a decidedly different product from the so-called theoretically correct Hindley gear, and in practice the writer has experienced none of the fancied difficulties in mounting accurately enough in the direction of the worm shaft to meet all practical requirements, in fact, much less accuracy is required in this direction than is required at right angles to the shaft with the straight worm.

There is no such thing as a so-called theoretically correct Hindley gear; that is, the threads of the worm do not have a full bearing against the whole area of all gear teeth in mesh, but the commercial Hindley gear approaches very closely to this condition.

Some engineers claim that even in ratios that can be covered by a single reduction bevel gear the worm has advantages in a high-grade car in being much easier to make silent and has nearly equal efficiency. The efficiency is undoubtedly lower than with a single reduction bevel gear made with equal accuracy and equal mounting, but is considerably higher than a double reduction bevel and chain or chain and chain. We will assume that the legitimate field for the worm gear is only where its efficiency and durability are equal to or better than in the case of other devices to do the same work. Then, keeping in mind the principle underlying efficiency, the requirements for road clearance, etc., it would on this basis be limited to ratios of from about 6:1 to 14:1. These limits are an approximation only, and are subject to modification, depending on exact conditions.

In order to put in convenient form the comparison of all features of the worm gear with other forms of gearing Table No. 1 has been made up. In consulting this table it must be understood that the comments on the different ratios are not unalterable, although they are ap-

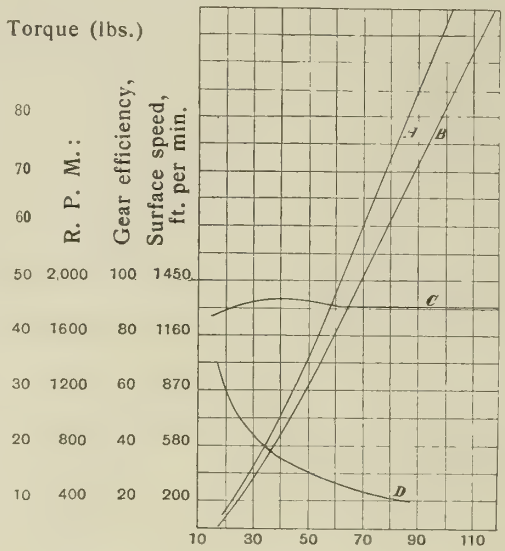


Fig. III.

Ampere Output.

A. Motor torque. B. Gear torque (reduced to motor speed). C. Gear efficiency. D. Speed.

proximately correct for the usual form of construction.

Table 1 is a comparison of all features of worm gears with other forms of gearing, and

ful discussion followed the reading of the paper, but there was read a letter from Mr. H. Kerr Thomas, who has had considerable experience, both in America and in this country. The letter runs:—

"In two respects, I find that I am in variance with Mr. Whitney. Firstly, he has accepted the formula for calculating the efficiency of worm gears, which was first published, as he advises, by Mr. F. A. Halsey. This does not apply to worms having high pressure-angles, and in automobile gears these high pressure-angles are necessary in order to allow reversibility. Inasmuch, however, as pitch angles are usually between 30 degrees and 60 degrees, between which Mr. Halsey's formula is approximately correct, Mr. Whitney is perhaps justified in giving the conclusion he does. Mr. Whitney has calculated the results for two co-efficients of friction (1) .025 and (2) .05, these I presume are given only as examples, as in practice these figures should be divided by ten, the actual co-efficient being anywhere between .002 and .004. It is through this fact being so little appreciated that so many manufacturers have objected to the use of worm gears, on the grounds of inefficiency.

"The other point on which I regret that I cannot agree with Mr. Whitney is his conclusion that worm gears are not adaptable to pleasure cars. In reference to this I can only say that successful pleasure cars driven by worm gears have now been running in Europe for over twelve years. In fact, this device, like so many others, had its beginning in pleasure cars, and was subsequently adapted for trucks. The first designer to use a worm driven axle was Lanchester. The Lanchester car has been consistently built with worm drive for twelve years. Gradually other English manufacturers, and more recently French ones, followed his example, until to-day in Europe the number of worm driven pleasure cars is almost co-extensive with the number of high-class manufacturers.

"While personal opinions may be justifiable as academic questions, I think it will be the general

Table 1.

GEAR RATIO	LOWER THAN 6:1	6:1 to 14:1	HIGHER THAN 14:1
Other forms of gearing.	Single reduction bevel.	Double reduction Bevel and Chain Chain and Chain	Double reduction (Only required on heavy electric trucks) Bevel and Chain Both chain Both spur.
EFFICIENCY	LOWER	HIGHER	LOWER
DURABILITY	EQUAL OR BETTER	BETTER	EQUAL OR BETTER
NOISE	EQUAL OR LESS	LESS	LESS

Table 2.

	Speed M.P.H.	Engine or Motor Speed	Gear Ratio for 36 in. Wheels
PETROL VEHICLES			
Pleasure car	40	1200	3·2
1000-pound waggon	20	1200	6.4
5-ton truck	10	800	8.5
ELECTRIC VEHICLES			
Pleasure car	20	2000	10.7
1000-pound waggon	15	1400	10.1
5 ton truck	7	1600	24.4

Table 2 has been drawn up to give some idea of the requirements for gear ratios with different types of motor-driven vehicles, and to indicate where the worm gear is applicable.

It will be seen that for conditions as indicated in this table and with limitations as fixed above, that the worm gear is not applicable to petrol pleasure cars or heavy electric waggons, but that it can be legitimately used for all petrol business waggons and trucks, for electric pleasure cars and light electric waggons.

Criticism.

It is obvious by reason of every-day practice here that the conclusions arrived at by the writer of this paper are erroneous and therefore the following criticism is interesting. Very little use-

conclusion that the only reliable judgment on the value of any mechanical device is the practical test, and in this respect, concensus of opinion points unmistakably to the fact of the constantly increasing adoption of worm gears on all kinds of automobile vehicles."

CATALOGUES, ETC.

A NEW GLASS REFLECTOR of parabolic form is now being used by C. A. Vandervell and Co. for their electric automobile head-lamps.

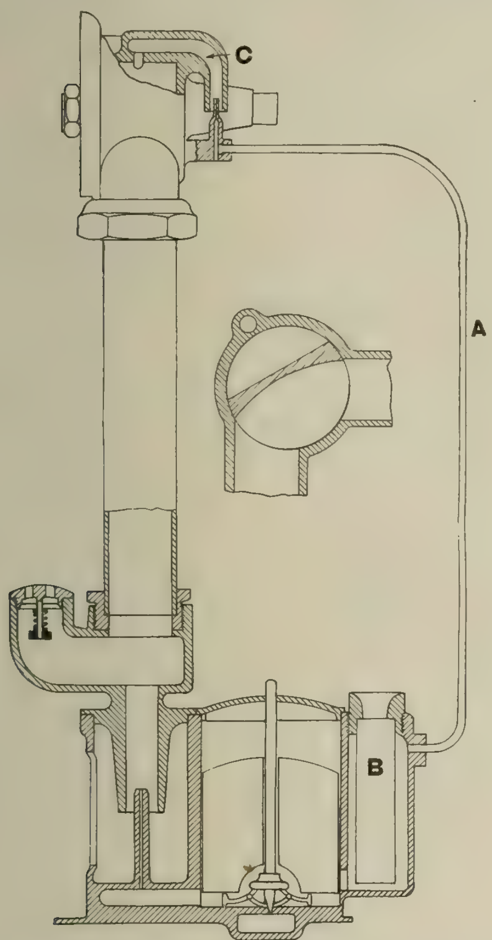
DETACHABLE RIMS.—The "Facility" circumferentially divisible and detachable rim is described in full detail in a new catalogue issued by The Facility Motor Rim Co.

RECENT AUTOMOBILE PATENTS.

By Eric W. Walford, F.C.I.P.A.

A Slow Speed Carburettor.

For slow running the main carburettor is cut out of action by closing the throttle valve. Suction then takes place along the pipe A which communicates with a chamber containing a gauze tube B. This stands in petrol maintained at a constant level in the ordinary float so that the gauze can be saturated, and the air drawn

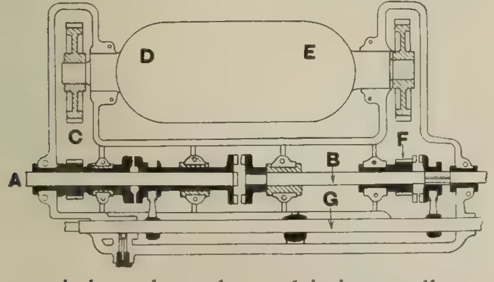


along the pipe A is richly mixed with petrol. This mixture passes through the nozzle shown, into a chamber C communicating with the throttle valve. The throttle valve is capable of cutting both carburettors out of action or putting either one into communication with the induction pipe.

No. 25,088/1910. Société Anonyme des Automobiles Delauney Belleville.

Hydraulic Gearing.

Hydraulic gear systems comprise a pump of variable delivery driven by the petrol engine and a hydraulic motor



coupled to the axle, and it is usually so arranged that on top gear the drive is direct and the motor and pump rotate solid and do no internal work. To effect this is the object of the present invention. The engine shaft is shown at A, while the shaft B is coupled to the axle. The shaft A is connected by chain gearing at C to the hydraulic pump D. The hydraulic motor is arranged at A and is coupled by chain gearing to a chain wheel F loose on the shaft B. It will be seen that the shafts A and B carry dog clutch members, and these are controlled by a sliding rod G. When in the position illustrated the gear is in neutral ; but when the rod G is moved to the left one set of dog clutches is engaged when the drive is transmitted througth the chain gearing and the hydraulic mechanism, which may be adjusted to vary the speed of the driven shaft B as required.

When the rod G is moved to the right the clutches at the abutting ends of the shafts A and B are engaged, and the chain gearing disengaged. Thus the power passes through the shaft A to the shaft B and the hydraulic gearing and chain transmission stand still.

No. 1,072/1911. Société Anonyme des AutomobilesDelauney Belleville.

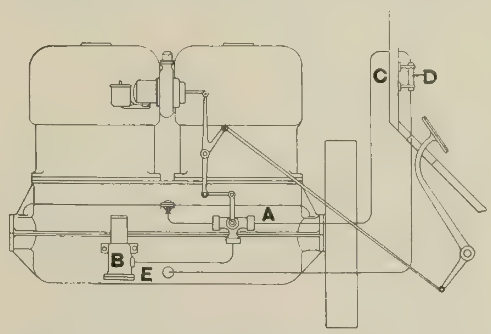
An Improved Piston.

With engines of the Diesel type, in which the fuel is injected during the working stroke, the top of the piston is liable to be burnt away or oxidised, and to prevent this the piston head is here provided with a central face of nickel. This possesses the same co-efficient of expansion as the cast iron forming the body of the piston, and for its attachment it is preferably cast in the piston, being provided with grooves or dovetails to hold it securely.

No. 6,311/1911. Gebruder Sulzer.

Lubrication Control.

The amount of oil supplied to the engine varies with the throttle opening. The accelerator pedal illustrated, in addition to operating the throttle, controls a bye-pass valve A, whereby oil from the oil



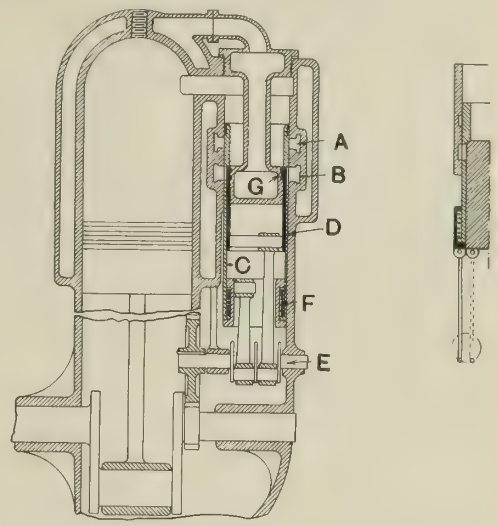
pump is passed along the bye-pass C from a tell-tale D mounted on the dashboard, and back to the crank chamber sump E. The oil which does not pass to the return circuit goes to the bearings, the quantity thus varying with the work which the engine is doing.

No. 14,104/1910. Daimler Motoren Gesellschaft, E. Moewes and A. Vischer.

A Slide Valve Engine.

The cylinder is cast with a cylindrical chamber at the side connected by a port at the top of the combustion chamber and provided with passages A and B, the former being the inlet and the latter the exhaust.

Lying between the wall of this cylindrical chamber and the central water-cooled column is a ported sleeve C, and within this is another sleeve D, which is actuated by a separate crank off the half-speed shaft E, the cranks being set at an angle of 20 deg. to 30 deg. apart. The sleeve C is actuated as regards its downward movement through the medium of a spring F, and the top of the



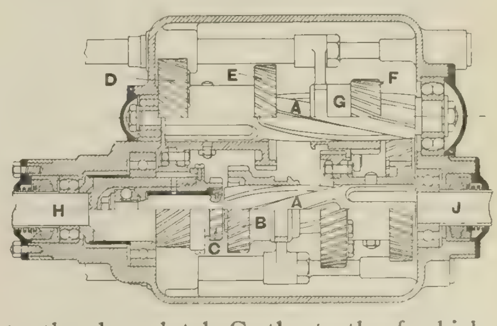
sleeve D seats against a shoulder formed at G on the interior of the outer sleeve C.

The two sleeves rise together when opposite the inlet port A and the relative positions of their cranks causes them to separate and put the inlet passage into communication with the cylindrical chamber and engine cylinder. During the compression and firing strokes the two valves are moving downwards, the port in the outer sleeve being closed as illustrated. When opposite the exhaust port the sleeve is moving in an opposite direction opening up the exhaust. The compression space essential for this invention seems to be very considerable.

No. 29,731/1910. G. C. Samain.

A Gear Box with Helical Gears.

The wheels in this gear box are provided with single helical teeth, the problem being to enable these to be slid into and out of mesh. For this purpose each sliding gear wheel is mounted upon helical splines A, whilst on the driven shaft is the gear member B which is moved to the left to engage the dogclutch C and effect a direct drive. The helical splines enable the wheels to be slid properly into mesh, in spite of the helical arrangement of their teeth, and the same applies



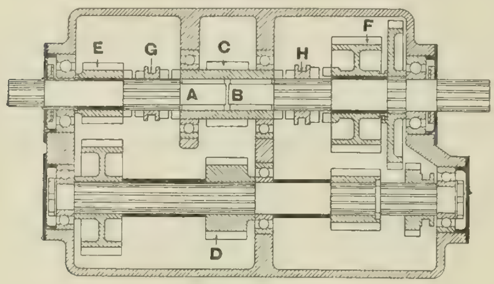
to the dog clutch C, the teeth of which are seen to be helical. To obviate any trouble from end thrust, the inventor

states that "the ratio of the pitch length of the helices of the teeth of the constant wheel D to the length similarly derived in the case of the wheels E and F on the lay shaft, is equal to the ratio of the power transmitted to the lay shaft G from the driving shaft H, to the power simultaneously transmitted to the driven shaft J from the counter shaft G." By adopting this construction the helical splines A tend to take the end thrust of the helical wheel, whilst the splines also permit of sliding of the helical gears into and out of mesh.

No. 4,549/10. J. H. A. B. Dailey.

An Ingenious Chain Transmission.

In this transmission system silent chains are used in place of gears and four

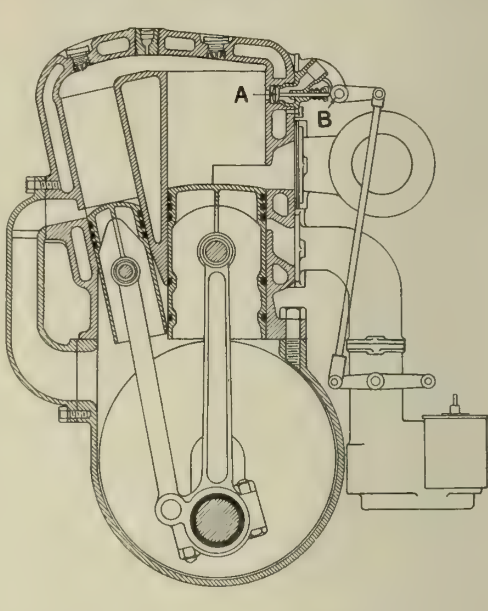


changes, including a direct drive, are obtainable with only three pairs of chain wheels. The driving shaft projects into the gear box as far as A and the driven as far as B and round these projecting shaft parts are arranged bearings on which runs a free chain wheel C which drives on to a wheel D fixed on the lay shaft. At the left hand of the gear box is shown a chain wheel fixed to the lay shaft and driving by a chain on to a wheel E loose on the driving shaft, and correspondingly at F is a wheel loose on the driven shaft. Sliding on the driving shaft is a clutch member G which can lock either of the wheels E and C, and a similar clutch H is on the driven shaft which can lock either of the wheels C and F. For the first speed the power is transmitted from the wheel E through the lay shaft to the wheel F, the dog clutches being moved as far away from one another as possible. For the second speed the clutch G is moved to the right, the power then passing from the wheel C which is now locked to the driving shaft, to the lay shaft, and thence back to the driven wheel F. For the third speed the power goes from E to the lay shaft, back to the wheel C which is clutched to the driven shaft by means of the clutch H, whilst for the direct drive both clutches G and H move towards one another, locking the wheel C and both the driving and the driven shaft.

No. 1,697/11. S. Bramley Moore.

A Two-stroke Engine.

This invention enables the compression of a two-stroke engine to vary with the throttle opening, and it is effected by providing the combustion chamber with a compression release valve A, actuated by



a cam at B, which is connected with the throttle valve.

No. 4,059/1911. F. R. Draper.

THE INSTITUTION OF AUTOMOBILE ENGINEERS.

The Summer Meeting of the Graduates' Section.

This year the Graduates of the London, Birmingham and Coventry branches of the Graduates' Section of the Institution met at Coventry for their summer meeting at the invitation of the Coventry branch. The programme, which was booked for Friday and Saturday, July 21st and 22nd, included visits to six of the most prominent works in Coventry, connected with the staple industry of the city, and a dinner and concert. The first visit was to the works of the Auto Machinery Company, where were seen a great variety of small repetition parts being turned out in great numbers. The second visit was to the machine tool works of Alfred Herbert, Ltd. Here were seen machine tools in all stages of construction, from the rough material to the contents of the show rooms. Of course, the most noticeable feature of these works was the rapidity of the cutting tools, for which the firm are famous. After lunch a pleasant variant was furnished by the large works of Rudge-Whitworth, Ltd., which showed object lessons in the production of bicycles, motor-bicycles and detachable wire wheels. In the motor cycle department activity was everywhere apparent, and in the wire wheel department wheels were to be seen in all stages of construction to fit a great variety of makes of cars.

Press work was the key-note of the methods of the Coventry Chain Company (the next visit), practically every machine in the vast new building being a press. Even the roller bushes of some of the chains are pressed out of sheet metal, as are the side plates, while the rollers are stamped out of circular metal discs. The fitting of the chain parts is completed by rapid action tapping machines, which rivet the heads of the cross pins. Needless to add, the main drives from the section drive motors were of the Coventry noiseless chain variety. In the evening a dinner, at which about 40 graduates and guests were present, was presided over by Mr. F. W. Lanchester, the President of the Institution, supported by Mr. Basil H. Joy, the secretary, and also Mr. Burford, of Messrs. Humber, Ltd., together with Messrs. L. J. Shorter and Rhys Pugh, chairman and hon. secretary respectively of the Coventry branch; Messrs. Hillhouse and Wilson, chairman and hon. secretary of the Birmingham branch; and Messrs. Burchall and House, of the London branch. Messrs. G. W. Lewis (member), Dorling and Shilson (associates), and A. L. Clayden were also present. The speech making gave good indication of the enthusiasm of the graduates and their determination to take their, properly, important place in the life of the Institution.

The toast of the Institution was submitted by Mr E. W. Lewis, and in the course of his remarks he pointed out the fluctuations that took place in the motor trade. They usually had, he remarked, two lean years, and these were followed by one fat year; or sometimes they would have one lean year and two fat years. However, as far as he could see it was likely that they were now to have a succession of prosperous years, and that, he thought, augured well for the institution of which they were members. As they as an institution had had some lean years, they were now in a fair way of having a continuance of good years. An indication of that was the strength and enthusiasm of their graduates' section.

The toast was heartily received, and the Chairman, responding, said he was of the firm conviction that the Institute had a great future before it. Mr. Lewis had reminded them that the automobile industry had a curious way of rising and falling, and he stated that it dropped two years and was only up one. That view, he felt sure, was exaggerated—at least he sincerely hoped so—because otherwise the gentlemen present would not have much of a future before them. The automobile industry, he felt sure, would flourish, and in connection with this industry the graduates' section was doing a great work. All would agree that the policy they had adopted of making their main object the furtherance of the automobile industry from the technical standpoint, was the best line of action any association could take. They must always do all they could to further the technical side of the industry of which they were, so to speak, the trustees. The graduates were quite a necessity in the trade; for young men had no knowledge of the restriction the older engineer had, and the demands of to-day were such as would have greatly surprised the engineers of the past. The engineers of to-day performed a job which the engineers of old would have shaken their heads at. In conclusion, he urged the graduates to make the Institute as successful as possible; for he said he was thoroughly convinced that the association had a great future before it.

Mr. Burford, returning thanks on behalf of the Coventry motor manufacturers, said that the motor industry was closely allied with the welfare of Coventry. That welfare depended very largely on the industry with which they were so closely associated. The motor industry required men of energy, tact, skill and enthusiasm, and he felt sure after listening to the able speeches of the Graduates that the Graduates' section in the Midlands, with the help of Mr. Joy, would become a very large and potent force in the association of which they were members. In conclusion he wished the graduates' section continued success, and he hoped that the interest and attendance which their secretary had asked for would be forthcoming.

The visits were resumed on Saturday morning, when a tour of the works of Messrs. Humber, Ltd., was made. Here the party followed the construction of motor cars, motor cycles, cycles and aeroplanes from beginning to end, including the body work of the cars and the making of the smallest screw. The testing shop especially excited the interest of the party, as there were no less than three different types of engines undergoing test, namely, car, motor cycle and aeroplane. In the erecting shop the arrangements for running and testing the back axles were also admired.

An examination of the processes at the Albion Drop Forgings Works was a fitting conclusion to a most interesting and instructive meeting. As the workmen had left the works, the budding automobile engineers were graciously allowed by Mr Brett to experiment with the drop hammers and to get thereby a real working knowledge of the processes of drop forging.

LUDWIG LOWE AND CO. always manage to produce a catalogue which shall compactly contain the whole of that particular information which they have desired to set before their customers. The latest booklet is of handy size, well bound, and printed on excellent paper. In it there is information in full concerning every type of small tool which could be supplied by the firm, beginning with those which are used for chuck operation, together with code order words and illustrations of the chucks and spanners involved. Afterwards angle plates, vices, centres, arbors, sockets, and carriers are dealt with in a similar manner.

The second portion sets forth the milling cutters, with diagrams of their use.

THE

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CONTENTS.

THE USES OF NON-RUSTING METALS.

In discussing the wonderful development of the automobile, it has become customary correlatively to make mention of the immense effect the demands of the motor industry have had upon the evolution, or at least the commercial production, of new alloy steels. To-day there is a marked tendency to make the utmost use of steel in one form or another, and much ingenuity has been displayed by the inventors of ever-increasing methods whereby steel can be made to replace other metals. In motor car construction more and more steel is used every day, while cast iron, malleable cast iron, bronze and aluminium are being displaced, as fresh means for forging and casting steel are developed. Already there are some cars in which practically the only parts not made of steel are the cylinders, and perhaps the pistons, while a fresh substitution of steel for some other metal is generally regarded as a veritable triumph.

Now, although it is probably often the case that a steel structure is better than that which is replaces, it would be well to inquire as to whether there are not yet other metals, not now in common use, but superior to steel for certain specific automobile purposes. Of course, the great drawback to the use of steel is its liability to rust when exposed to the atmosphere, and the consequent painting or protection which is necessary for it where it has to be exposed to the chemical action of air and water. Supposing for a moment that some new material was found as strong and as cheap as mild steel, but unaffected by ordinary oxidising—would not it almost immediately replace steel for all out of doors machinery? The question admits of only one answer, so, in cases where cheapness is not a paramount consideration, but strength and lightness are important, there is reason to believe that certain alloys, well known to metallurgists, possess advantages above that of steel.

About twelve months ago, the introduction of a new aluminium alloy of great strength caused a great deal of discussion as to whether its lightness would make it an important factor in future design, but its non-liability to weathering did not receive much attention as a desirable quality. Until the last few years cars were not expected, by their makers, to last for more than half a decade, because design was improving so fast that eighteen months was long enough to make quite good cars obviously inferior to new products, but now a good car ought to last ten years at least, and give good satisfactory service all the time. During such a period it is safe to predict that every joint, bearing or surface, to which moisture had access, would rust sufficiently to cause looseness and rattle, or even serious diminution of strength. The rims of the wheels, the brakework, the undershield, the wings, the spring shackles and the springs themselves would certainly lose efficiency through rusting; unless very exceptional and unusual care was taken to repaint directly chips and scratches appeared, and to grease all exposed joints. With a touring car it may be reasonable to expect great care for pleasing exterior to result in similar provision for external mechanism, but, as the heavy vehicle comes more into general use, there can be no doubt that thousands of chassis will be run, year in and year out, without more overhauling or re-embellishment than is absolutely necessary.

For springs there is probably nothing, at present, which could be used instead of steel, but there is much to be said in favour of the scheme of placing thin strips of non-rusting metal, such as zinc or brass, between the leaves, and thereby preventing binding. On the other hand, for the other exposed moving parts of a chassis, a brief perusal of the, now old, reports of the Alloys Research Committee will be enough to show that there are quite a fair number of aluminium-copper compounds possessing the necessary qualities. Many of them are costly, but for a number of parts they would be well worth the price, if long service without replacement was a desideratum. To take one example from the ninth report of the Research Committee, which deals with the alloys of copper, aluminium, and manganese, the alloy No. 2, which contained 10.02 % aluminium, .92 % manganese, and 89.06 % copper, sustained an ultimate stress of from about thirty tons up to over fifty tons per square inch, according to the method of preparation and subsequent treatment. At the same time the yield point varied from thirteen tons to forty-two tons per square inch, and the metal proved to be most adaptable for the engineer, making good castings, rolling easily, and machining, under most conditions, as well as ordinary yellow brass, while it was quite unaffected even by the action of sea water.

Such an alloy would be quite strong enough to allow it to be used for some vital parts of an automobile, and it would seem to possess a special advantage for exposed bearing work (such as brake links), because in some special abrasion tests the alloy was found to have better wearing properties than steel, even with the latter entirely protected from oxidation.

No doubt many of our readers are well acquainted with the report to which we have referred, but the total neglect of copper-rich aluminium alloys proves that their entirely peculiar qualities are not generally realised. Metals which do not require protection by a coating of some other substance are likely to play an important part in the engineering of the future, and for automobile construction they will surely some day find a wide field of usefulness.

AMERICAN TOURING CAR DESIGN.

A consideration of the difference between American and European practice, and the causes therefor.

(*Continued from page* 424.)

Engine Design.

IN the last issue differences in frames, springing, axles and gearboxes were dealt with broadly, and it can at once be said that these are the only parts of a chassis in which the American designer can compete with the European. For service in America, or in other comparatively rough countries, the American chassis has advantages over the European, but in engine work a very different state of affairs prevails. At first sight it is not easy to account for the peculiarities of American automobile engine design, but it is not too strong a statement to say that the average American car engine bears a close resemblance to the typically French designs of 1906-1909. The high speed engine and the long stroke engine are practically unknown; bearings are usually inadequate, timing gears are noisy, and most of the attachments and engine fittings are of a decidedly clumsy nature. The extremely careful proportioning of internal parts, the smooth external lines and the convenient accessories of the average European car engine have yet to be discovered in America.

It is difficult to write on this subject of American engine design without commenting on the extraordinary ignorance of American engineers concerning the detail of British or Continental work. This is, of course, probably more than equalled by our own lack of knowledge concerning American doings, but the Americans have avowedly studied and copied European work, whereas there are not many firms in this country who have taken their cue from America. It appears that the only European cars which have been imported to any extent are the very largest and most expensive and, as was pointed out in these columns last December, the largest and most expensive cars now made on this side of the Atlantic are nearly all of them extremely old fashioned in design, by reason of the fact that the demand for big cars has fallen off so enormously with the improvement of the 15-20 h.p. class. This is, of course, not so in America, for there very large and expensive cars still sell in enormous quantities. The most common type of car in America is typified more or less by such vehicles as the Hudson, the National, the Velie, the Chalmers, and a host of others the very names of which are entirely unknown here. These mostly have a four-cylinder four-inch engine with a stroke but slightly in excess of the bore. In the design and construction of these cars, quality receives about an equal consideration with cheapness of production. That is to say, an improvement in design is not necessarily sacrificed because it necessitates an awkward machining operation, but small sacrifices are made to avoid waste of material and, above all, to avoid labour costs. These cars therefore do not carry the refinements to be found on their bigger brethren, and they are seldom as well equipped as our own "second class" cars. They are, however, vastly superior to the half-dozen or so of the cheapest possible cars which so far have represented America in England.

Engines smaller than 4 inch bore are quite uncommon, and there is a strong tendency for any car of less power than about 40 b.h.p. to degenerate into more or less of a "freak," often with two horizontally opposed cylinders, an epicyclic transmission, and so forth. It is not meant to convey the impression that a car of this type is necessarily bad, because there are many answering this last description which give very good service. However, to consider the American four-inch engine in detail will cover the larger motors, and the smaller may be neglected altogether.

For cylinders the multi-casting is only just coming into vogue, and one hears brought against it all the old arguments which were used here two or three years ago. Two cylinder casting is most usual, but many cars are still turned out with separate single cylinders. There is reason to believe that American foundry work is not so good as our own; about in the same proportion as our own (taken all round) is not so good as French and Belgian work. This shows itself in the usually greater thickness of cylinder wall, while cylinders are far more often strengthened by thickening up the section than by the introduction of webs or fins. Head plugs are extremely popular, although the orifice at the top of the bore is seldom made use of for machining, and therefore it may be said that these plugs are used chiefly as an aid to the founder. Valve arrangement, too, makes for heavy cylinders, because the T-head system is standard practice, only quite a few cars having the valves arranged all on the same side. The pistons also in American engines are immensely heavy; the material is always ordinary cast iron, and there are almost always four separate rings, spaced well apart, with a deep thick skirt lipped at the bottom for strength. Close fits are the rule rather than the exception, though stepped tapering in grinding is often resorted to. In fact, studying the running of a comparatively long stroke four-inch engine in one of the manufacturer's experimental rooms, brought home to the writer very forcibly the enormous difference to the British engine in which the adoption of light, short, loose-fitting pistons has resulted.

Connecting rods are almost always nickel steel stampings, and they too, are quite needlessly strong and heavy. All this reciprocating weight does, however, result in the use of most generously proportioned crankshafts, but the low maximum speed of revolution which the heavy reciprocating parts compel, has led to the evolution of three and even two bearing designs with the almost total extinction of the five-bearing shaft.

Crankshafts are, of course, invariably stamped, and not infrequently they are stamped with so small an amount of stock that they can be ground up at once without turning. Only in the rarest instances are the webs machined, but not a few firms are balancing the shafts on Norton running balance machines of the type described in *The Automobile Engineer* for April last. A noticeable point in crankshaft design is the large radius always used where the shaft or crank pin runs into the web, and this applies with equal force to all shafting throughout the chassis. For bearings white metal is universal, and it is usually fitted inside brass bushings, although die-cast babbit is gaining in favour. The majority of manufacturers finish these bearings by hand, but minimise the scraping by running a series of very finely graduated reamers through each big end individually, and through the crankshaft bearings when these are in position on the crankcase. In the Cadillac factory the bearings are not scraped to fit individual shafts, but are touched up by hand so as to make practically perfect contact with a standard shaft, each fitter being provided with a dead hard piece of shaft ground up to within extremely fine limits of diameter. For oil grooving many different designs are in use, just as in this country, and no one of them appears to possess any outstanding advantages over the others. One of the smaller manufacturers has lately been trying a spiral grooved bushing for bearings (this is not new, but deserves better appreciation). Three turns are made near the middle of the bearing, the groove being of ordinary oil groove dimensions, and having a pitch of something like three-eighths of an inch. The direction of thread is, of course, chosen so as to screw the oil back into the box, the idea being to prevent escape, and, when new at least, the device appears to answer extremely well. Ball bearings are used in one or two engines, notably for the big six-cylinder cars made by Lozier Company and, for this particular car, they are claimed to have given complete satisfaction for several years, but they are quite audible in running, while they are very large and correspondingly costly.

Having thus discussed bearings, it will be in natural sequence next to consider lubrication systems. The ideal, which in the writer's mind is a fully forced system, is as rare in America today as it was in Europe ten years ago, but the old-fashioned plain splash system is giving way to modifications which aim at controlling the amount of oil available by each cylinder. There is a strong tendency to divide the bottom half of the crankcase into separate divisions—one for each cylinder—and to feed these separately by an external supply pump. Sometimes external leads are taken from the same pump to the main crankshaft bearing. Sometimes the pump supplies through drips on the dashboard, and very often there are gravity fed drips supplying either the crankcase alone or the crankcase and the main bearings. In other words, in cars which pride themselves upon their lubrication system one finds a more or less faithful copy of a 1907 Mercedes system, while those who are less particular employ an adaptation of one or other of the

This article is the second of a series written by a member of the staff of the "Automobile Engineer," who has recently made a tour of the American Motor Industry. The next article will deal with American works organization and methods of manufacture.

"mechanical" lubricators commonly associated in Europe with the name of "Dubrulle." A very few firms are using the trough system with a circulating distributing pump, and no doubt the fact of the Stearns and Columbia companies having taken up the Knight engine will result in still further popularising this particular method of oiling.

Of course, this simplification of the lubrication system assists greatly to cheapen the engines, not only by reducing the number of parts, but by lessening the complication of the crankcase castings. For the latter, aluminium alloy is always used, and the engine is almost invariably carried from the main frame, sometimes by four crankcase arms, but very often by two arms at the back and a single central swivel joint in front. As has been remarked before, there is a tendency either to combine the crankcase and gearbox castings, or to place them as close together as possible, but this does not really affect the design of the engine. Of course, the pitches used for timing gears are much smaller than those employed in transmissions, but it is usual to find a plain steel pinion on the crankshaft driving two phosphor bronze camshaft wheels, which in turn drive further steel pinions for the magneto and water pump. As a result the noise is very considerable, and experiments are being made here and there with vulcanised compounds or with forms of celluloid, in the endeavour to find a quiet material. he few makers who are using helical gears are getting better results, and one or two firms have been extremely successful with cast iron helical gearing, using a steel crankshaft pinion with a cast iron camshaft wheel. In engines, too, for driving cross shafts or vertical shafts, small bevel wheels are standard practice, the skew gear being used but very little.

In cooling, the American designer has a much harder problem to attack than anyone in Europe, for the cooling system must be capable of performing satisfactorily in air temperatures varying from 40° below zero, up to more than 100° above. Obviously, if the water system is efficient at a higher temperature it will be more than efficient at a lower, and so one finds most cars designed for summer work, the winter being left more or less to take care of itself. This being so, freezing troubles are very often heard of, and it is the absence of winter trouble that has been the war cry of the Franklin Company, who have now for many years pinned their faith to an engine with forced circulation air cooling. Of course a pump is very rarely dispensed with, most often being situated on the upper half of the crankcase and driven at crankshaft speed from an extra gear outside one of the camshaft driving wheels. The magneto frequently occupies a similar position on the opposite side, and on most cars there is a separate accumulator ignition system, having a commutator on top of a vertical shaft, driven by a bevel on any convenient part of either camshaft. Owing to the popularity of dual ignition, one but rarely sees a car without spark control fitted both to the magneto and the other system, while an air lever very often makes a third item.

Mention of the latter leads to consideration of carburettor design, although this can be dismissed very quickly by saying that the ability to procure fuel at an average price of 5d. per gallon has not encouraged manufacturers to elaborate carburettors to any great extent. Only in quite a few instances are care supplied with carburettors by their own makers, specialising firms with very large outputs controlling practically the whole of this manufacture. The annular float and centrally situated jet are very popular, and cork floats are at least as common as the hollow metal variety. Still, speaking generally, carburettors are of the crudest possible nature, few of them being comparable for a moment with any of the better known devices here, from the point of view of either power, flexibility or economy.

In silencer design no great differences are apparent, except that the public demand and receive a very much smaller degree of excellence. Also an exhaust cut-out is fitted on every chassis, and is in use during 90 per cent. of country running by the majority of drivers. The reason for this is not easy to find, being, in fact, only capable of explanation on the assumption that the driver enjoys the noise created.

So far, most comparisons which have been made between the American and European engines have been to the disadvantage of the former. There is, however, one point in which they are superior to the European, and that is in their consistency. Just as a whole series of cars turned out by one maker will probably be equally quiet, so one does not find "good" and "bad" engines in a series of cars all supposed to be exactly alike, unless there is some easily discoverable reason to account for it. It appears that this desirable feature is obtained by careful flywheel balancing on a rotational balance machine, a favourite being the "Defiance" which has been employed for several years in America, is now being used by the Daimler Company in England, and has been employed by the Austrian Daimler Company in Vienna for a season or two. From one of the first few flywheels to be balanced in the Daimler works it was found to be necessary to remove 13 ozs. of metal before a good rotational balance was obtained, albeit the material was the very best quality of cast iron. Of course, the flywheel was fairly large and heavy. The effect of the unbalanced force created by a mass of 13 ozs. revolving at a radius of say 10 inches, and at 1,000 r.p.m. can easily be imagined, and it is not difficult to see that an engine thus handicapped could not fail to compare most unfavourably with another exactly the same in every respect, but with a homogenous flywheel. An illustration of the Defiance machine, and a short description of the method of handling it appears on page 480 in this issue.

Clutches.

Having now disposed briefly of the engine, transmission, frame and axles, it only remains to discuss the small details and connecting link betwen the engine and transmission. In Europe the metal disc clutch threatened at one time to become the standard type, but since then a return has been made in favour of the leather cone. The latter is used fairly extensively in America, and the metal disc clutch also is not uncommon, but there are one or two other types not at all widely known in Europe which seem to be well worthy of attention from designers. One of the best clutches which the writer has ever had the pleasure of handling was fitted to an experimental car now being tested by the engineering staff of the Hudson Company. This is a plate clutch having five ground, thick steel plates and five inner plates faced with Raybestos. This clutch could be slipped very easily, could be run dry or run with oil, and showed no signs of heating under either condition. It is, of course, not expensive, and should be more durable than an ordinary disc clutch, while, owing to the lesser drag from the small number of contact surfaces, it had no tendency to "hang in" and so cause trouble in changing speed. Another clutch which worked very well, at least when new, was somewhat similar, having thick and thin plates alternately, the thick plates bearing a number of cork inserts, and these could be run either lubricated or dry. Yet another successful clutch of quite a different type is the Pierce Arrow, in which a cone is used faced with phosphor bronze segments instead of with leather, the flywheel being of the usual cast iron. In other details, as regards striking mechanism, position of thrust bearings, situation of springs, and so on, there is but little to distinguish an American clutch from an English.

Small Details.

As regards the lubrication of small parts it is hard to say whether there is much difference in style : the more expensive the cars the better cared for are they in these respects all the world over, but there is very little slovenly spring shackle work to be found in America, while springs bushed with phospor bronze are quite common. This, of course, is explained by the bad roads which throw so much more work on the springs. Tank and pipe work is usually good, heavily lead-coated sheet being a popular material for petrol tanks with the idea of minimising "drumming," and tanks have often very large filling caps or hand holes, the non-existence of the two gallon can (its place being taken by five or ten gallon canisters) and the national regard for the importance of time, have made any device which assists quick filling up of greater value there than here. Filtration of fuel usually takes place between the tank and carburettor, and petrol piping generally is large, thus minimising the chances of stoppage therein. Pedal and lever work generally is distinctly poor, adjustments not often being found.

Body Work.

It has already been remarked that the American chassis is usually designed to be more or less flexible, and that the principal components are not so attached to the frame as to increase its natural rigidity. This being so, it is easy to see that special difficulties will be met with in making bodies, because, if the frame is to be allowed to whip, the body must bend as well. Now, with the ordinary European body, in which the strength is all supplied by wood, and the metal panels are mere loose insertions, continual stressing by frame flexion results in the more or less complete disintegration of the whole structure. The wood frames work loose at all their joints, the panels become loose in the wood, the doors jam or gape, and the whole body rattles and groans. To get over this, American engineers have

given very serious attention to the problem and have produced bodies made, in some cases entirely of metal, and in other cases principally of metal, with the simplest possible strengthening. By far the most interesting bodies, however, are those employed exclusively by the Pierce Arrow Company and by a few other concerns as well. These are made entirely of cast aluminium, and in an open body there will be about half a dozen castings. The upper part of the front seat would be one piece, and the lower portion a separate piece. The upper part of the rear seats would similarly be one casting, or two joined up the back, while there would be two or three pieces forming the lower part of the body according to whether a wheelhouse was employed or not.

The castings themselves are really wonderful pieces of work, being extremely thin and very free from blow holes. Externally they are cast with all beadings and mouldings, and internally have a few strengthening fins, while they are webbed for rivetting along lines where they join to adjacent panels. In the back seat castings, for example, the webs would join the upper and lower portions, being secured by copper rivets and would also carry the seat-board, while just sufficient wood would be attached to the upper lip of the top castings to enable the upholstering to be performed. The doors would be cast separately, of course, and their hinges would be secured direct to the aluminium by small bolts. Unfortunately, the writer was not able to obtain the exact weight of one of these bodies, but personal estimation of some of the parts leads to the belief that they are very little, if any heavier than a European built-up body. Nor are they very costly, the castings being ordered in large lots, and very many bodies made to one pattern. The castings are wonderfully clean when they arrive from the foundry, and may be trimmed up very rapidly with files by hand. All that it is really necessary to do is to remove accidental roughness, and to clean up the edges of the beading. When this has been done, and the whole face of the casting is free from lumps, it is scratch brushed with a steel wire mop, care being taken to crosshatch most of the surface so as to give a good hold to the paint. The first coat or two of priming is then sufficient to fill up all small pin holes, and an extremely small amount of stopping is needed. Owing to the completely unabsorbent nature of the surface the paint dries rapidly and with a much more even surface than is the case on wood, where it is always absorbed more in one part than another. This reduces the labour of rubbing down, and the final finish obtained could not possibly be surpassed by any known method. The front and rear sets of castings when they are made up, are mounted on ash runners which serve to connect them together, and it is these runners which are bolted to the frame. Having of themselves no very great strength, this allows a sufficient movement in the door spaces to permit the frame to give without straining the body, and, for transverse vibration, there is a certain amount of elasticity in the aluminium itself sufficient to compensate for the flexion. As a result, cracks do not appear even after long use, nor do the surfaces deteriorate by the action of sunlight or damp, as nothing can get beneath the paint.

Another type of body, and this perhaps is the most extensively used, derives its strength from a wooden frame, but the whole of the exterior surface, including the beading, is hammered sheet aluminium. In this case a single sheet is commonly employed for the whole of the rear portion, another single sheet for the front seats, and, of course, two separate pieces for the doors. Most elaborate curves and even wheelhouses can be hammered out of a single sheet, the operation being performed with fair rapidity by a skilled operator, simply by passing the sheet through a small, rapid acting power hammer. Although such a body is not so strong as the cast aluminium structure, it possesses a more or less flexible surface entirely free from joins. Thus there is nowhere for cracks to appear, and no woodwork to paint.

Covered bodies are not very common, the landaulette being quite rare, and the limousine or coupé seems to be regarded as only suitable for use by elderly people or for purely town work. As the roads are bad enough to put serious difficulties in the way of the ordinary open-body builder, it is easy to see that the task for the constructor of the limousine is very hard indeed, and this perhaps accounts for the comparative scarcity of the closed body. Another point which must exercise some influence as well, is the popularity of horse drawn vehicles of the buggy type, in which there is no more shelter and protection for the occupants than that provided by a folding cape cart hood. Of course, an open car without a hood is scarcely ever to be seen in America, because when it is not needed as a protection against rain, it is required as a sun shield. A black waterproof material is generally used, and there are a variety of ingenious attachments for preventing rattle between the hoops and other portions of the framework. Wings are usually made of rolled steel, and are very often finished in black enamel, irrespective of the colour of the car. Cast aluminium is displacing wood and rubber for running boards and even for the front seat foot-boards. It seems to possess great advantages in that it is always clean, neither warps nor breaks, and seems no more liable to rattle than wooden boards.

Wheels.

In curious contrast to the bodywork the wooden wheel is supreme in America, although there are indications that steel will replace it before very long. The wire wheel is more rare than it was in England in 1903, and the few sets which have been imported are regarded more as mechanical triumphs than practical improvements. Thus one designer told the writer that, although he had been experimenting with a set of British-made wire wheels for some time, he was afraid to use them when in very rough country, because they did not "look strong." In other quarters, however, it is realised that the strength of the steel wheel must result in its becoming universal sooner or later, and lateral rigidity is of such enormous importance on rough roads that the case for the steel wheel is twenty times as strong in America as in Europe. Mechanically, the survival of the fittest is a rule without exception, and while this will doubtless bring about many changes in American car construction, as well as in design on this side of the Atlantic, reduction of weight and increase of strength are matters of much greater importance to the American than to the European automobile engineer.

Tendencies.

During his tour the writer had the opportunity of conversing with about three hundred different engineers, all engaged in automobile work, and found that the greatest amount of interest was taken in engine improvements. Wherefore it is reasonable to assume that the development of engine design in America is likely to be rapid, in the near future. Also the interest in new forms of valve mechanism is even greater than it is here, and there are but few firms who are not making experimental models with entirely peculiar engines. Of course, the Knight engine success in Europe has been the stimulus, and the manufacture of this motor being now commenced by two American car makers no doubt will encourage still more experimenting. Already there are several rotary valve engines which have given so much satisfaction in practice that they are likely to be found as the standard equipment on some of the chassis to be exhibited at the New York show in January. This being so the development of the poppet valve engine is retarded somewhat, so that European refinements, such as forced lubrication and enclosed valves—to mention a couple of examples—are not gaining support at the rate their importance deserves.

With regard to the long stroke engine much scepticism prevails, and such makers as are trying it are doing so very gently, using a stroke-bore ratio not much removed from unity. In fact, what passes for a long stroke in America would be considered rather short here. Still, just one or two firms are making motors with modern proportions, and are very pleased with the results. In one particular instance which came to the notice of the writer a certain firm were on the point of producing a new model, and the choice lay between cylinders with a square proportion, or cylinders with the same capacity, but a stroke-bore ratio of 7/5. On making the latter they were surprised to find that they obtained over ten per cent. more power than was anticipated, at the normal running speed of the short stroke engine, and much more at still higher speeds. During the next couple of years no doubt this experience will be repeated by the majority of American manufacturers and, once a fair number of long stroke cars appear on the market, it is probable that the demand for the present type will cease utterly.

Of course, the adoption of long strokes will act again just as it has here, by producing lubrication, vibration and noise troubles; so it is likely to be quite three years before the average American engine is as good for all round purposes as is the average European engine now.

In transmission work Europe is, of course, ahead in respect to worm drives, and the better success of many of the American firms in the endeavour to produce quiet running bevel gears will retard the adoption of the worm. Also the question of clearance between the axle and either the road or the floor boards counts for so much more in America that the difficulty of applying the worm is greater.

PLATE I., accompanying article on "Steel," by Dr. Walter Rosenhain. (See page 456.)

Fig. I. Section of an ingot of cast lead. Showing the crystalline structure.

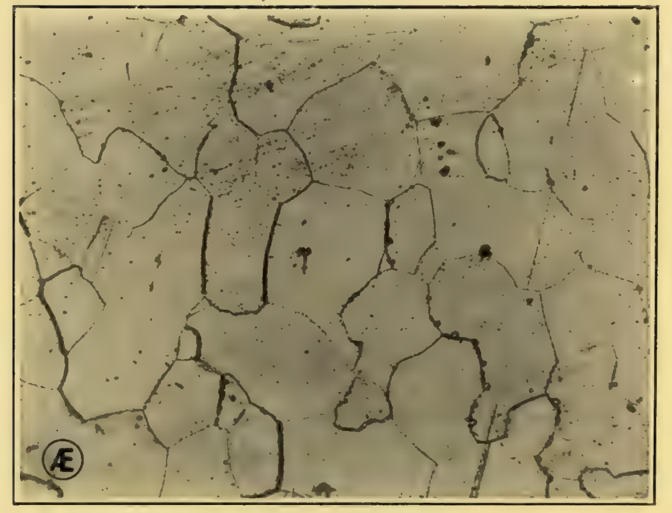

Fig. III.*

Fig. IV.†

Fig. V.†

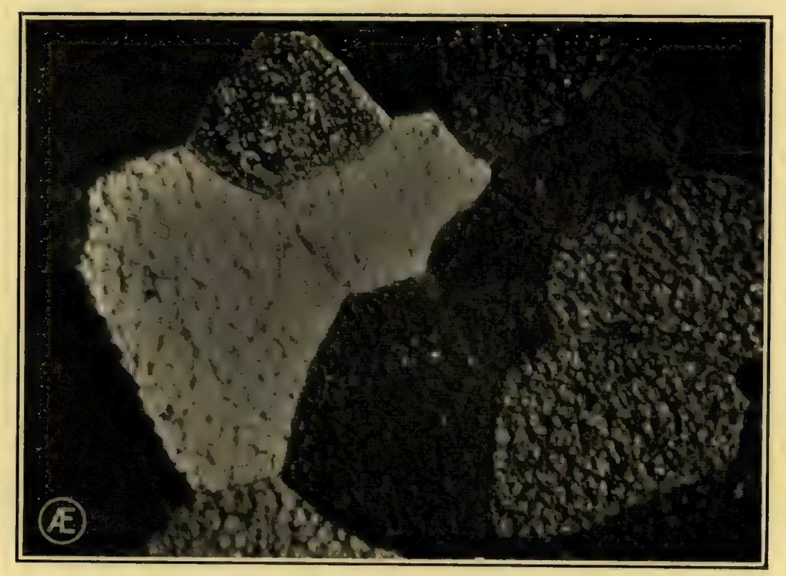

Fig. VI.*

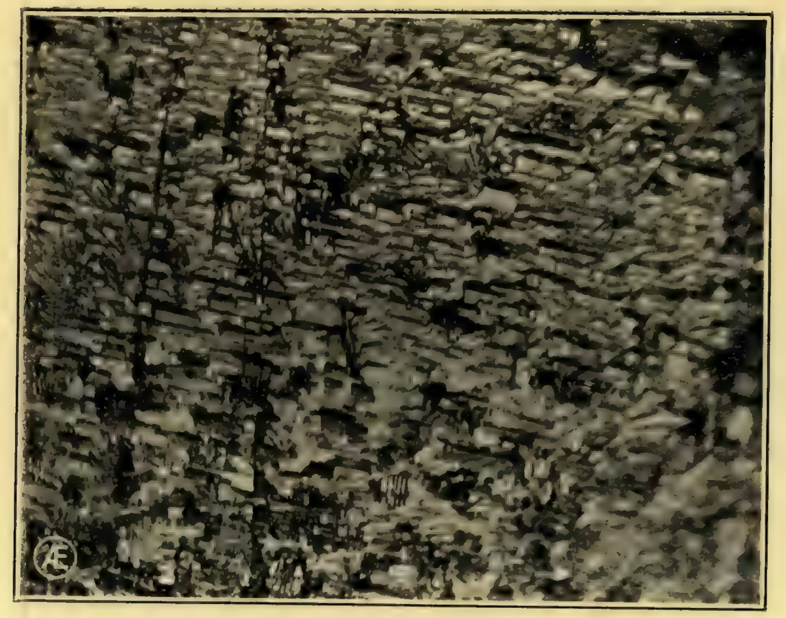

Fig. VII.

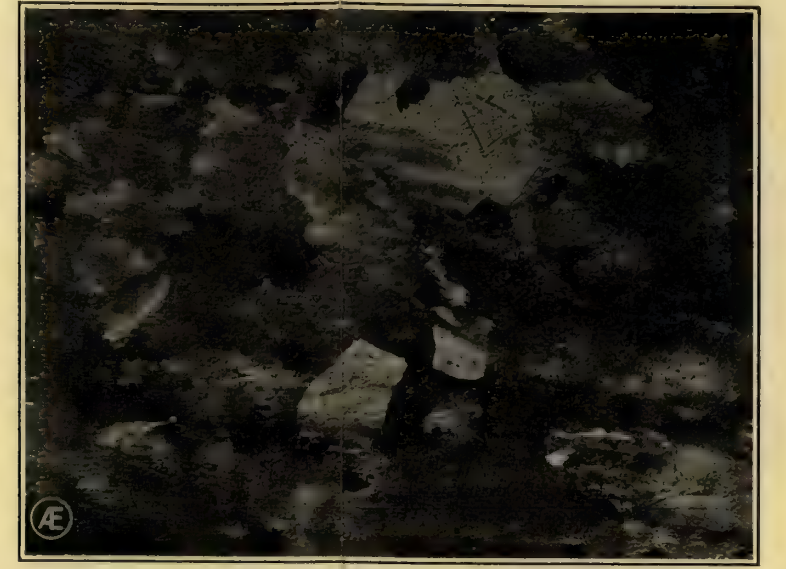

Fig. VIII. Swedish wrought iron. Showing Ferrite crystals and a few dark particles of cinder or slag.

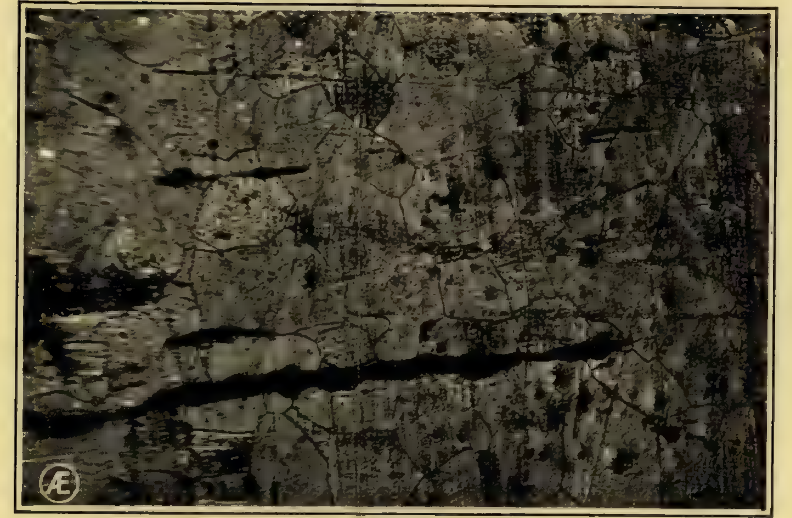

Fig. IX. Ordinary wrought iron. Showing Ferrite crystals and large strips of cinder or slag.

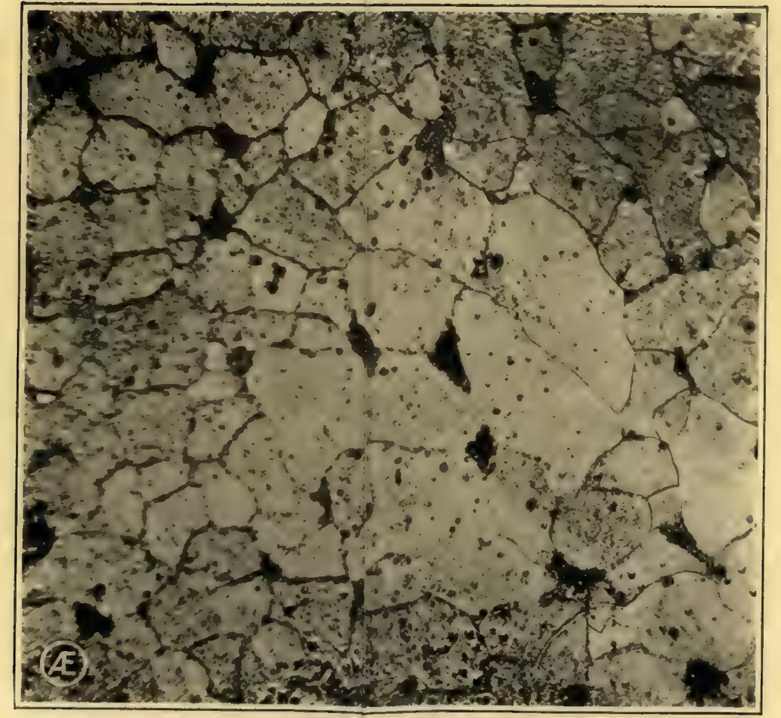

Fig. X. 0·05 % carbon steel. Showing Ferrite crystals and a little Pearlite.†

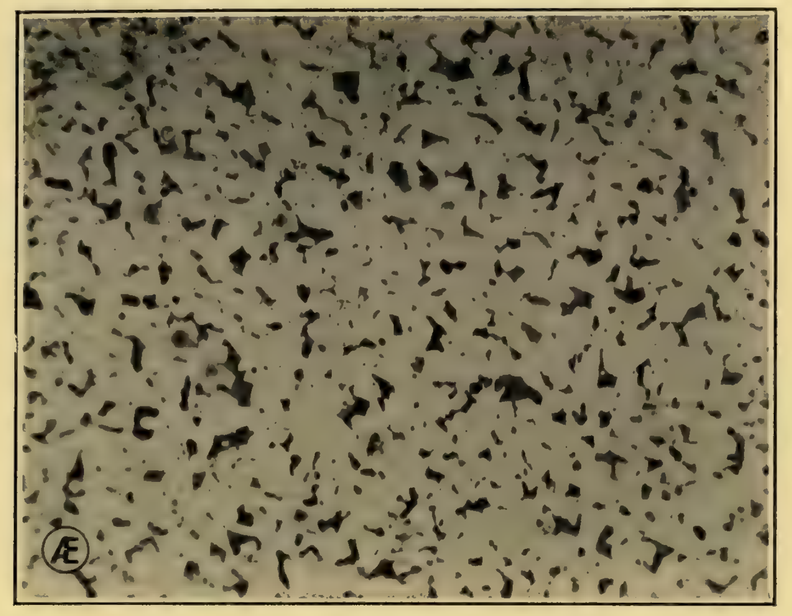

Fig. XI. 0·2 % carbon steel magnified by 150. Showing Ferrite and Pearlite.

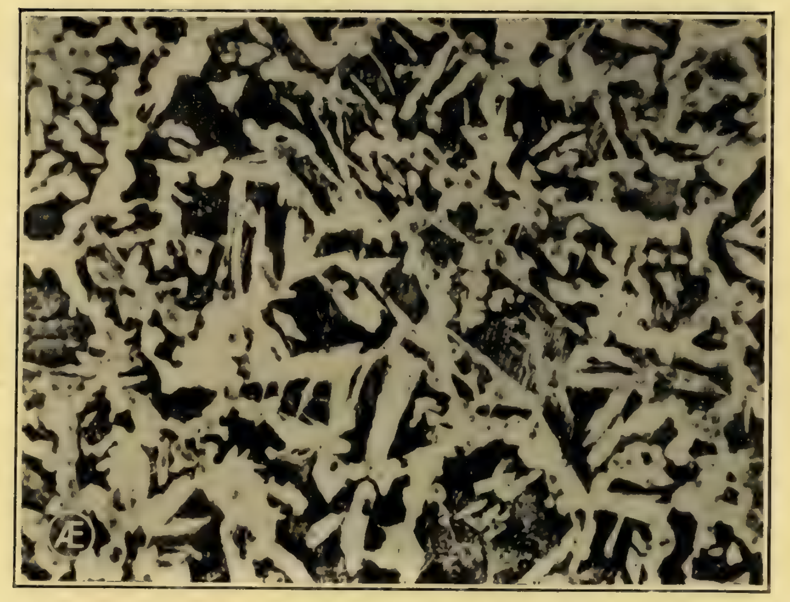

Fig. XII. 0·4 % carbon steel magnified by 150. Showing Ferrite and Pearlite.

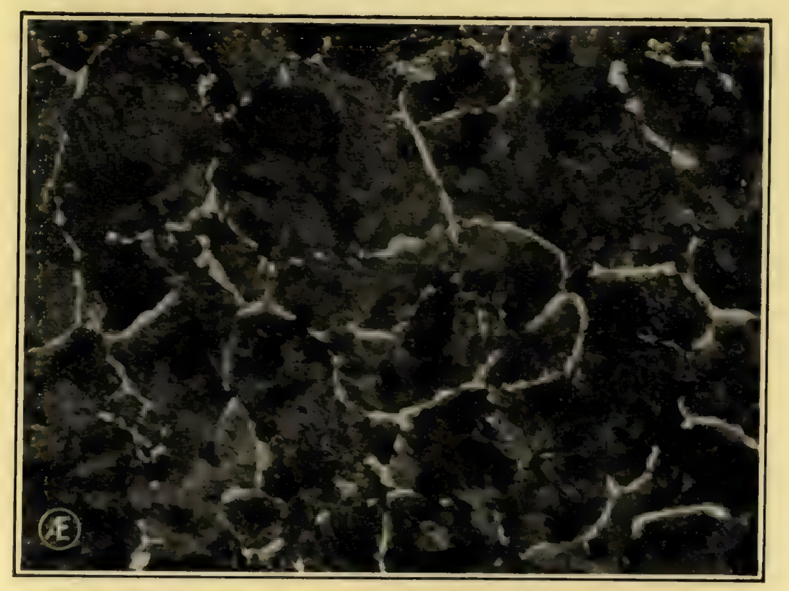

Fig. XIII. 0·7 % carbon steel. Showing Pearlite with a little Ferrite.

Fig. XIV. Coarsely laminated Pearlite in annealed steel.

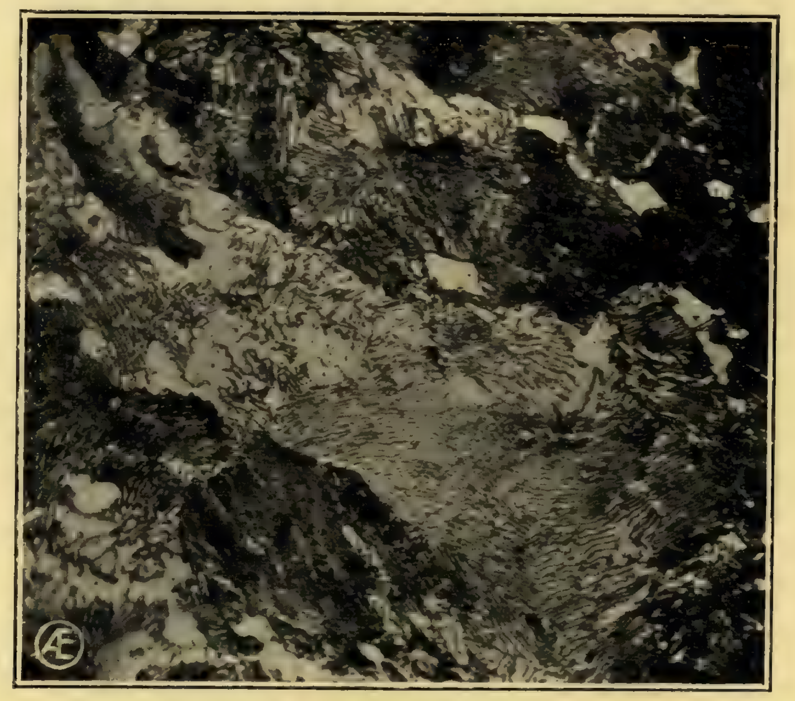

Fig. XV. Finely laminated Pearlite magnified by 800.†

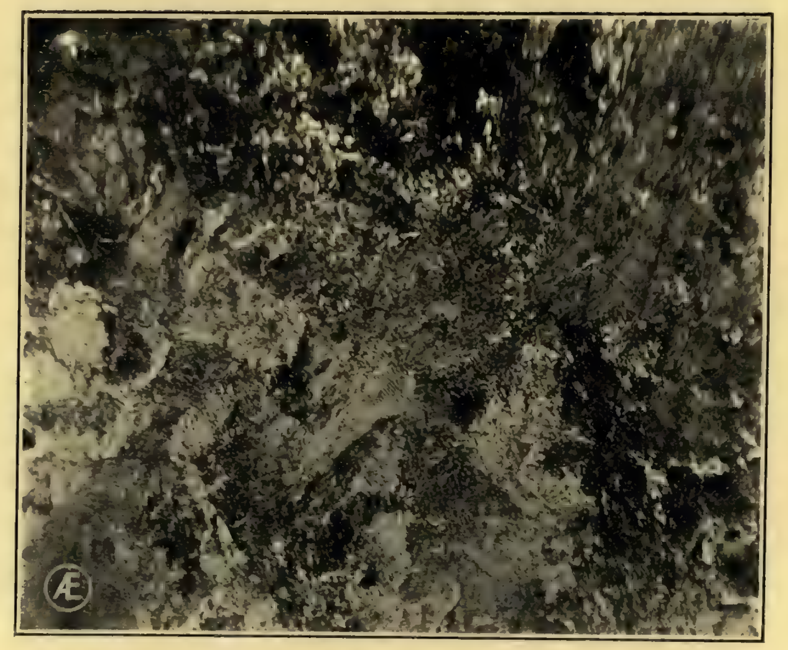

Fig. XVI. 0·9 % carbon steel, consisting entirely of Pearlite.

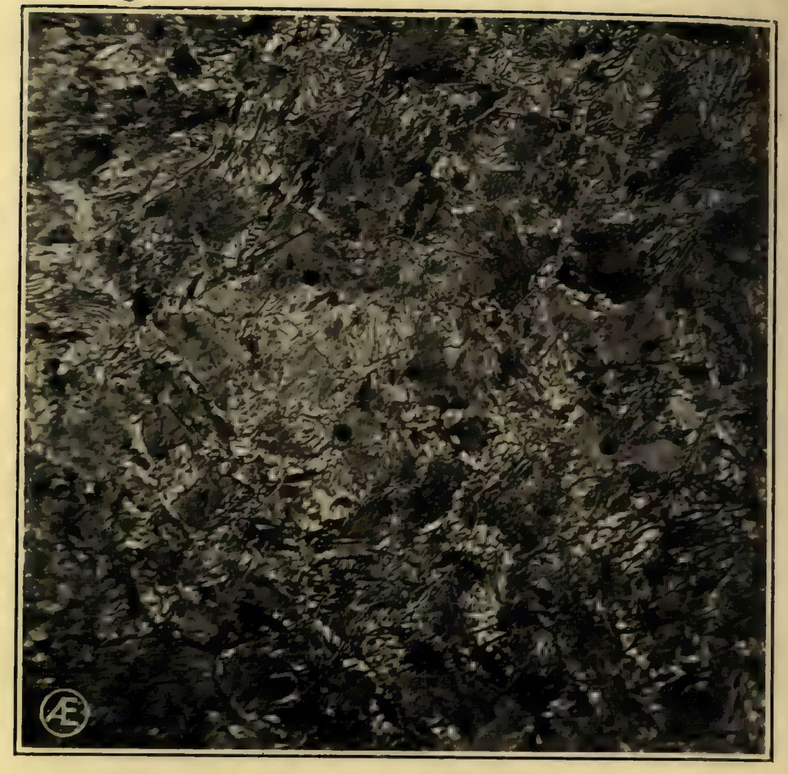

Fig. XVII. 1·4 % carbon steel magnified by 500. Showing a little Cementite embedded in Pearlite.

Fig. XVIII. 2·02 % carbon steel magnified by 500. Showing more Cementite embedded in Pearlite.†

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† These photographs are reproduced by permission of the Institution of Mechanical Engineers, to whom the originals belong.

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Fig. VIII. Swedish wroug
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Fig. IX. Ordinary wrought
large strips

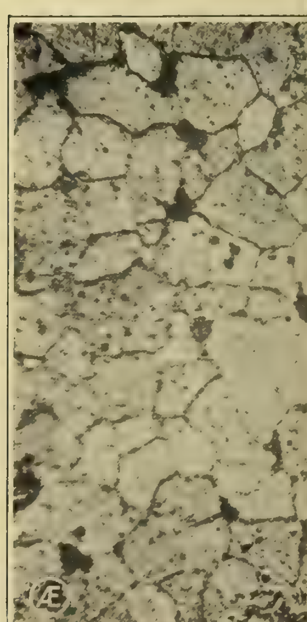

Fig. X. 0'05 % carbon stee
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PLATE II., accompanying article on "Steel," by Dr. Walter Rosenhain.

(See page 485.)

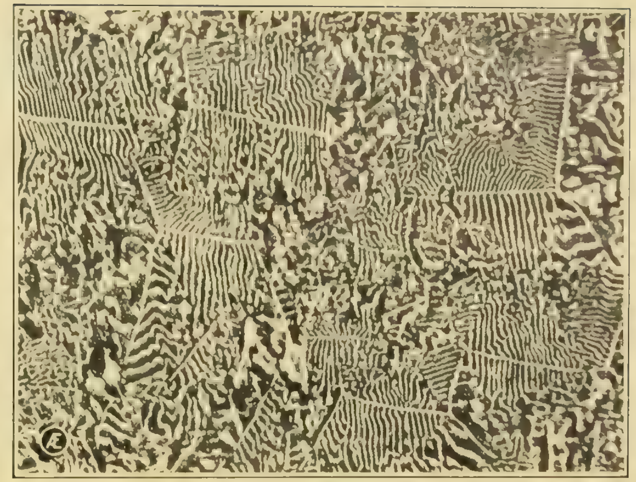


Fig. VIII

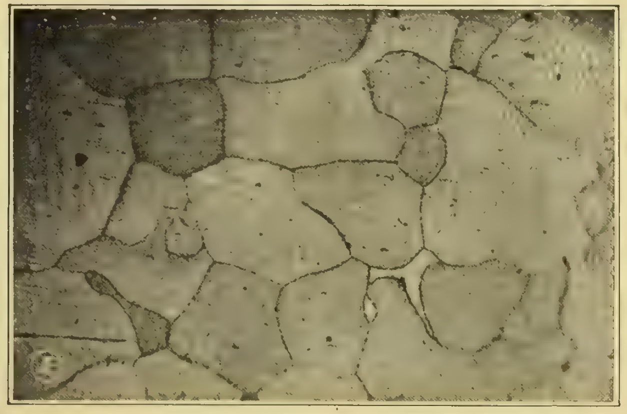


Fig IV.

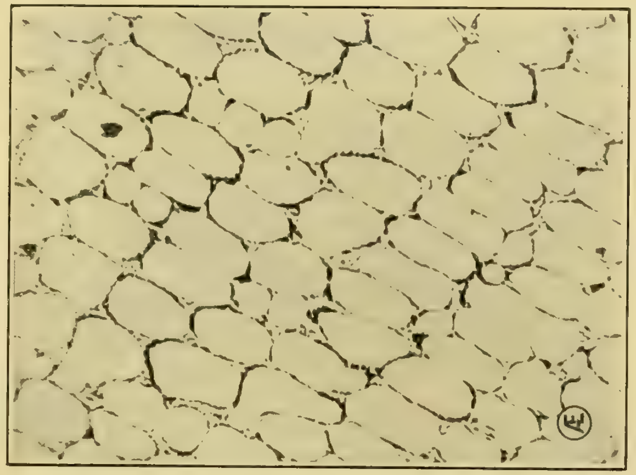


Fig. VI.

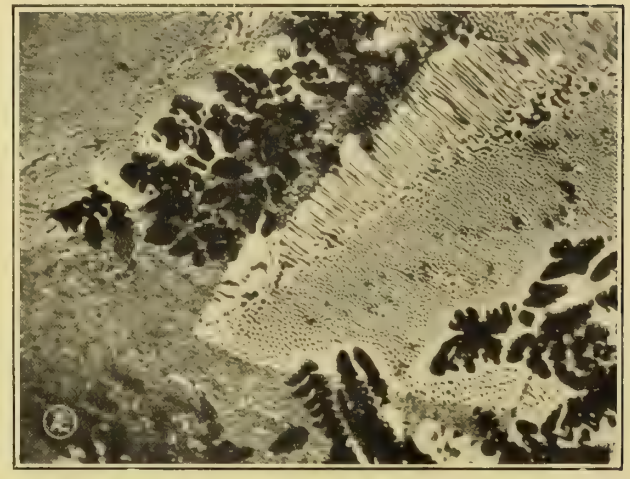


Fig. IX.

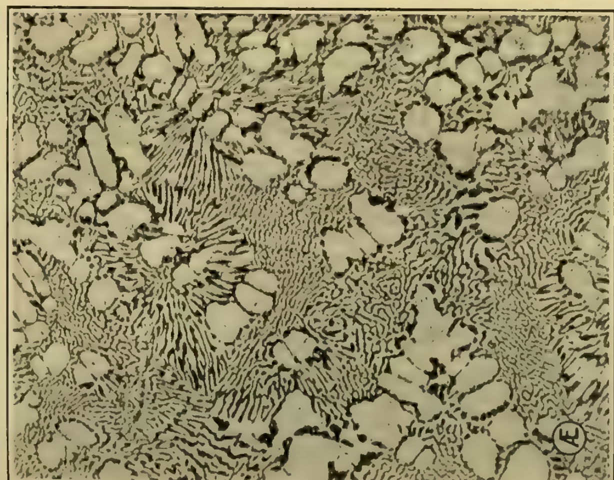


Fig. V.

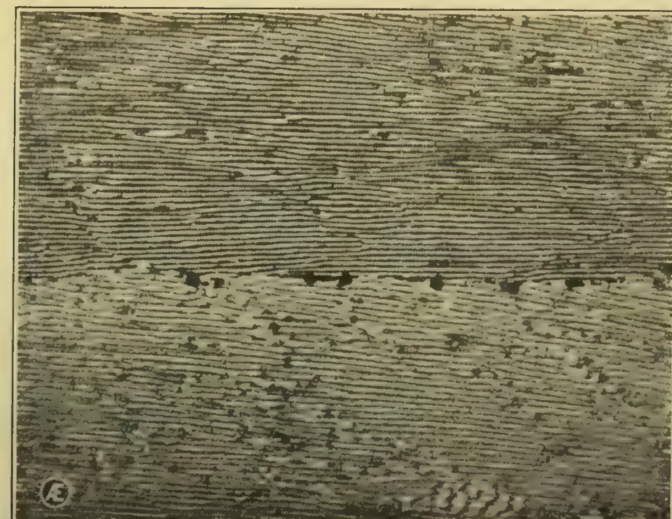


Fig. VII.

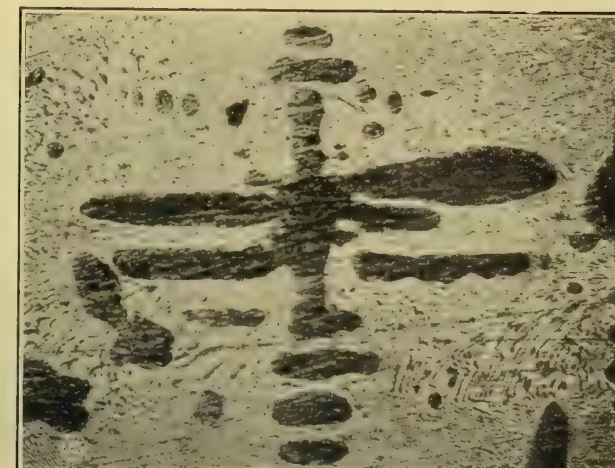


Fig. X.

STEEL.

By Walter Rosenhain, B.A., D.Sc.

THERE are probably few branches of modern engineering construction which make such stringent demands upon the qualities of our materials as those met with in automobile construction. It follows that for the automobile engineer every advance in our knowledge of these materials is of special interest, and this applies particularly to the great forward strides which have been made during the past twelve or fifteen years in our knowledge of the internal structure and constitution of steel in all its forms. In the present series of articles it is therefore proposed to give a brief outline of the more important features of this new knowledge of steel. In doing this it will be necessary to deal with some rather theoretical matters, but an understanding of some fundamental principles is necessary for the proper appreciation of the more directly practical portions of the subject.

The testing of steel, and of other materials of construction, by means of mechanical tests has probably been practised as long as engineering construction itself, and it remains—in its modern developments and refinements—as vital a matter to-day as ever, but, on the other hand, it must now be recognised that mechanical tests alone cannot give all the information which is desirable or even necessary. The indications of mechanical tests are undoubtedly reliable so far as they go, but this limitation must be borne in mind, and it is not always possible to argue from any particular mechanical test as to the probable behaviour of the metal in various circumstances. For this reason the results of recent investigations into the internal structure and constitution of metals are of such profound interest; they give us an insight into the inner mechanism by which metals resist or fail under various forms of treatment, and they give us a clue to the causes for the wide differences in properties and behaviour which are met with in daily engineering practice. More than this, the knowledge of structure and constitution serves as a safe guide to the best forms of treatment to be applied to a given material, and it has also aided the metallurgist in placing at the disposal of engineers those new products, such as alloy steels, which are of such fundamental value, particularly in automobile construction.

This new insight into the structure and constitution of metals has been obtained principally by means of investigations carried out by the aid of the microscope and the pyrometer. The microscope has revealed the internal structure of metals, the presence and arrangement of their various constituents and the manner in which these constituents behave under various forms of mechanical and thermal treatment. The thermal study of metals, on the other hand, by means of the pyrometer, has enabled us to follow and interpret the modes of formation and growth of the various constituents revealed by the microscope and has rendered possible the construction of certain diagrams or constitutional maps in which the whole range of constitution and structure of a series of alloys can now be fully expressed. By the aid of such diagrams the important questions of annealing, quenching, hardening, and tempering can be followed and understood in a detailed and rational manner, which helps to remove those operations from the region of "rule of thumb" to that of accurately controlled and regulated practice.

We shall begin by considering the microscopic examination of steel—similar processes and ideas being, of course, equally aplicable to other metals.

Metals are examined upnder the microscope by using that instrument to look *at* a small specially prepared surface of the specimen in question. The method more generally employed in examining organic tissues or rock-sections, of examining thin sections by transmitted light, is not feasible with metals because they cannot be obtained in sufficiently thin slices to allow an adequate amount of light to pass through them. We are accordingly limited to examination by reflected light, and the first essential for this purpose is the preparation of a perfectly smooth, polished surface. The process of polishing such surfaces is a somewhat delicate one, but it is now widely practised; the surface is rubbed down—preferably by hand—on successively finer grades of emery paper, and finally polished on a pad, preferably mechanically rotated, and fed with some very fine polishing powder, such as rouge or alumina, specially prepared for the purpose. The resulting polished surface is not so bright as could be obtained by burnishing, for instance, but there should be a total absence of microscopic scratches. A surface in this condition seen under the microscope presents a practically featureless blank, unless, indeed, we are dealing with a material which contains non-metallic impurities or enclosures, since these latter are visible even on the merely polished surface on account of their different colour and smaller lustre.

The perfectly polished surface consists in reality of a thin film of amorphous metal spread over the sectioned surfaces of all the constituent grains or crystals of the metal. When such a surface is attacked by an acid or other dissolving reagent the surface film is immediately dissolved away, and then the reagent attacks the constituents which are thus uncovered; the rate at which a given acid, however, acts upon these different constituents varies very widely according to the nature of the various substances present. As a result, some of the constituent grains or crystals are eaten away more rapidly than others, and the acid therefore produces upon the previously flat polished surface a minute pattern in bas-relief or intaglio, corresponding with the various constituents present. In some cases, also, the solution of one of the constituents gives rise to the formation of a coloured film or deposit of insoluble residue, and this film serves to accentuate the relief pattern by means of more or less intense colour effects.

When these etching effects were first discovered it was necessary to approach their interpretation with considerable care, and, in fact, to regard the whole of the patterns as a species of hieroglyphics. The whole matter has, however, been very carefully investigated, and it is firmly established that the patterns seen on etched surfaces accurately indicate the structure of the metal beneath. Thus, repeated polishing and etching of the same specimen constantly shows the same pattern, except for the gradual changes due to the wearing away of the metal by successive polishing; different etching reagents also give substantially the same indications, and—finally—these indications can be readily interpreted from the results of the thermal study of the metal in question.

Before describing in detail the etched patterns found on various grades of iron and steel, we may for a moment consider the microscopes by means of which these patterns are observed. In some metals, particularly those having a low melting point, the structure of the constituent crystals is often large enough to be seen by the unaided eye. An example of this kind is shown in Fig. I., Plate I.,* which shows the etched surface of a piece of slowly-cooled cast lead, full natural size. But such coarse structures are the exception, and in all metals which can be used for constructional purposes, the scale of structure is enormously more minute; in fact, we may safely forestall a point to be explained later—that in general terms, the smaller the scale of structure, the better the material from the mechanical point of view. In iron and steel the scale of the structure is generally such that a magnification of 100 diameters is required to show it at all, and in many cases the highest available magnifications of the modern microscope (up to 1,500 diameters) are barely sufficient to show the structure clearly. The microscopes used for obtaining these magnifications on metallurgical specimens do not differ vitally from those used for other microscopic work, although certain modifications in the mechanical arrangements of the instruments have been found desirable. These arise from the fact that, in viewing an opaque specimen by reflected light, it is necessary to illuminate the surface under examination by means of a beam of light thrown down through the tube, and condensed upon the point under observation by the objective of the microscope itself. The manner in which this is done, by means either of an opaque reflector which covers one half of the objective, or by means of a thin slip of glass which reflects only a portion of the light which falls upon it, while the rest passes through undeflected, is illustrated diagrammatically in Fig. II. (a) and (b). To allow of such an arrangement, the metallurgical microscope is provided with a means of admitting light at the side of the tube, while the tube itself is generally

* The Plate referred to takes the form of a loose supplement accompanying this issue.

kept in one fixed position relatively to the source of light, all focussing being done by moving the stage, which is provided with suitable screws for this purpose.

We will now consider the micro-structures, as revealed in this manner, of a series of typical varieties of iron and steel. For a clear understanding of these structures, however, it is necessary to begin with the structure of pure or nearly pure iron.

When a specimen of the purest obtainable iron is polished, etched and examined in the manner just described, the structure which is seen is that illustrated in Fig. III., Plate I. Provided that the etching has not been carried very far, the

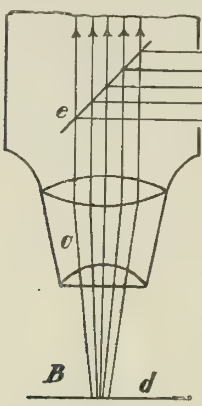


Fig. IIa.

whole field appears of one colour, but it is seen to be divided up into a number of roughly polygonal areas. This structure, it may be remarked, is typical not only of pure iron, but of practically every pure metal. The meaning of this structure is simply that the metal consists of a large number of grains or—more properly, crystals—all of which have the same chemical composition; in a pure metal it is obviously impossible to find two constituents of different composition. But it may be asked, if the chemical composition is uniform throughout, why does the action of an etching or dissolving reagent develop a pattern at all—why is not the dissolving action uniform also The answer is that although the chemical composition is uniform, the physical arrangement is not uniform; in fact, the metal is built up of an aggregate of crystals of iron which have grown together in all manner of shapes and directions. These crystals—owing to the fact that each has been impeded in its growth by the presence of its neighbours—have not developed the characteristic geometrical outlines which we associate with the term "crystal," but they none the less possess the most fundamental property of a crystal, viz., that the matter within each of them is arranged in a definite and uniform manner or "orientation," and that this orientation changes from one crystal to the next. It is a result of these different orientations that the surfaces of the various crystals exposed in our polished and etched section are attacked by the acid at different rates. This difference in rate of attack produces slight differences of level betwen the surfaces of adjacent crystals, and the short, steep slope from one of these surfaces to the next appears under the microscope as a fine black line—hence the polygonal structure seen in Fig. III. If the etching process is carried a little further, particularly if a suitable reagent is used, such as a solution of copper-ammonium chloride, the process of solution takes the form of a gradual un-building of the crystals, and their surfaces consequently become roughened by the formation of very numerous and minute geometrical facets or pits. The inclined surfaces of these pits reflect the light regularly over the entire area of each crystal, and if this reflection is such that a part of the light is sent outside the microscope, that particular crystal appears comparatively dark. The effect as seen in the microscope is shown in Fig. IV., Plate I.; there the specimen is illuminated by "normal" light, i.e., by light falling down the tube of the microscope, and therefore incident "normally" upon the speci-

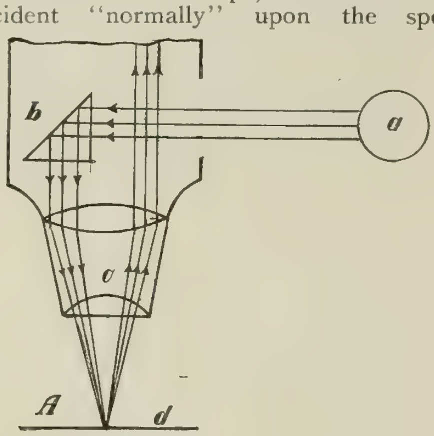


Fig. IIb.

men; but we may also illuminate such a specimen—if we do not require to use too high a magnification—by means of beam of light incident obliquely from outside the microscope-tube, and this kind of illumination will produce a different effect. Facets which reflected the normal light away from the microscope may now be so placed as to reflect the oblique light into the microscope, so that under favourable conditions a crystal area which appeared dark under normal light will appear bright under oblique light, while in many cases the dark crystal boundaries will also be so placed as to catch the oblique light and will then appear as bright lines instead of black ones. An example of this effect of oblique light, which goes a long way to demonstrate the truly crystalline facetted character of the etched surfaces, is shown in Fig. V., Plate I., which represents the same field of view as that already illustrated under normal illumination in Fig. IV. In favourable circumstances it is quite possible actually to see the minute crystal facets which have just been referred to. Figs. VI. and VII. give examples frrom various metals, such as lead and tin, whose tendency towards coarser structure makes it easier to observe what would, in the case of iron, be almost ultra-microscopically minute.

Before leaving the subject of the micro-structure of iron it may be interesting to compare with the micro-structure of pure iron which has just been described and illustrated, the structures actually met with in commercial "iron." Such substances as "cast iron," and even "malleable castings," we must exclude for the present because their chemical composition places them outside the class of "nearly pure" iron altogether. The closest commercial approach to pure iron is, perhaps, to be met with in the best Swedish wrought iron, whose micro-structure is illustrated in Fig. VIII., Plate I. The polygonal crystals are clearly seen, but, in addition, we see particles of a black substance—this is the mechanically enclosed slag or cinder, which is drawn out in the direction of rolling. In ordinary wrought iron of lower quality, the amount of this cinder becomes very large, as shown in Fig. IX., Plate ?. In this connection it is interesting to note that such wrought iron appears to be "fibrous" only on account of these drawn-out bands of cinder, while the crystals of the metal itself are approximately equi-axed, just as in pure iron. The cinder "fibres," however, are a source of weakness and not of strength, so that the virtue of the "fibre" of wrought iron can only be a mythical one.

We may now pass on to consider the effect of the presence of carbon on the micro-structure of iron. The ordinary carbon steels are, of course, essentially alloys of iron and carbon, but they contain, in addition, minor quantities of other substances, such as manganese, silicon, phosphorus, sulphur, etc. Although these substances exert a very powerful effect on the properties of steel, we must disregard them for the present and—for the sake of simplicity and clearness—confine our attention to a series of practically pure iron-carbon steels.

Our first specimen is a steel contaning only a very small amount of carbon—about 0.05 per cent. At first sight it would seem incredible that so small a proportion as one part in 2,000 could be seen at all under the microscope, but, as Fig. X. shows, the presence of the carbon makes itself felt quite visibly in the micro-structure of this steel by the appearance of a number of dark-etching patches on the micro-section. The reason lies in the fact that the dark-etching constituent which we see on the section is not pure carbon, but a constituent which itself contains rather less than 1 per cent. of carbon; consequently a steel containing one part of carbon in 2,000 will actually contain one part in 20 of this dark-etching constituent. In Fig. X. this carbonaceous constituent appears merely as small, dark patches lying in the interstices of the iron crystals. As we pass on to steels containing higher percentages of carbon we find the number and area of these dark patches steadily increase. In Figs. XI., XII., and XIII. we see the structures of steels containing 0.2, 0.4, and 0.7 per cent. of carbon respectively. In the last of these the iron—or "Ferrite," as it is called in the form of a micro-constituent of steel—occurs only in comparatively narrow bands between the dark-etching areas. As these areas themselves become larger and better defined with increasing carbon-content, we notice that they are not uniformly dark-etched, but themselves have a minute structure of their own. If we take steps to render this structure more apparent, for instance, by exposing the steel to prolonged heat and slow cooling, we find that this dark-etching constituent really consists of closely-spaced laminæ of white and dark-etched material. This duplex character is clearly seen in Fig. XIV., which represents such an annealed steel. By the use of higher magnifications, however, this duplex laminated structure of the dark constituent can also be observed without the aid of special annealing; this is shown in Fig. XV. This minute lamination, when observed by obliquely-

reflected light gives rise to a coloured sheen similar to that of mother-of-pearl, and hence this constituent of steel was originally called the "pearly" constituent, and has now received the name of "Pearlite."

When we pass on to consider steels of still higher carbon-content, we find that, with about 0.9 per cent. of carbon, the entire area of the micro-section is occupied by "Pearlite" (see Fig. XVI.); this has sometimes been called the "saturation" point of steel, but its true significance can only be understood when the thermal phenomena of steel are considered. Meanwhile we find that if we increase the carbon content beyond 0.9 per cent. a new constituent makes itself apparent. This appears in the micro-sections (see Figs. XVII. and XVIII.) in the form of white, angular crystals interspersed in the pearlite, and their behaviour on polishing, and under a scratch, indicates clearly that these are crystals of an exceedingly hard, brittle body. This fact alone would lead us to suppose that we were here dealing with a definite iron-carbon compound, and this compound has actually been isolated. It has the empiric formula Fe_3C, and as a micro-constituent it is known as "Cementite."

In carbon steels ranging in carbon-content from 0 to about 2 per cent. we thus meet with three distinct constituents, viz., Ferrite (or nearly pure iron), Cementite (Fe_3C), and Pearlite, and this last is merely an intimate mixture of alternate plates of the other two. Now Ferrite is soft and ductile, but also comparatively weak, while Cementite is very hard and brittle; Pearlite bears an intermediate character, for it is strong and much harder than Ferrite, but it is not ductile, although not so brittle as Cementite. In the steels containing less than 0.9 per cent. of carbon we therefore have a mixture of ferrite with pearlite, and the latter acts as a species of stiffening or reinforcement to the former—with increasing pearlite, therefore, the steel becomes both harder and stronger but decreasingly ductile. A maximum of tensile strength is, in fact, attained when the steel consists entirely of pearlite. With the appearance of Cementite in steels containing over 0.9 per cent. of carbon, the hardness increases further, but the steel becomes increasingly brittle. In this way we recognize the intimate connection between the general micro-structure of steel, its carbon content and its mechanical properties.

(To be continued.)

THE SHORTCOMINGS OF THE AEROPLANE.

Being the opinions of a close observer of the recent "Circuit of Britain" contest.

IN the initial stages of any new form of locomotion the manufacturers are engaged for the most part with purely experimental designs, and the whole construction of the machine can be said to be more of the trial and error type than of ordinary mathematical design. As a result of this, it would not be correct to urge the adoption of ordinary engineering principles during the first few years of the new form of transit.

The recent aeronautical "Circuit of Britain" corresponds to a great extent to the original London to Brighton run of the Automobile Club of Great Britain and Ireland, though it can scarcely be said to have had such a disastrous result as the Brighton run had in the early stages of the automobile industry. In this instance the majority of aeroplanes which started in the race were designed more with a view to satisfying those conditions which are necessary if flight is to become a recognised mode of transit, than from the point of view of making a satisfactory engineering job of the whole machine and taking advantage of that experience which has already been gained in other branches of a similar form of industry. Unlike the Brighton run, the "Circuit of Britain" is not at all likely to have a bad effect on the industry as a whole, since those machines which successfully completed the circuit performed in such a thorough manner that they counteracted the feeling caused by the vast majority of competitors failing to arrive at the Harrogate control. Moreover, aviation is coming upon the public not as a rival to an extremely old and established form of transit, but in competition with yet another type which has already had to fight its own battle, and to a great measure succeeded.

Now that the first serious competition in this country has been concluded, it would be well to urge the whole aviation trade to regard their machines more as an engineering project than one could reasonably suggest beforehand. At the present moment, the majority of firms are bringing out new designs from experience suggested by use of their old machines, and are adopting details without consulting other stages of engineering with a view to finding how far these details are likely to satisfy the existing conditions. It is to be hoped that with an entirely new type of vehicle, the mistakes which were originally made with the motor car will not be repeated, and that full advantage will be taken of that which has been learned, not only with high speed internal combustion engines, but about strength of materials and the particular type most useful for any certain part of the machine.

Overdoing Lightness.

At the present moment there is an undoubted tendency to make the whole machine of extreme lightness, purely for the sake of speed, and it is quite certain that the future lies, not in the direction of an extremely light machine, but rather in that of a heavier and considerably more reliable construction. Engines at present are suffering from a serious attack of over-lightened parts which render them anything but as reliable as they should be, and entail a considerable amount of attention on the part of the owners of machines. Seeing that quite a number of the competitors could have carried two men, it seemed curious that they should not do without this advantage and adopt a more reliable, if heavier type of internal combustion engine. The rotary engine which was used in such a quantity of the aeroplanes seemed to have been kept going to a great extent by the unceasing diligence and dexterity of the mechanics attached to the machines or provided by the manufacturers of the engine. The only machine of which the engine could be said to resemble car practice in the least, had its radiator placed in a position which was long ago found to be most unsatisfactory in connection with road vehicles. It is, therefore, scarcely surprising that the engine in question caused damage by overheating and bursting a water joint, forcing an awkward descent which resulted in the abandonment of the race on the part of the pilots.

Looking at some of the engines entirely from the engineering point of view, it is exceedingly surprising that the automatic inlet valve should still be in such great favour. It has been proved time after time that an engine so fitted will not develop the same power that can be obtained when the valves are operated mechanically, and the corresponding increase of weight will most certainly be worth while in view of the extra power and extra reliability which can be obtained thereby. Again, in road car practice, it has been found to be absolutely necessary that the lubrication should be positive, and, moreover, a certain amount of control over the amount of oil delivered must be in the hands of the driver. Practically every engine in the race relied on a couple or more of drip feed lubricators with all their attendant drawbacks which were generally allowed to deliver very freely to the engine. In every case considerably too much oil was used, and a copious supply of lubricating oil was spread all over the machine through the exhaust ports of the engine. As a natural result, the valves of such engines were in an extremely precarious state after a run of any considerable length.

Absence of Controls.

Very few engines possess an adjustable firing point, a thing which could have been incorporated without in any way complicating the design and could not but have added to the efficiency of the engine. The mixture supply was only correct at one particular point, and the supply of that mixture was extremely crude, generally consisting of the very simplest form of carburettor with a screw adjusting tap for the petrol feed. That such an arrangement cannot possibly lead to economy is at once obvious, and this is another example of a design brought out without any consideration of what has already been done with motor-car engines. The only argument which can be brought against the fitting of a really efficient carburettor, and a control that will at all events give some range in speed, is that weight is saved by the contrivance which has already been adopted. As, however, this is a question of ounces only, it would not seem worth it when the extra power developed by adopting better devices is considered.

There will come a time when economy and power will be very great factors in

aeroplane design, and the comparatively huge engines which are at present prominent on most racing aeroplanes will give place to considerably smaller patterns designed to give the uttermost power for their cylinder capacity, and saving weight by producing a more efficient engine.

While the open exhaust may not as yet form the nucleus of a popular grievance, it would be very much better for the pilot and passengers if all engines were fitted with some efficient form of silencer, or at all events, an exhaust pipe which would conduct the noise away. Beginners have found that the noise of an aeroplane engine has been extremely disconcerting, and hinders that quick brain action, which must be always necessary in a sudden emergency. Later on they may possibly get used to it, but it cannot be said that any particular gain is apparent by discharging the exhaust direct into the open air.

Starting Troubles.

Bearing surfaces as a whole are much smaller than would be dictated by ordinary usage in a commercial machine and, here again, it is certain that the craving for extreme lightness has interfered with engineering design to the consequent detriment of the engine's reliability. On the machines in the race, with one single exception, engine starting was accomplished by pulling round the propeller and switching on when a swing had been obtained. This is not only extremely dangerous, but makes starting considerably harder than would otherwise have been the case, and renders it absolutely impossible for the pilot to start his own engine if circumstances may force him to descend in some out of the way field. As a result of this trouble, a number of pilots lost a considerable amount of time waiting for their mechanics to arrive from the nearest control, and others had more trouble in inducing the ordinary crowd which usually collects round a stranded aeroplane to act as temporary mechanics. The Etrich monoplane overcame this difficulty by attaching a chain and sprocket starting gear to the rear end of the crankshaft in the position usually associated with the large car flywheel, and the ease and comfort with which this engine was started up by the mechanic stationed in the aeroplane was most noticeable. In this machine also provision was made for carrying a man, whose sole duty was to attend to the engine while flying, and for the purpose a miniature engine room was provided, in which were spanners and an oil can, so that small repairs might conceivably be carried out during the progress of a flight, and a considerable amount of valuable time saved thereby. It is a very great pity that the particular machine mentioned should have had trouble at an early stage of the race, and been unable to demonstrate its real possibilities. Probably the future will show all aeroplanes equipped with an engine which can be attended to while the machine is actually in the air, as the stoppage of the engine for any very trivial defect forces a descent within the short gliding range of the machine, and considerable trouble in getting into the air again, unless people with a knowledge of aeroplanes are situated near the spot on which the aeroplane lands.

Another lesson driven home by the race itself was the extravagance of the competitors' engines. Both castor oil and petrol were invariably used in very considerable amounts, rendering the engine extremely expensive to run and adding to the upkeep of the whole machine by the damaging effect of oil spread about on the wing and tail fabrics. Here again, if the light engines were to give place to something more like the highly efficient small bore engines at present fashionable in car design, such extravagance would vanish, and the upkeep of an aeroplane would become more reasonable.

Landing Chassis.

As regards the construction of the machine itself, it is still curious to observe that rubber tyres are almost universally fitted to the landing wheels of an aeroplane. Most, if not all, machines have an extremely flexible spring or rubber device capable of absorbing all the shock that the wheel could reasonably be expected to stand, and the tyres themselves play little or no part in effective shock resistance. With a broad metal rim a machine would be able to start off the ground quite as well as at present, and the expense of fitting special size small tyres would immediately vanish. Another curious development of wheels was the extremely small size of those fitted to the very high speed machines. Both the Breguet and the Nieuport, whose speeds for their respective tyres was very considerable, had extremely small wheels, whose rotational speed before the machine left the ground must have been extremely bad for the bearings, while the strength obtained by the use of smaller wheels seemed to have been negligible, as the Nieuport was put out of the race by the buckling of one of these minute wheels and the consequent twisting of the landing chassis.

For the rest of the landing chassis, the winning monoplanes owe their freedom from trouble entirely to their pilots, as with the exception of the spring held castor wheel, there was no landing chassis worthy of the name on either the Bleriot or the Morane, and in other than the most capable hands it would have been quite easy for a bad smash to have occurred, owing to the machine tipping up on to its propeller. Wires and the vertical wooden struts are still fastened with extraordinary little metal clips, sometimes held by means of a few coarse-threaded wood screws, while, in very few cases were joints of the type which would seem likely to give any lengthy service. It would seem probable that, as new designs are brought out, these small clips will give place to something considerably more substantial and very little heavier. The control still depends on a considerable number of fine cables, which are not always fitted in a manner which will enable them to give perfectly satisfactory service, and for the most part all these cables are allowed to impede the passage of the machine by adding to the wind resistance owing to their vibrations. The Deperdussin got over this difficulty to an extent by enclosing most of its control wires in the fabric covered fuselage, but it would be better if the greater quantity of wires used for diagonal bracing could be done away with. A point which was strikingly emphasised by the catastrophe to one of the competitors at Brooklands was the absence of any protection for the ends of the wings. It is true that it would be extremely hard to devise some contrivance which could effectively deal with the shock caused by an awkward landing partially on the end of one wing, but in many cases, if such a device could be used, the ultimate destruction of the whole aeroplane could have been avoided, and it would seem that a fair amount of flexibility would at all events give a cushioning effect to a blow delivered to the wing in such manner.

One feature of aeroplane engine design which can be commended is the extreme accessibility of most of the parts, this being particularly noticeable when the engines are being overhauled, and it is probably due to the necessity for this complete overhaul that adequate provisions have been made for an inspection of all the running parts with the minimum trouble. It is greatly to be regretted that amongst the actual competitors there were not more machines in which steel formed a very considerable unit in the construction. Had there been a number of Nieuport, Breguet, or R.E.P. machines, some conclusions could have been drawn as to the advantages which can be obtained by suitable steel tube constructions. As it was, trouble overtook the examples of two of these makes at an early stage in the race, and no data is at hand by which one can compare their performance with the flimsier wooden machines which won the race. One of the defects of a metal construction was brought out by the smash to the Nieuport landing chassis, as it was found that the metal parts were so twisted that no temporary repair could be effected in the time available. This difficulty could speedily be overcome if the chassis in question were to be more carefully designed, and, at all events, the trouble was due in the main, not to the metal landing chassis, but to the extremely small wheels used on the machine. It is to be hoped that in any future race held in this Kingdom, or elsewhere, considerable improvements will be visible not only in the engines and methods of starting, but also in the main body and wings of the machines.

Engineering is young in the aeroplane trade, exactly as it was when Gotlieb Daimler and Panhard's first vehicles demonstrated to the world at large the latent possibilities of a new, if crude, form of mechanical transit. Those days were replete with extraordinary rather than engineering constructions, bringing down on the heads of their builders the criticism of elder branches of the great engineering industry, and eventually costing those very constructors endless trouble and much money before the lessons which could be learned from other industries were incorporated in the newer vehicle's design. Gradually the abnormal sank under the rising wave of reason and with it many of the early manufacturers. For a while something like chaos reigned supreme, eventually giving way to saner councils and more suitable designs.

Let us hope that the aeroplane trade will remember, and that every effort will be made to improve the vehicles from well-proved knowledge rather than to evolve the peculiar or rely on freak construction.

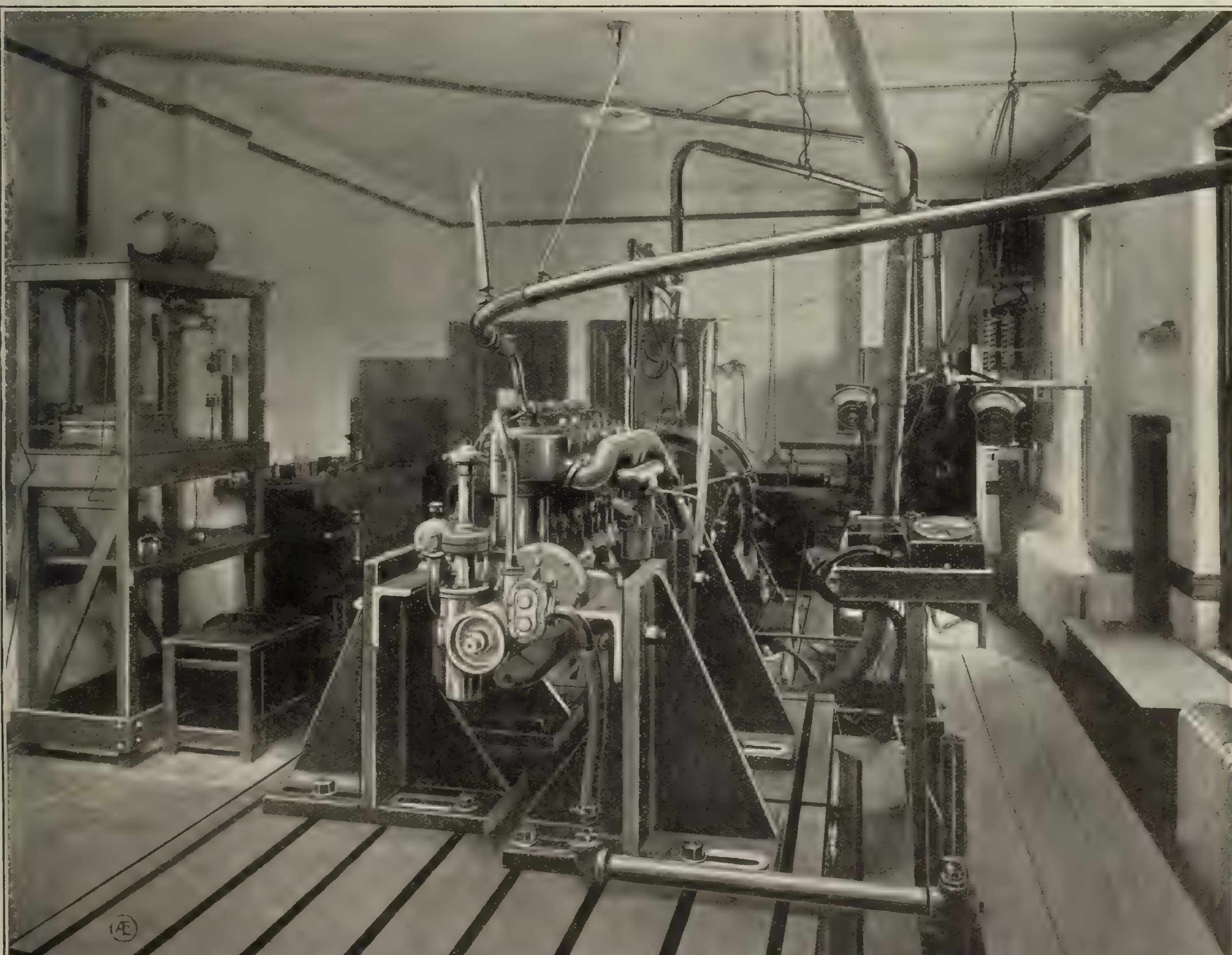
AN ELABORATE ENGINE TESTING PLANT.

The equipment used by the Automobile Club of America in their experimental laboratory.

THE testing equipment of the American Club has been famous for a good many years by reason of a large dynamometer which was installed some years ago, and which enabled horse power readings to be taken from the road wheels of any car. Since the installing of the club in its new premises, however, this elaborate machine has been discarded owing to its lack of accuracy, and its place has been taken, more or less, by the elaborate engine testing apparatus shown in the accompanying illustration. Before describing this in detail it is interesting to observe the method of mounting the plant, bearing in mind the fact that it is situated on the thirteenth floor of the club building, so that vibration is to be avoided at all costs. To secure this end the extremely heavy bed of the dynamometer is not bolted to the floor, but rests in a cast iron box filled with finely broken cork. The bedplate on which the engines to be tested are mounted, is treated similarly, with the result that no tremor is to be felt in the building, even when a badly balanced engine is running at its worst. Curiously enough the bedplates exhibit no desire to shift and so make contact with the retaining boxes, and therefore the cork insulation appears to be perfectly satisfactory.

The dynamometer was built specially for the club by the Diehl Manufacturing Co. The normal capacity of the dynamometer is 64 h.p., and the overload capacity 90 h.p. for a period of one hour, and it is designed to work at speeds between 500 r.p.m. and 2,000 r.p.m. The whole machine is mounted on large ball bearings, and the torque arm is supported by a spring balance, in turn hung from a frame which stands on the base of an ordinary beam arm weighing machine. This can be seen in the background of the illustration on the right hand side, and the purpose of the arrangement is that, while the weighing machine is more accurate than the spring balance, the latter enables rough readings to be made very quickly.

This torque reading together with the speed of revolution is, of course, all that is necessary to determine the power, but a great deal depends upon the accurate determination of the number of revolutions that the engine is making at the instant of reading the torque. Therefore reliance is not placed upon any one means of speed determination. Firstly, as giving a rough idea, corresponding to the spring balance torque reading, there is a centrifugal speed indicator. Additional to this there is an electrical indicator, calibrated to the machine and reading in revolutions, and finally there is a positive mechanical revolution counter of a most ingenious and elaborate pattern. On the tailshaft of the dynamometer there are a series of gear wheels, and above them is a platform carrying a corresponding number of revolution counters, made to resemble a Veeder cyclometer in size and form. Each of these counters has a pinion which can mesh with the corresponding wheel on the dynamometer tailshaft, but each is held on a hinge, so arranged that normally the two gears are out of mesh. Beneath each counter there is a small electric magnet controlling a small armature fixed to the counter, so that switching on the current at once depresses the counter and brings the little gears into mesh. There are a series of



The dynamometer equipment in the testing laboratory of the Automobile Club of America.

push buttons arranged along the side of the supporting board, and the wiring is such that the act of switching in one counter also switches off the counter immediately behind it, and so on. Therefore, by working with a watch in one hand, and the other hand resting on the supporting board, the whole row of counters can be used one after the other. For instance, the number of revolutions can be recorded for the first, second, third and fourth quarters of a minute with very great accuracy, thereby showing whether the engine is running steadily or not.

Next, with regard to the measurement of fuel consumption, there is another ingenious and automatic device. The supply tank is carried on the bed of a weighing machine, itself set in a wood frame, as can be seen in the foreground on the left hand side in the illustration. The beam is graduated in pounds and tenths, and reads by estimation to hundredths. Readings are taken by measuring the time for a definite number of pounds to be consumed, rather than the number of pounds consumed in a definite time, the former method requiring less attention by the operator, and being facilitated by an electrically operated stop watch. The scales are wired to a battery in such a way that a point on the scale beam dips into a cup of mercury when the scale beam falls, thus closing a circuit which rings a bell and stops the electrically operated watch. A reading is taken as follows :—

Suppose that just before starting a test, the tank and contents weigh a little over 54 pounds. With weights and rider set at just 54 pounds, the beam stays up. As soon as the weight decreases to 54 pounds, the beam drops, and at that instant, the watch is started by hand. The rider is then slid back to 53 pounds, and the beam rises. The observer closes switches in the bell and watch circuits and goes about taking other readings. When a pound of fuel has been consumed, the beam drops again, completes the circuits, and thus starts the bell ringing (to attract the attention of the observer) and stops the split second hand of the watch. The observer then sets the scale beam for 52 lbs., enters the elapsed time on the log, snaps the split-second hand forward into step with the main hand and goes about his other work until the bell rings again and the operation is repeated.

For regulating the temperature of the cooling water, a hundred gallon tank is provided, into which the water forced through the jackets is discharged. From this tank, it flows back to the engine by gravity, through piping of ample size. As the hot water from the engine enters the tank, it flows round a small perforated pipe through which cold water is supplied from the city mains, in sufficient quantity to cool the hot water from the engine to any desired temperature. An overflow on the tank takes off the excess water, while thermometers in the inlet and outlet lines record the water temperature.

The exhaust gases from the engine are conducted to the four-inch opening of a 12-inch "Y," through a telescoping joint, two flexible joints and suitable piping. The joints allow for heat expansion without straining the exhaust pipe, and give a free sweep to the gases entering the "Y." In the latter, they expand and are cooled by a jet of water, the result being a contraction of volume and silencing. Pet cocks, one near the pipe and one in the "Y," may be connected to a manometer for measuring the back pressure.

A pyrometer for measuring the temperature of exhaust gases is a part of the laboratory equipment, and is most useful for indicating changes in mixture quality, or whether all the cylinders are firing.

The testing plant is used very considerably by manufacturers for testing new engines, and tests are conducted with a despatch which serves to encourage the use. An example of a test sheet heading is shown on this page, and indicates the very full information supplied to a maker whose engine is tested in this splendidly equipped laboratory, while the writer, having witnessed a test, is confident that the observations are entirely careful.

THE AUTOMOBILE CLUB OF AMERICA.
TESTING LABORATORY.

Page.........
Log Sheet No. ....

TEST OF

Run No....... of Test No. ......

For details of test see Page.................., Note Sheet No....... ....

Date of Test........................

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
Number of Observation.	Time. H. M. S.	Speed Indicator.	Counter	Revolutions per Minute	Dynamometer Force in pounds at ....ft. radius	Brake Horse-power of Engine.	Armature Voltage.	Armature Current.	Armature Kilowatts	Temperature of Jacket Inlet Fahrenheit	Temperature of Jacket Outlet Fahrenheit	Temperature of Room Fahrenheit	Time of Fuel Weighings H. M. S.	Weight of Fuel Tank and Contents Pounds	Weight of Fuel per hour Pounds	Weight of Fuel per Brake Horse-power-Hour Pounds

Examples of test sheet headings.

THE SIX-CYLINDER DENNIS-GWYNNE FIRE ENGINE.

A speedy chassis for its weight and power.

THE vehicles which from time to time have formed the fighting appliances of fire brigades are always interesting, and have attracted an unusual attention from the days of the hand-operated swivel nozzle manual to the latest motor-propelled vehicle. At the present moment the high powered, self-propelled machine is undoubtedly proving its mastery and, with the coming of a rotary pump, threatens to become the standard type for regular and local brigades.

Its advantages are enormous since, not only are slow horses eliminated, but less ground is needed for the more compact vehicle and, in smaller stations, "turn out" speed is increased very greatly. At larger stations the adoption of the American idea in station work keeps the horses at their post for a certain stretch of time and lessens the time advantage of a "turn out," but neither here nor in the smaller stations can the road speed be equalled by the horse-drawn engine, indeed the comparatively light "ladder" cars are unable to hold their own and the engine consequently arrives before its "ladder" crew.

As a result of the peculiar conditions we find serious efforts to construct a motor-propelled engine which shall be an all-round, useful unit in fire brigade work. Such a machine is the Dennis-Gwynne fire engine hereinafter described.

The six-cylinder White and Poppe engine, bore 127 mm., stroke 130 mm., is rated at 50 h.p., with a normal engine speed of 1,000 r.p.m., this speed being maintained during pumping as well as driving. Fig. I. gives a clear general idea of the engine arrangement, and shows the immense overall length, made necessary by the provision of a separate bearing between each of the crank throws. Separately cast cylinders are used with a 6 mm. wall thickness, except at the corners of the combustion chamber, where an additional 4 mm. is used. Water spacing is well carried out, especially near the valve pockets, at which point care has been used to prevent an alteration in section affecting the expansion or retaining injurious heat. The valves are 10 mm. in diameter—larger in comparison than those for most small engines—and are wisely provided with detachable valve guides of considerable bearing surface. One point of trouble has been eliminated as the usual cotter gives

place to a friction cone device extremely easy to manipulate. A roller-ended adjustable tappet with a large piece of fibre at the contact end is housed in the aluminium base chamber and, being of a plain, straightforward design, should give no trouble. In order to remove the tappet gear complete it is only necessary to unscrew the two bolts seen at either side of the tappet guide, when the whole can be withdrawn.

Separate cams are pinned to their shaft, a somewhat expensive method probably used because the naturally limited demand for such an engine renders it unwise to design suitable dies.

Plain phosphor bronze bearings carry the camshaft, which is driven from the forward end of the crankshaft by a steel spur gear, noisy running being of no great consequence.

One of the most unusual points, and at the same time one of the most interesting in view of the efforts made of late to obtain rotational balance, is the crankshaft on which each web has an extension piece to which a crankpin balance weight is attached. Thus there is little chance of a disturbing effort due either to crankshaft whip or to want of rotational balance, and the engine should be remarkably sweet in running. Bearing surface is ample, the forward journal running in a white-metalled bearing 78 mm. long by 56 mm. diameter, and the intermediate bearings being 62 mm. in length by the same width. Each of the white-metalled big ends is of the same dimensions as the intermediate crankshaft bearings.

H section connecting rods are bushed at the small end to take a gudgeon pin 21 mm. in diameter, while the length of the bush is 61 mm. These bushes depend to a great extent for lubrication on a drilled hole immediately opposite the piston crown, a method which is open to all the disadvantages of splash lubrication and, owing to the size of the hole, is likely to allow the ingress of carbon deposit from the piston walls.

The cast iron piston is mainly remarkable for the number of rings allowed contact with the cylinder walls. Although allowances must be made for an engine not primarily designed for high speed, yet five rings would seem many too many, three, counting the lower scraper ring, being ample. Each end of the gudgeon pin is rounded, and the pin itself is located in position by one of the five rings mentioned above. It is possible that there is something to be gained by drilling out the centre of the gudgeon pin, thus saving a certain amount of wasteful reciprocating weight, a quantiy which, when deleted from all six pins, might be quite appreciable. The radiating pins under the piston head should be noticed.

The lubrication arrangement is distinctly unusual, as two rotary pumps are employed, one forcing oil to the bearings from a brass tank fixed on the frame side, while the other sucks from the base chamber and supplies the tank. Although the function of each pump is separate and distinct, both are housed in the same casing, which is bolted to the bottom of the base chamber and is driven from the camshaft by a long rod, which is socketed to allow the pumps to be detached with ease.

As on vehicles designed purely for touring, such an arrangement is of a necessity inaccessible and capable of some improvement, and it is altogether strange that advantage could not be taken of the many excellent arrangements which combine accessibility with unimpaired efficiency. Only the journal bearings are supplied with oil under pressure, as a scoop fitted to the big end is relied upon for the delivery of oil to that bearing, and sufficient splash to ensure adequate lubrication for the small end.

An interesting lubrication safeguard may be observed at the junction of the pump delivery pipe with the journal bearing lead. Here a double ledge has been cast in the aluminium which collects sprayed oil and guides it to the bearing, together with that supplied from the pump. In the event of the failure of the latter organ splash alone will keep the engine bearings lubricated until such a time as a repair can be effected, albeit such an arrangement negatives the pump pressure. One particularly striking point, indicative of careful design, is the size of the oil leads, which are no less than 10 mm. in diameter, precluding any obstacle from choking the delivery.

Circulation is by pump, each cylinder discharging to its neighbour via the cast aluminium pipes seen connecting the cylinders at the top. This allows the sixth cylinder to obtain water already heated by passage round the other five, and is scarcely as well arranged as one would expect. On occasions all water can be drained from a tap placed in a boss near the lowest part of the jacket, and two other similar bosses closed by 20 mm. hexagon plugs allow access to the jacket for core removal, cleaning or for inspection.

As the engine will have to run continuously while the vehicle is stationary it would be liable to overheating but for an ingenious system whereby water is taken from the turbine fire pump and delivered to the cylinders via half-a-dozen turns inside the lubricating oil tank and the radiator, ensuring both a cool jacket and a supply of cold oil, excellent points which must add to the efficiency enormously.

At the forward end of the engine a gear train drives the magneto spindle, through the usual unequal toothed dog clutch, which allows easy timing adjustment, while spindle lubrication is provided for by drilling drain holes in the casing and allowing surplus oil thrown off the gears to dribble into gutters in which these holes are drilled. At the forward end of the spindle there is a pulley for driving the fan.

Auxiliary accumulator ignition is used as a stand-by in the unlikely event of magneto trouble, and accordingly a distributor is mounted at the centre of the engine and driven by an extension of the oil pump shaft, thus bringing it to a readily accessible position.

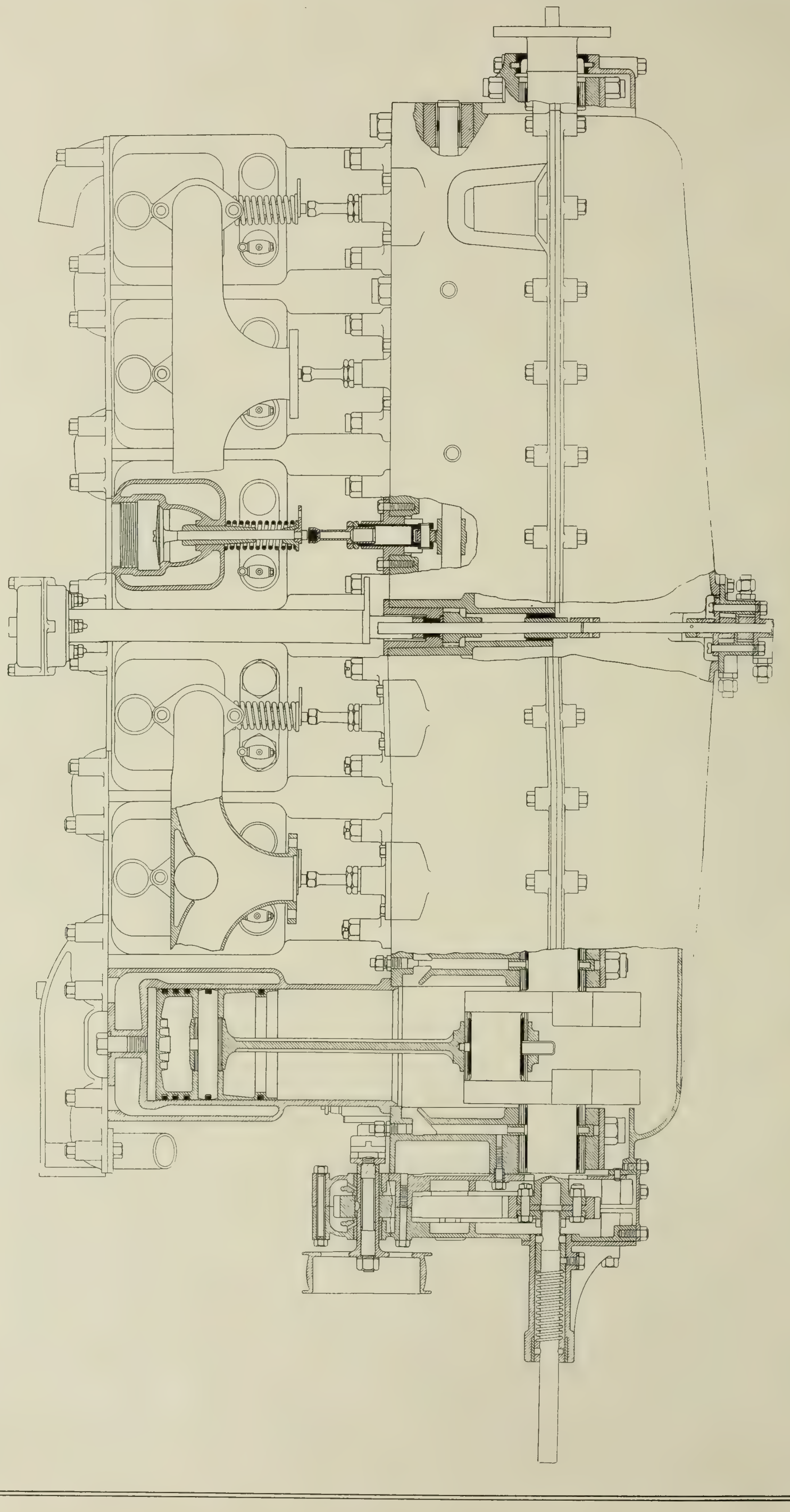
A White and Poppe carburettor with an additional cone-shaped air inlet supplies gas through an induction pipe 50 mm. in diameter, looped in such a manner that an approximately equal suction is maintained by each cylinder.

On the left side of the engine is the exhaust pipe, divided and fitted with baffles as shown in Fig. 1. This conducts the exhaust to a silencer slung below the frame at the rear by a couple of pedestal type brackets.

Between the engine and pump coupling gear an ordinary Hele-Shaw plate clutch is fitted with the standard form of cone-shaped fibre-to-metal clutch brake. Fig. II. shows a sectional view of the spur gearing which is used to connect the turbine pump to the engine, while the vehicle is stationary during service, and, on the chassis plan, Fig. IV., will be seen an external view of the aluminium case which covers the spur gears in question, and also the short lever moving in a quadrant which operates the gears. In the sectional view the upper shaft is connected with the turbine pump, and the lower shaft is coupled directly to the Hele-Shaw clutch. A striking lever shifts the upper gear along its squared shaft into mesh with the lower, which being mounted also on a square shaft, takes the initial drive from the engine and transmits it to the geared up turbine at 1.4 to one ratio. One of the most noticeable points of this particular gearbox is the cone washer which is fitted underneath the ball bearings at the forward end of the turbine driving shaft, this providing a simple method of assembling with heavy type bearings.

Wisely, the shafts have been made of large diameter, with their bearings as close together as the design of the sliding gear will allow, and as the bearings have a considerable diameter no trouble should occur during actual service. The only points which one might criticise are the somewhat inadequate felt washers relied upon to prevent the escape of surplus oil from either of the shaft bearings. It would perhaps have been better to have provided a better joint than the De Dion type shown in the illustration, as fitted both to the turbine shaft and that taking the drive from the pump gear to the change speed gear. The latter shaft, however, being extremely short is hardly likely to have to deal with the same movement that would take place in the longer pump shaft. It is therefore only on the top universal joint, which does most of the work, that a real amount of play or wear is likely to take place.

Turning now to the gearbox, Fig. V., the length of unsupported shafting is at once apparent, and is rendered additionally curious because the touring vehicles manufactured by the same firm show all the modern tendencies towards an extremely short, robust gear shaft. Sliding gears are mounted on a five-splined shaft and operated by the usual form of striking lever working in a groove shown in the sectional view. Direct drive is obtained by a somewhat unusual form of square tooth dog clutch, the secondary motion shaft remaining in mesh during the time the direct drive is in operation. The reverse is obtained with the two pinions, to be seen immediately behind the secondary motion shaft, which mesh with the gear seen on the extreme right of that shaft and the third speed wheel. Care has been taken to space the ball bearings on the forward end of the driving shaft so that stresses are more equally divided between the two separate ball races, but it would perhaps have been better to house these bearings in gunmetal linings rather than in the aluminium of the gearbox casing; however, in a vehicle of this description it is not a matter of so much importance



THE WHITE AND POPPE ENGINE OF THE DENNIS-GWYNNE FIRE ENGINE.

Fig. 1.

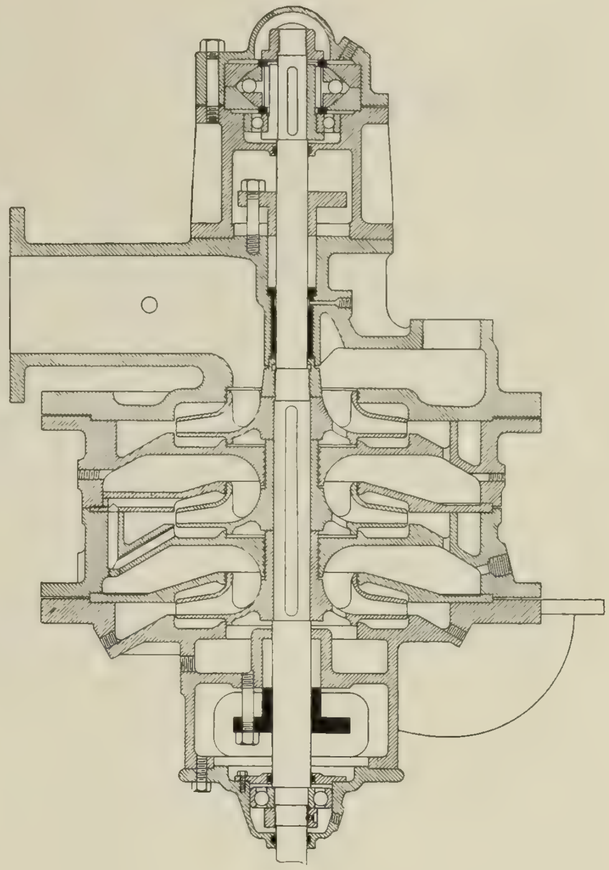


Fig. VIII.

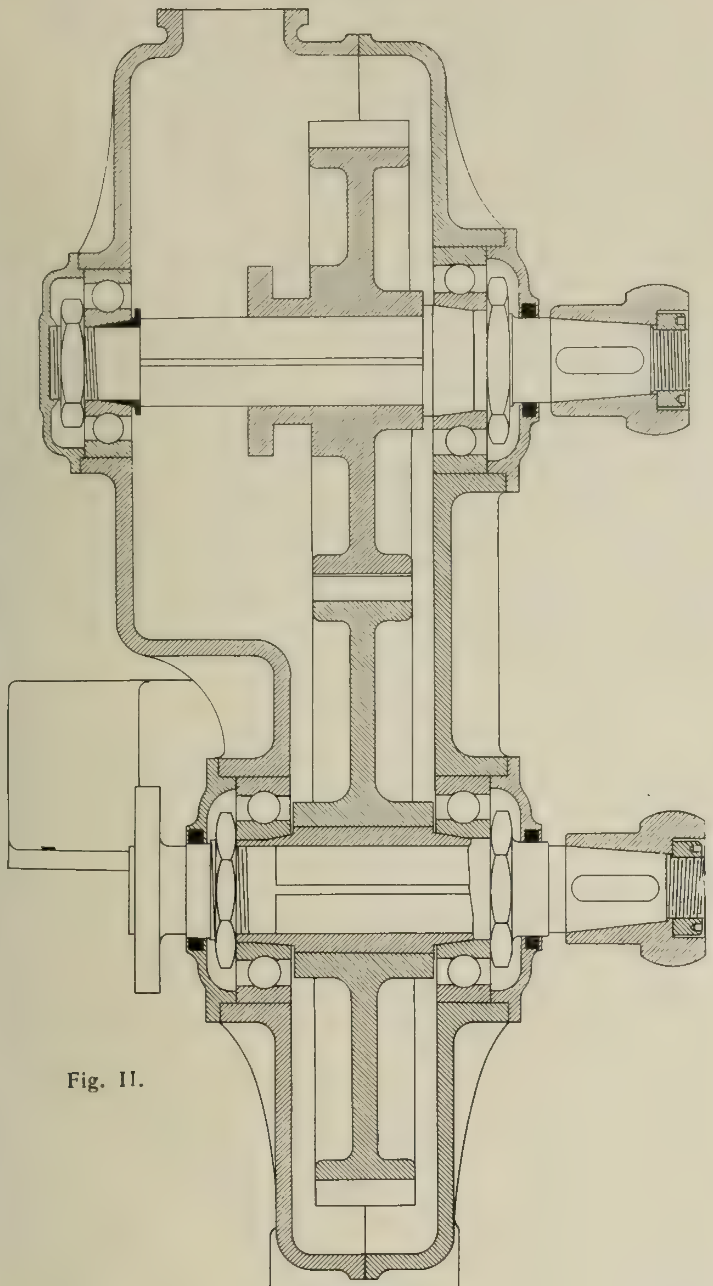


Fig. II.

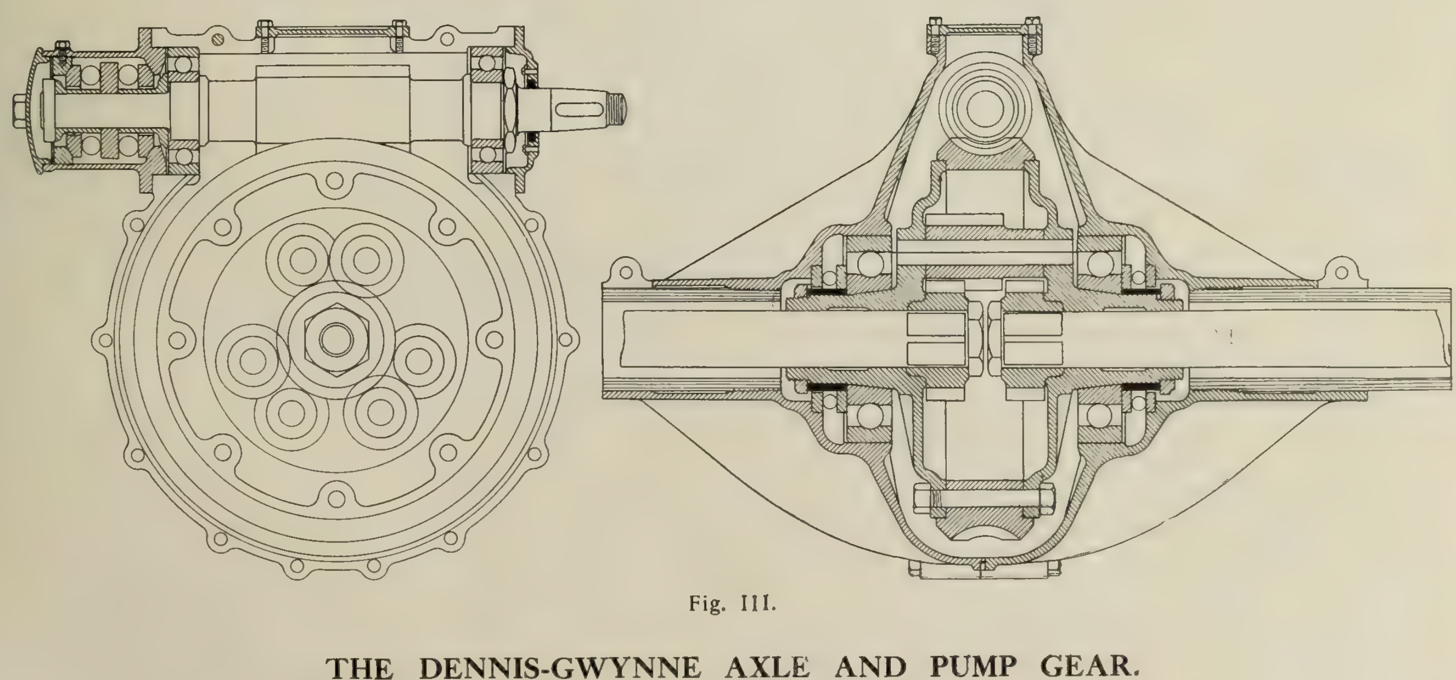
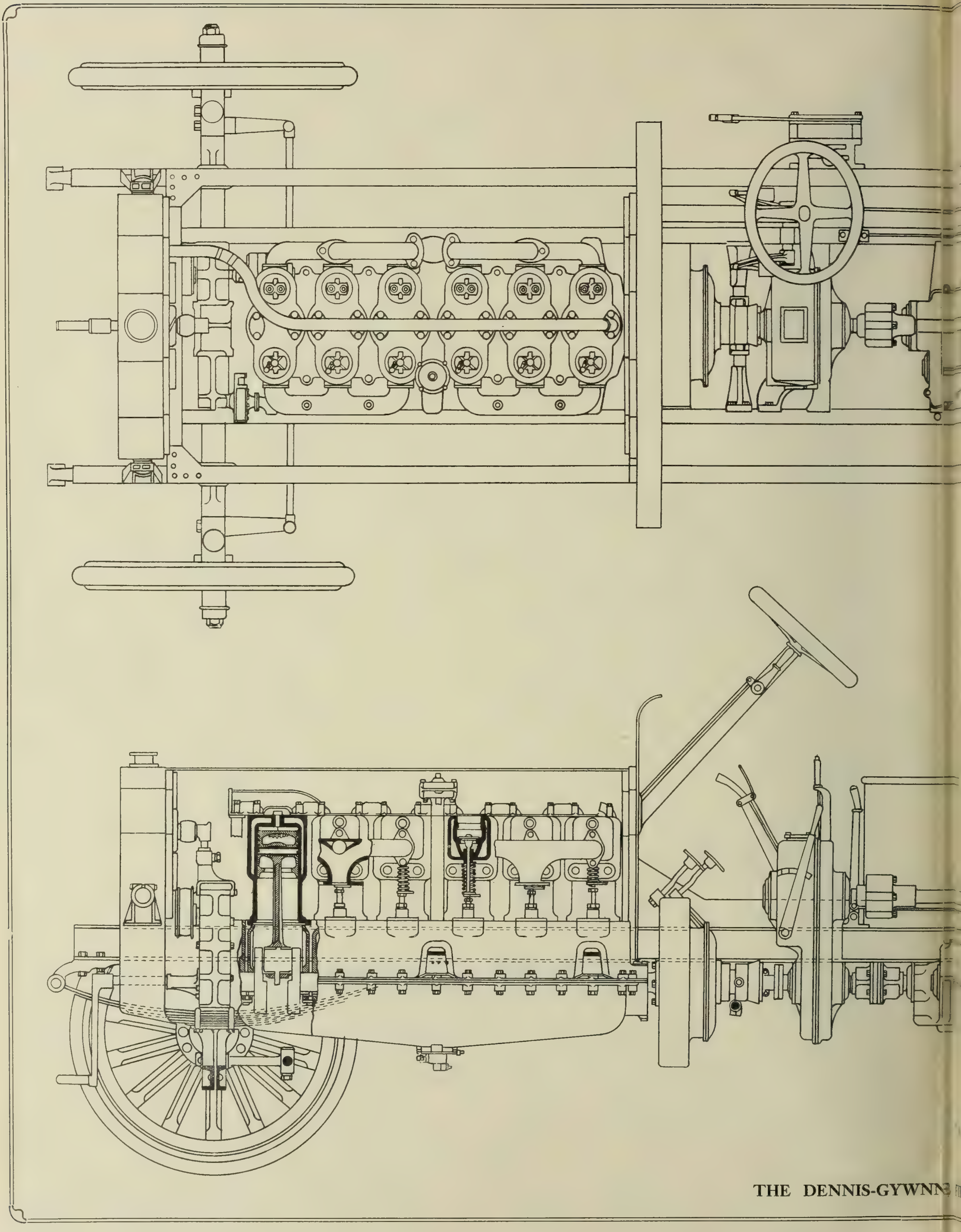
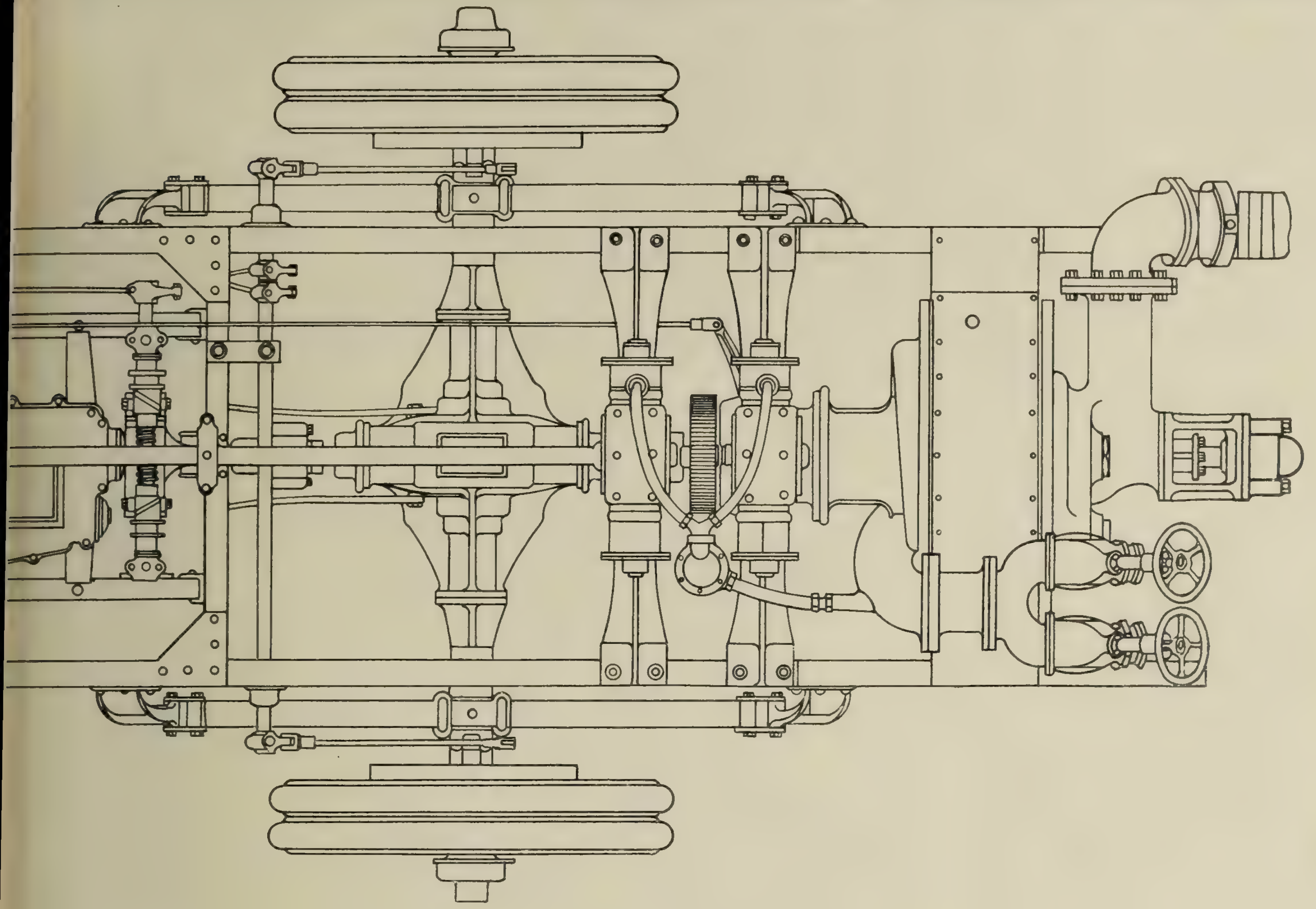


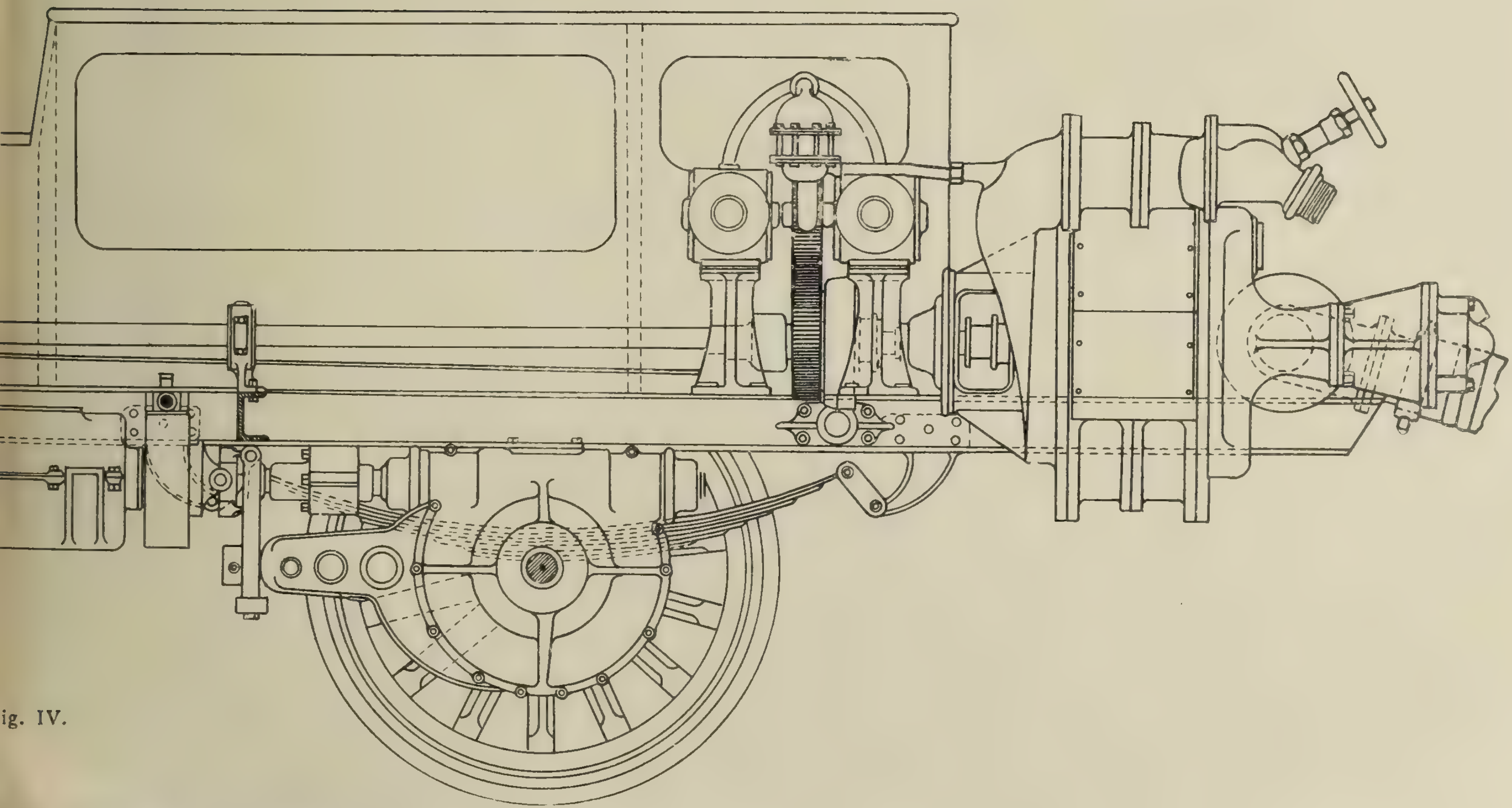
Fig. III.

THE DENNIS-GWYNNE AXLE AND PUMP GEAR.



THE DENNIS-GYWNN





ig. IV.

IRE ENGINE CHASSIS.

as for other heavy cars. A curious feature of the direct drive is the very large ball which is used to take the thrust, and which is placed in a groove at the front end of the first motion shaft. Ball bearings of considerable size are used for the spigot end socketed inside the driving gear. The set pin and lock nut seen on the lower half of the gearbox are a somewhat unusual means of locating the secondary motion shaft, this being accomplished by the curious washer and spigot which may be seen at either end of the shaft in question. No particular care has been taken to prevent the ingress of minute steel particles to the journal ball bearings and the gearbox and it is probable that the life of these bearings could be considerably prolonged had such a safeguard been originally fitted, but, in any case, a vehicle of this description is not required to give the continuous and excessive mileage demanded from the majority of ordinary commercial vehicles, while it is looked after with much greater care. The felt washers, which are supposed to prevent the escape of oil, are distinctly on the small side, and it would be very much better if the cap at this point had been prolonged and a double washer inserted, even if the provision of the usual thrower ring would not have been better than any felt washer at all. It must be remembered that in course of time at least a quarter of the oil contents of the gearbox usually escape via the felt washers, and that a replaced felt washer is seldom a satisfactory job. At the present moment there is a tendency to underrate the seriousness of such a problem, but when the upkeep of vehicles has to be kept to the very lowest, and they are competing with other machines, such little points as these count very greatly in favour of the machine on which they are adopted.

Keyed on a taper at the rear end of the first motion shaft is the brake drum for the external contracting brake which is toggle-operated and spring-returned in the approved touring car manner. Care has been taken to provide a very large plug at the bottom of the gearbox in order that the whole may be cleared of lubricant and any dirt, or other foreign substance removed without an excessive amount of trouble. A plate held by four butterfly nuts covers the top of the box and allows a fair degree of accessibility.

In the end sectional view of the box, Fig. V., a clear view can be obtained of the lever action which brings the reverse pinions into mesh with the ordinary driving gears, this being of a somewhat unusual form, giving a greater movement of the gear for a comparatively small movement on the part of the striking arm. In the same figure can be seen the spring plunger, which locks each striking gear rod by falling into slots which come opposite the plunger at the moment that the gear is successfully meshed. Another device is used to prevent two of the gear shafts moving at one and the same moment, and thus locking : the gear control rods enter the box through a very long bearing, which has been happily designed to take a convenient form of stuffing box, thus obviating one of the most serious points at which oil can escape from the gearbox and play develop which may have a detrimental effect on the change gear mechanism as a whole. It will be noticed in the same view that each arm of the gearbox is cast as a [-shaped girder, thus forming a double flange by which it can be bolted to the frame on which both engine, turbine driving gear, and gearbox are carried. It is conceivable that such an arrangement would lengthen the lining-up process to a great extent, as there is rather more surface to be removed before any canting action can take place in order to correspond with a setting of the engine, and it would probably be better to adopt a similar form of arm to that shown in the side view of the engine. From the brake drum an ordinary universal joint is attached to an extremely short shaft, which ends in a De Dion coupling and the worm-drive shaft. This worm gear, which is shown in Fig. III., is of the type usually associated with the Dennis touring vehicle from the first few years of their manufacture, and has been in successful operation in the same form from then until the present day.

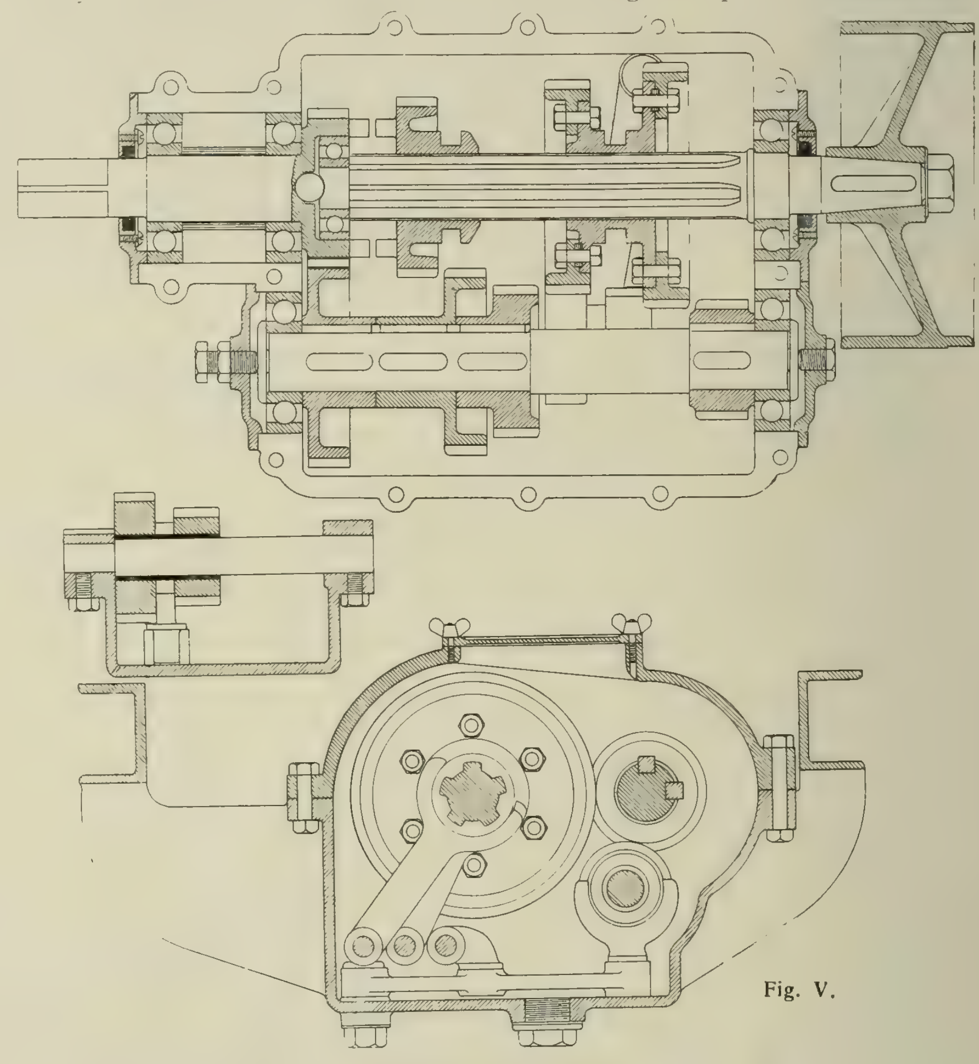


Fig. V.

A straight-sided worm is used, mounted in large ball bearings with an adjustable double thrust bearing of liberal proportions housed at the tail end and readily accessible by a screw cap and lock nut. By removing the set pin shown, forward movement of the internal washer adjusts the bearing through the medium of the outside race. It will be noticed that a distance piece is fitted to the spigot at the end of the worm shaft, which positions the centre race of the thrust bearing, and also that special care has been taken to allow the free access of lubricant by means of slots cut in the casing itself. Above the worm is a small plate, more for oil-filling purposes than for any real inspection of the driving gear, although a certain amount can be seen through the opening in question. Here again, the provision for escaping oil is not as good as one would expect, and it seems hard to believe that the felt washer provided should be suitable for a design of this description. A phosphor-bronze worm wheel, considerably wider than usual and giving a 6 to 1 ratio, is bolted to the differential case by eight 16 millimetre bolts, and, contrary to expectation, the differential is of the spur pinion type, having six pinions carried on very long spindles and provided with quite a lengthy bearing in the side of the differential case. The driving shaft pinions are themselves bolted to the square end of the shaft instead of the more usual taper and key fixing being used. One would imagine that the ordinary bevel pinion differential would be considerably cheaper both to machine and to manufacture, but this is largely a case of exactly what design the shops' machines are more readily capable of dealing with. The case runs on two extremely large ball races which should give ample bearing surface for anything which the axle may be called upon to withstand. Thrust bearings on this do not appear to be as large as usual, but care has been taken to provide them with a suitable packing washer held to the differential case by a large nut. Neither worm nor worm wheel can be extracted without taking adrift the axle casing and, contrary to usual custom, the driving shafts cannot be withdrawn through the wheel hubs, although the axle is of the type known as fully floating.

Turning now to the axle case itself,

this is a very large and comparatively complicated malleable iron casting. The arrangement of the floating drive is distinctly interesting, taking the form of an internal pinion which meshes with the hub cap and drives through this to the wheel flange. As usual in most heavy commercial vehicles, a thin floating phosphor-bronze bush carries the whole weight of the vehicle, and large holes are drilled through it in numerous places in order that the oil may obtain free passage

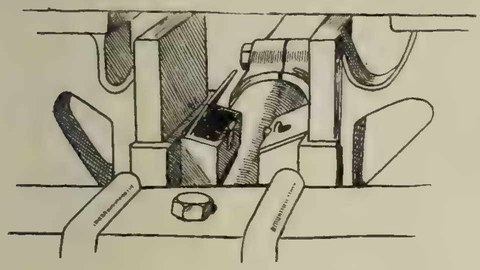


Fig. VI.

to all parts where friction is likely to occur. The expanding brakes operated by the side lever have remained practically unaltered since the original expanding brake was incorporated in the design of Dennis cars. Care has been used to arrange that the fulcrum joints shall be kept a considerable distance apart, a point which has been neglected in the design of a good many brakes in the past, but which considerably increases the power of the brake. Care has been taken to provide lubricators for the small bearing through which the operating arm moves the toggle joint responsible for the expanding of the brake, and a sheet metal cover prevents dirt or mud thrown from the wheels entering the brake drum and destroying the brake surface, whilst there is no possibility whatever of oil leaking from the wheel bearings, rendering the brake inoperative by finding its way on to the brake drum face. In the chassis plan the familiar torque rods can be seen slung from the cross member immediately in front of the axle. It is extremely hard to see why it should be necessary to fit an additional pressed steel torque rod, as the one would undoubtedly be perfectly capable of looking after all stresses and axle movements which are likely to occur. An unusual feature of the axle which is not very plainly shown in any of the drawings is the locomotive type of horn plates used instead of radius rods, and provided with a special form of lubricated slide to allow for the action of the spring. In connection with this slide there is a small fitment which is a good example of the care bestowed on the detail design of this particular vehicle, the sketch, Fig. VI., illustrating the manner in which the lubricating of this slide has been accomplished, and a neat, strong, and accessible form of lubricator which has been finally adopted in connection therewith, it being only necessary to raise the lid with the spout of the oilcan and fill the wick box shown. Although probably heavier than the average radius rod construction, the horn plates are a satisfactory method of obtaining the necessary rigidity. One would, however, wish that the slides themselves were either covered in entirely or that both sliding members were arranged in such a manner that they could be removed and replaced with great ease. It is extremely hard to believe that any sliding joint can be satisfactory as long as dust and mud can have free access to the surfaces, and it would not take any great trouble to arrange some form of metal cover which could be responsible for warding off any foreign matter from the parts in question.

Beyond and above the rear axle is located the whole of the pump-operating mechanism and suction and delivery attachment device. These, as before mentioned, are driven by a long shaft through special gearing, already described. One peculiarity, the arrangement of which is noticed in the chassis drawings, Fig. IV., is the ball steady bearing used in the centre of the shaft on the cross member which carries the radius rod pins. The idea of this small bearing is to steady the inevitable vibrations of a long shaft driven at a fair speed, but it is obvious that when the said shaft is provided with a universal joint at both ends a fixed joint in the centre would neutralize the efficiency of both movable joints to a considerable extent. It is therefore rather hard to see why this particular centre bearing should not be made of the swivel type to which it could be adapted without much expense and without altering the design to any great extent.

On the end of the pump shaft there is

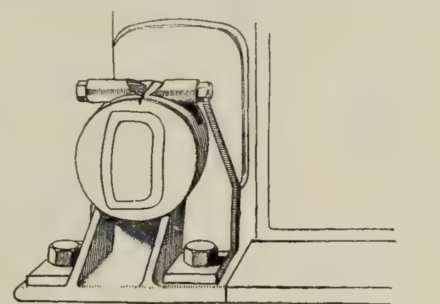


Fig. VII.

a small cone leather clutch and a spur gear meshing with a pinion keyed to the shafts of two reciprocating air pumps. The cone clutch has a striking fork connected by a long rod to a lever, shown in the elevational view of the chassis, and the air pumps are both connected to the turbine through a ball valve situated in the small circular casing, clearly visible in Fig. IV. The idea of these two air pumps is to facilitate the starting of the turbine pump by creating a vacuum therein, and thus allowing the casing to fill up, although the suction pipe may be dropped some considerable depth below the road level. Operation of the air pumps is automatically checked when the turbine pump has a sufficient quantity of water by means of the cut-off ball valve, which will only remain open as long as there is a vacuum in the turbine casing. Immediately after filling, operation of the cone clutch starts up the turbine shaft and the three-stage main pump comes into operation.

A diagram of this pump is shown in Fig. VIII., where it will be at once seen that the rotors are of the ordinary centrifugal type, taking water from the centre and delivering it to passages in the outer cases through a cone type nozzle. Pump No. 1 delivers to the second stage through a channel in the casing of the turbine. From the second pump it is again delivered to the final stage pump through another channel. Thus at 1,000 r.p.m. the pump in question is capable of extremely high pressure, varying only by the amount of water delivered to it through the suction pipe. When an extremely abnormal jet is required, hoses can be connected to the turbine casing through apertures near the centre of the pump seen in the plan view of the chassis.

The delivery pipes for two hoses are on the extreme right hand of the pump in the same figure, and are controlled by the wheel valves seen immediately above them. In connection with these wheel valves there is an exceptionally neat fitment which helps the machine to start pumping slightly more rapidly than the ordinary reciprocating steamer. Each of the delivery pipes has a flap valve which remains firmly closed when there is a vacuum in the turbine casing and which, of course, opens when water pressure arrives at the delivery pipes. Consequently it is not necessary to close the valves with the hand wheels shown, and hoses are connected direct to the delivery pipes without any further motion on the part of the fireman. It will be noticed that through the whole of this pumping scheme, every effort has been made to overcome the original troubles and defects of the turbine steamer to provide an engine, which, on the arrival at a fire, would be able to start pumping almost instantly.

Gear control is operated by a gate change, while the expanding rear brakes are brought into operation by a movement of the hand lever seen outside the gate quadrant. Concerning the hand brake gear, an arrangement is used whereby it is possible to obtain a brake shaft separate to that used for the gear control, although the brake lever is actually on the same quadrant. It is possible that the same results might be obtained by placing the brake lever directly behind the change speed lever and cranking it in such a manner that it will not come into contact with the quadrant of the latter. A long compensated rod is taken across the frame behind the last cross member, and the usual turn-buckle adjustment is provided to all necessary parts.

A stout H-section girder is used for the frame side members, having dumb irons for the front springs rivetted into position, and a series of cross members with gusset plates in order to obtain a certain amount of rigidity in an abnormally long side member. An angle steel under frame of considerable length is attached to these two cross members for bedding down both the engine and the two auxiliary gearboxes. Behind the rear axle the brackets which fasten the air and turbine pumps to the frame act in some measure as cross members, but it would seem better if care had been taken to provide an extremely strong tubular girder which might also act as a support for the dashboard, as this point is the centre of great length of practically unsupported frame side member.

The rear springs are carried at both ends on swivel links in brackets rivetted to the side frame members, and grease cups are provided for each of the pins in these joints, in approved touring car fashion. In front of the forward cross member, the usual Dennis radiator is slung in swivel brackets shown in Fig. VII., which are capable of allowing a certain amount of free movement limited by the flexibility of the rubber joint on the water pipe.

TUNING UP A CAR FOR THE TRACK.

By R. W. A. Brewer, A.M.I.C.E., M.I.M.E., M.I.A.E., F.S.E.

IN the following article I will endeavour to point out a few small detail alterations, which are perhaps not so generally known as some of the more familiar dodges which are resorted to, in order to improve the running and speed of a car on the track. Naturally, one cannot unburden the whole of one's experience and knowledge gained, in a journal that is sent broadcast throughout the world, and some slight reservations must necessarily be made, but sufficient indication will, however, be given as to "How it is done," though even then the knowledge alone will not be finality. There is always a considerable amount of success depending upon how that knowledge is carried out in actual practice.

I do not intend to deal so much with the actual constructive details of a racing car or engine, as such a machine can be specially designed to give the extraordinary results we have witnessed during the past few years. Much, however, can be performed by any one who has mechanical skill, when he knows how to apply it to an ordinary standard car, and improved running will result therefrom. The two reactions which balance out when a car is doing its maximum, are engine power on the one hand, and the combined resistance of friction and the air on the other. It is our object, therefore, to increase the former as much as possible and to diminish the latter so that the critical balancing point, in other words the car speed obtainable, is higher up the scale.

Increasing Engine Power.

When one tests an engine on the bench, a curve of speed and power can be plotted and, where the power units are ordinates, this curve will rise to a summit and may remain flat for some time, finally to droop away. On the other hand, the summit may only be short and it is desirable so to calculate the car resistance and the maximum speed which will be obtained, that the engine will run at a rate of revolution representing the summit of the curve. There is a point, however, which is probably not generally known, and that is, that the best results are obtained when the engine is undergeared by an amount of about 20% (or 10% when the peak of the power curve is short) when used on Brooklands track. When one carefully reasons out why this should be, the first point for consideration is the acceleration at starting, as valuable seconds can easily be lost by a car which does not get away owing to the gear ratio being too high. During the process of changing up from the first speed to the top a high-geared engine never gets its speed up to that which corresponds to the peak of its power curve, so that it is therefore running considerably below its maximum output during the whole of this period.*

Supposing we are considering an average powered car as seen on the track, it will have attained its maximum speed somewhere on the half mile straight, and by that time the minimum resistance will be experienced, unless of course there is a strong headwind, and the engine will be running at rather below its maximum efficiency. However, supposing there is a headwind, which is not at all unlikely at this part of the track, the engine speed will be dropped slightly and the engine itself will then be exerting its maximum effort. Continuing the course of the car round the track, it eventually has to climb the hill at the back of the members' enclosure rising up round the small banking. It is here that time is lost by a car that is geared too high, and the *low*-geared car drops in speed only slightly, to the most efficient speed of engine revolution. Coming off the banking, where a lesser power is required, the engine again speeds up, owing to the decrease of resistance and there is a loss of engine efficiency. I think that the foregoing argument will give a clear indication of the gear ratio which best suits the engine.

The next consideration is that of piston speed, and as the Brooklands rating takes no consideration of this, it is our object to make the stroke as long and the rate of revolution as high as can be conveniently and possibly done. Piston speed in such a case means displacement, which in unit time can only be increased (when the stroke is fixed) by increasing the rate of revolution. Much has been done in recent times in the way of lightening the reciprocating parts by the use of steel pistons, etc., and without further going into these well-known details I will give as an example a 90 mm. diameter piston with gudgeon pin and rings weighing one pound, and another maker's piston of similar diameter for a long stroke engine together with its gudgeon pin, rings, connecting rod and the bearings for both ends of the rod, weighing altogether three pounds.

Great importance should be attached to the elimination of friction between the cylinder walls and the pistons, and this can, to a great extent, be accomplished by so arranging the fit of the pistons, that sufficient clearance is allowed for their expansion when at a high temperature, without their causing any sort of binding on the cylinder walls. The choice of a suitable lubricant is also very important, and in this connection care should be taken that the lubricant does not wholly expend itself in creeping up the cylinder walls, to be burnt in the cylinders themselves or to be dissolved off the walls by the incoming carburated charge of air. I personally attach some importance to the use of a bottom ring on the piston, as when this is properly arranged it prevents some of this trouble occuring. If there is any tendency for such a ring to exclude sufficient lubricant, the pistons can conveniently be drilled through their walls and a supply of oil splashed on to the cylinder walls through the pistons.

It is perhaps unnecessary to point out the great care that should be taken in balancing an engine for high speeds, and not only should the reciprocating parts themselves be in static and running balance, but the pressures behind the pistons should also be as nearly as possibly equal, when the engine is performing its work. With further regard to this question of balance, it is a well-known fact that six-cylinder engines have caused an enormous amount of trouble through periodic vibration. I have recently had the opportunity of discussing this matter somewhat fully with a well-known designer, and we came to the conclusion that the principal cause of the trouble was due to the lightness of both the crankshafts and the base-chambers employed by the majority of six-cylinder engine makers. This is not only a theory, for it has been proved in actual practice: I personally can testify that the adoption of robust dimensions for these portions of the engine has completely and entirely eliminated periodic vibrations at all speeds up to 3,500 r.p.m.

Reverting again to volumetric displacement; when the question of rate of revolution and balance is settled, there is left that of filling the cylinders with carburated air and expelling the burnt products of combustion. When a gas has to be transferred from one position or receptacle to another, its rate of flow depends upon two factors: namely, the difference of pressure on each side of the intervening medium, whether it be a plate, a tube, or a valve, and secondly, the amount of obstruction which such a medium offers to the flow of the gas. Neglecting friction, gas will, in flowing from a region of, say atmospheric pressure, into a partially vacuus space, follow the $V = \sqrt{2gh}$ law and considering therefore the inlet side of the engine it follows that if V has a high value, that of h will vary as the square of any increase of that value, which may entail serious loss of efficiency, as I will endeavour to point out. For example, taking the values of Weisbach for the co-efficients of flow of air through an orifice, the following table gives the ratio of the initial pressure to the final pressure and also the co-efficient for such an orifice by which the second part of the expression $V = \sqrt{2gh}$ must be multiplied in order to obtain the value of V. This value C is the co-efficient of discharge of the orifice.

FLOW OF AIR THROUGH AN ORIFICE DIAM. 2.14 C.M.

Ratio of pressures	1.05	1.09	1.36	1.67
Equivalent absolute pressure in inlet pipe, lbs. sq. in.	14.0	13.5	10.8	8.8
Difference of pressure on jet in inches of water	19.0	32.4	105	—
Coefficient of discharge	.558	.573	.634	.678

Atmospheric pressure is taken 14.7 lbs. per sq. in.

The above table is given as an indication of one of the problems which must be borne in mind when tuning up an engine, and it shows by means of a few figures where loss of efficiency may be remedied. This example may be taken as an indication of what is taking place in the smallest part of the carburettor in the vicinity of the jet, and to demonstrate this point a short table of co-efficients of efflux (from the same source) is given.

* This accounts for the lack of success of the big Mercedes cars in short events on the track, as compared with their behaviour in long races.

Co-efficients of efflux.

Through various orifices with pressures of 0·23 to 1·1 atmospheres.

	C =
Conoidal or Vena-contracta ...	0.97 to 0.99
Circular orifices in thin plates ...	0.56 to 0.79
Short cylindrical mouthpieces ...	0.81 to 0.84
The same rounded at the inner end ...	0.92 to 0.93
Conical converging mouthpieces ...	0.90 to 0.99

It will be seen from the above that the amount of air which can be made to pass through a given sized hole under a certain fixed pressure, depends upon the shape of the hole, and this is one of the main principles which has to be borne in mind when tuning up an engine. In addition, the nature or shape of the path of gases traversing the engine and its piping must receive careful thought, and it must be shaped to give the minimum resistance to flow. This resistance may be due to several causes, for we must consider the effect of restricted areas and the change of direction of a flow path. These not only set up friction between the gas itself and the sides of the pipe, but they create turbulent eddies, in themselves a loss of power. We will now commence a consideration of the flow path of the air from its entry to the carburettor and choose a suitable sized orifice for the admission of the air to the vicinity of the petrol jet.

Taking as an example a certain engine with which I have been carrying out numerous experiments, and making the assumption that at the maximum speed obtainable the cylinders are 70% of their working volume full of fresh charge, when reduced to atmospheric temperature and pressure, the following figures give absolute values of the conditions prevailing. The 70% assumption does not affect the case, and it is sufficiently near the mark for our present purpose. The engine in question has a total swept volume of 1.65 cubic metres at a 1,000 r.p.m., and the gear ratio in the back axle is 16 into 54, the tyres being 815 mm. diameter. When the investigations were started the maximum speed was 48 m.p.h., and the carburettor diameter 22 mm., having a net sectional area of 3.4 square cms. Under these conditions the air velocity at 70% was 310 ft. per sec., equivalent to 23 ins. of waterhead. The carburettor was then changed for one 24 mm. diameter, having a net sectional area of 4.13 sq. cms., and at the same speed the air velocity at 70% volume was 255 ft. per sec., equivalent to a waterhead of 15.5 inches. The engine speed under these conditions was 1,680 r.p.m., and the maximum obtainable 1,800 r.p.m. Steps were then taken to increase this engine speed, and by other means it was increased to 2,100 r.p.m., at which speed the gas velocity on the 70% basis, using the 24 mm. carburettor, was identically the same as in the first experiments, namely 310 ft. per sec. when using the 22 mm. carburettor.

This velocity I consider much too high, although one does receive a certain benefit when picking up speed, and for a short race, as previously pointed out, this matter is of the greatest importance. Further alterations were then made by re-balancing the engine and slightly modifying the piston form, so that an increased speed of revolutions up to 2,500 r.p.m. was obtained, and a mean of 2,400 r.p.m. maintained on the track. Calculating now on the higher value for a 26 mm. diameter carburettor, with a net area of 5.25 sq. cms., it will be seen that the gas velocity, even at this higher speed, is brought down to 280 ft. per sec., and under these conditions most satisfactory results accrued. At this time the identical engine and chassis were travelling 13.3 miles per hour faster than in the previous best performance, obtained on the track by the official timing.

A subsequent increase in the size of the carburettor to 30 mm. diameter, with a net area of 7.0 sq. cms., did not show any improvement, and in this case the velocity of air through the carburettor was reduced to 210 ft. per sec. This velocity was undoubtedly too low efficiently to disintegrate the particles of petrol, and evidence that sluggish combustion was taking place was apparent in the increased noise of the exhaust. Unfortunately this carburettor was not fitted until the last moment, and there was no time to replace it by the 26 mm. previously used. We may therefore conclude that with this type of carburettor, i.e., Claudel Hobson, the most suitable maximum velocity for racing is of the order of 250 to 280 ft. per sec.

Having made the above investigations and setting them forth here, I made reference to the AUTOMOBILE ENGINEER for March, 1911, to compare my figures with those I had calculated in the article on Small Bore Engine Development. It will be seen on reference to page 283 that the winning Hispano Suiza, in the French Voiturette race in 1910, showed a calculated gas velocity of 269 ft. per sec. through the carburettor. My late experiments therefore go to confirm the most suitable value for this figure. We have now arrived at a fundamental point from which we can proceed to work. Bearing always in mind the fact that the speed of gas should not undergo any great variation after it has been formed, and that all variations should be in the reduction of its velocity, the inlet pipe can be shaped suitably, so that the outlet from the carburettor continues with a gradually increasing sectional area if possible, to a branch pipe formed with easy bends and without pockets or dead ends. I personally prefer a pipe consisting of suitable curves to a "Y" shape, but better still is probably the arrangement adopted in the 19.6 Austin "Pearly III.," where the two carburettors and four inlets are arranged on a continuously circulating system. The object in view is to eliminate pulsations in the inlet pipe as far as possible, as they only tend to precipitate liquid petrol, especially when the mixture is rich. Bends in a pipe should be very easy and, as an instance, the following table is appended, showing the frictional resistance which is set up by bends, where the *radius* of the bend is given in terms of the pipe diameter, and the *resistance* of the bend in terms of equivalent lengths of straight pipe being multiples of the diameter.

Effect of bends in pipes.

Radius.	5	3	2	1.5	1.25	1.0	0.75	0.5
Equivalent length of straight pipe.	7.85	8.24	9.03	10.36	12.72	17.51	35.09	121.2

It will be seen from the above table that in addition to the deleterious effect upon carburation, the shape of the pipe may cause losses through considerable frictional resistance.

The losses on the inlet side are very important as the limits of permissible pressure between which the inlet system works, are very small, varying as they do from that of the atmosphere to that behind the piston during the inlet stroke.

The system of inlet valves is naturally of the highest importance, and for this reason we hear so much of slide valves and other contrivances designed to lessen the losses which ordinary valves incur. Apart from special engines fitted with inlet valves of an area practically equal to the piston area, we will take ordinary standard valves and consider what can be done to enable them to pass as much mixture as possible whilst offering the least resistance. Suppose the valves to be of the mushroom type, as is usual, the gas in its passage from the stem side round the seating and over the head, will suffer certain changes in direction, which should be made as easy as possible, while the area of flow should be kept as large as possible in order to comply with the axiom previously pronounced. Should the valve guide seriously obstruct the passage, or come too close under the valve head, it should be machined or filed to give increased clearance, and all projections on the pocket casting or carbon deposit or any other foreign matter should carefully be scraped away.

The underside of the valve then requires attention and, in order to prevent any possibility of the stem catching on the guide, so throwing the valve over and preventing its true descent, it is advisable slightly to reduce the diameter of the stem by means of a file, for some little distance below the valve head. The valve head itself now requires some attention in order to decrease its resistance to the passage of the gas, by giving it a better formation than is usually employed, and at the same time increasing the area of the annulus through which the incoming charge passes. If the valve is taken and placed with its stem upwards, its chamfer can be treated with a file so that the smaller diameter is filed away in the form of a radius running out on the lower side of the valve head. Only the larger diameter of the chamfer for about 2 mm. is then used for seating, which is quite sufficient, and the lower radius considerably increases the valve area whilst giving a more easy and uniform direction of flow for the gas. The setting of the cams is not usually at the command of the private owner, and in order to affect any alteration in the valve setting, a new cam shaft is necessary. This is, however, not always a sine qua non, as a well designed cam contour will give excellent results at high speeds if the cam roller can be kept in contact with the cam itself.

The simplest method of attaining this object is by the employment of exhaust valve springs for the inlet valves, but if there are any abrupt changes of direction on the cam faces these can, with advantage, be modified by grinding.

When the valves are situated on opposite sides of the cylinders a marked gain is experienced by allowing the valves, i.e., inlet and exhaust, to overlap in their period of opening, and, if the inlet opens on the dead centre or 5 degrees late, the exhaust may remain open 15 degrees late

when combined with an ejector pipe. In the event of it being impossible to alter the cam arrangement in any way, and in the event also of the valve setting being not quite suitable, it is often possible to make a slight modification by adjustment of the tappet heads. Generally speaking, however, it is desirable to take full advantage of the cam lift, and set up the tappets so that there is minimum clearance between them and the valve stems. It is sometimes possible to modify the caps over the inlet valves slightly so as to increase the compression, but in the ordinary way the result does not warrant the trouble incurred. Personally I think there is very little to be gained by abnormally high compressions, and should consider 70 to 75 lbs. per sq. inch gauge to be a good average, but the maximum value depends upon the shape of the compression space. When the valves are in the cylinder heads the maximum permissible compression is much higher than when the valves are in pockets.

Passing now to the exhaust valves, somewhat similar remarks apply here to those made in respect of the inlet valves, but as the passage of the gas is in the opposite direction, alteration of the formation of the head should be made by filing a radius on its larger diameter. In some engines the exhaust valve heads are parallel for one or two millimetres above the larger diameter and project this amount above the valve seating and into the pockets. When this is the case it sometimes occurs that the clearance between the valves and those portions of the pocket remote from the cylinders are very small. Judicious treatment with a file can improve the conditions prevailing here, and the flow path thus modified reduces the resistance of the passing gases. In engines which have been in use for some time, considerable carbon deposits may be found in the pockets immediately below the valve seatings, and these deposits should be cleared away to give every facility for the gas flow. Before leaving the exhaust valves, which, needless to say, should be carefully ground on their seats, I should like to draw attention to a point generally overlooked.

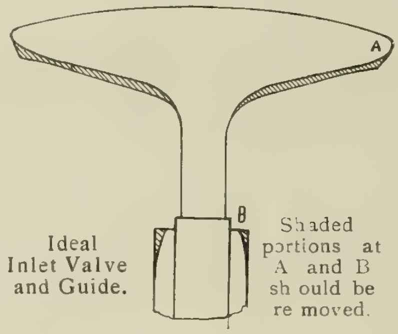
When an engine is working at full duty, even for a few minutes, the expansion of the exhaust valve spindles becomes very important, and, in a particular case in mind, one of these spindles permanently increased in diameter due to the heating and the load impressed upon it when lifting the valve, so that it became a perceptibly tight fit when the valve was withdrawn cold. In order to eliminate any risk of the spindles sticking in their guides, they should be filed or ground to give ample clearance and, in distinction from slack inlet valve spindles, which materially affect the carburation, slack exhaust valve spindles are in a measure desirable for work on the track.

The shape of the exhaust pipe has very marked effect upon the working of the engine and the following points must be borne in mind in the design of the special exhaust pipe which is so essential for high speed working.

1. The pipe must be so designed that the bends are few, the radii of the bends as large as possible, and their internal area at least as large as the opening on the cylinder casting at the one end, to which they are fixed.

2. The arrangement of their coupling up to the common pipe, if one is used, should be such that rather than the emissions of burnt gas under pressure from the various cylinders interfering with each other, they should assist one another.

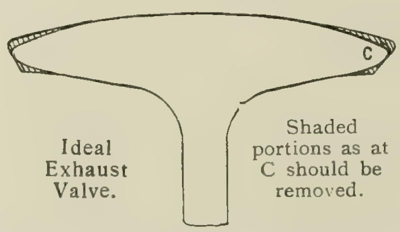
3. The pipes should be so carried that they do not obstruct any part of the engine which might require adjustment, and their joints should be such that they are not likely to come adrift. Even if one of them did so it should be impossible for the pipe or its flange to slew round and obscure the passage of the exhaust from that cylinder. Considering the first set of conditions, the same remarks apply here as were made in connexion with the inlet pipe, with the exception that in the latter case the presence of obstructions tends to raise the temperature of the obstruction itself on account of the intensity



Ideal Inlet Valve and Guide.

Shaded portions at A and B should be removed.

of the impinging flame. A single bend of large sweep from each cylinder should suffice to carry the exhaust clear of the engine, and this bend should sweep outwards and towards the tail of the car. Under heading (2) it is my personal opinion that the best arrangement, and the simplest, is to allow all the ends of the separate bends to enter one common pipe of large dimensions. Thus no one pipe obstructs by its presence the flow of the exhaust gases from any of the other pipes. Furthermore, as the various pipes discharge close together into a single pipe of at least four times the area of any one of them, there is a certain amount of ejector action taking place, thus assisting the whole combination. According to Brooklands rules the exhaust must be taken to a point at least as far back as the



Ideal Exhaust Valve.

Shaded portions as at C should be removed.

rear axle, and again as a personal opinion I believe that, if suitably carried out, better results are obtained in this way than where the gases are allowed to escape from the valve pockets directly into the atmosphere. Undoubtedly the inertia of the gases down the pipe materially assists in scavenging the combustion chambers.

In high-speed engine work, one of the most important, if not the most important, detail is the ignition, and we may take it as a fact that the great improvement and development in magneto design and construction has made the high-speed engine a possibility. In all probability it is the absence of metallic inertia in the magneto, as compared with a trembler coil system, which is accountable for the great rapidity of spark production with no measurable lag. We may take it, therefore, that the spark is produced at the plug electrodes precisely when the contacts on the magneto are separated, the electrical lag being immaterial. The timing of the ignition point is therefore, far more exact than in the older methods either of separate tremblers or high tension distributor. A great deal has been written and said with regard to the position of sparking plugs in the cylinders and the number of firing points per cylinder. I have tried in a particular engine one, two and three plugs per cylinder working in various arrangements. The plugs were situated in the side of the inlet pocket, on the top of the same, screwed through the valve caps, and I have also fitted Bosch plugs with long electrodes into the centres of the cylinders, so that the sparking points were about an eighth of an inch inside the plug pockets in the cylinder castings. The best results were obtained when the firing point was set 22% of the piston stroke early, although advancing to 25% made little appreciable difference at high speeds. The engine could not, however, conveniently be run at normal speeds with this latter amount of advance. With a Bosch two-spark magneto connected to the plugs in the centre of the cylinders and to those in the sides of the valve pockets, very little difference was noticed when either one set of plugs was in operation, but, with the firing point fixed, the effect of switching on both sets of plugs was apparently that of advancing the firing point. In other words, the adoption of a pair of plugs firing simultaneously across a part of the cylinder and valve pockets remote from the exhaust valve was to increase the rapidity of firing. Evidently the waves of pressure set up from two different points simultaneously were much more pronounced and vigorous than was the case when the increase of pressure was set up from one point alone. Taking for a simile waves of water, one notices that the apparent energy is very much greater when two waves meet, one of which is, say, a rebounding wave from a sea wall, than when the same wave expends the whole of its energy on the beach. I venture to make one further point in advocacy of the two-spark magneto, and that is that the maximum pressure is reached more quickly, and can be therefore, controlled more exactly, than is the case when a single firing point is used, while the pressure throughout the mixture has consequently to be generated throughout a greater length of flame path.

As against this theory it may be argued that the velocity of flame is very much greater than that of a piston and, although we have definite figures relating to mixtures of pure gases, I am not aware that any such figures are available for the type of gases, both active and inert, found in the combustion chamber of an engine. The probability is that the velocity of propagation of the flame is lower than we expect, particularly when the proportions of the carburated mixture vary in actual working. One can only be advised to adopt the best method of ignition available, and from personal experience I can say that is the two-spark system. In using it I have never had a misfire under the most trying conditions, and, although I have varied the adjustments of the carburettor, and been unsparing with lubricating oil, the plug points have given no trouble from fouling.

Chassis Losses

It is not necessary to go deeply into all the multifarious sources of loss in the chassis, nor into the exact amount of wind resistance afforded by the various parts of the complete car at high speeds. We know these exist and their magnitude depends upon the design of the machine as a whole. These losses can be minimised by streamline formation and the arrangement of such form must be left to the taste of the individual and to the facilities available in the matter of construction. In passing, it is somewhat remarkable that streamline formation for high speed motor cars has only been really appreciated during the last couple of years. For track work the benefit derived from correct body design is very great, but it does not appear to be so marked for road racing. There is no doubt that the winner of last year's Voiturette race at Boulogne would have shown some improvement had the car been fitted with a streamline body, but I think it is the general opinion that the improvement would only have been slight. In the Boulogne race this year it was found necessary to remove the "Brooklands tail" from one car as the disadvantage experienced on account of its centrifugal force effect at the corners outweighed such advantages as the tail gave in the minimisation of the vortex at the rear of the car when travelling fast on the straight.

The above fact does not in any way discount the value of a tail for the purpose intended, as without doubt the effect of its presence is most marked. For road racing, however, where there are numerous corners, both the weight of the tail itself and the side pressure of the air upon it cause lateral stresses in corner work. A car fitted with a tail cannot therefore, as a general rule take a corner with safety at so high a speed as when the tail is absent. For work at Brooklands a tail is naturally most desirable, especially as arranged in modern practice, so that the tail envelops the bevel gear casing of the back axle.

It is perhaps not digressing from the subject to call attention to the effect of wind resistance upon a body moving at a high speed, such as a racing motor car. In the first place it is accepted that the wind resistance varies as the V^2 law and that the rolling resistance does not vary appreciably with the speed, but is slightly greater at high speeds, not on account of axle friction so much as on the various minor frictions in the transmission, and any slight want of balance in the rotating parts such as the propeller shaft and universal joints. At the speeds we have under consideration, it may be taken therefore, that the wind resistance is by far the largest to be overcome and, if we take for example a car weighing a ton, we get approximately the following values for the resistances. The rolling resistance should not exceed about 60 lbs. per ton, and could be made considerably less, and supposing we have an engine developing 40 h.p. at the road wheels we find that at 75 m.p.h. the available engine thrust transmitted to the chassis is 200 lbs., leaving a balance of 140 lbs. to overcome wind resistance. Taking the wind pressure on normal surfaces at 16.87 lbs. per sq. ft. at 75 miles per hour, we find that for conditions to balance, the projected area must not exceed 8.3 sq. ft. normally or its equivalent in streamline form. Supposing now a car speed of 80 miles an hour is to be attained, the engine thrust at this speed is 188 lbs., leaving a balance of 128 lbs. to overcome wind resistance. It will be seen that the wind resistance is now more than twice the other resistances, and at 80 m.p.h. the air pressure is 19.2 lbs. per sq. ft. on normal surfaces. Calculating out we find that the area must not exceed 6.67 sq. ft. of normal surface or its equivalent in streamline form. It is a somewhat difficult matter to ascertain with any degree of accuracy what will be the actual wind resistance of any particular car, but, if it can be found at one speed, it can be approximated for other desirable speeds.

With reference now to rolling losses, that due to the wheels leaving the track is most pronounced. Probably at no time after high speed has been reached do all the four wheels remain on the track at the same time, and considerable increase of performance can be noted when the springing is suitably designed to keep the driving wheels down. It is not a question of chassis weight, or of loading up the back of the car for small or medium powered engines, as we have seen in the previous calculation that the tangential force between a single wheel and the track is only of the order of 100 lbs. at high speeds. What is required, however, is reduction of the unsprung weight, combined with an elastic medium which will keep the treads of the driving tyres in contact with the surface of the track so that they do not spin.

I have personally tried devices designed for such a purpose, and the adoption of light subsidiary springs, which give to the small irregularities, has undoubtedly resulted in producing an improvement in the contact. Springs used for this purpose should damp out fairly quickly and are quite distinct from any system of shock absorber employed to damp out the long vibrations in the main supporting springs.

Such springs as are now referred to are similar to those adopted in the J.M. system, consisting of ordinary coil springs in compression, and the other arrangements, such as The Lever Spring, but whatever system is adopted should be carefully proportioned for the particular case. The elimination of rapid small vertical vibration from the rear of the chassis naturally improves the running of the engine as a whole, and not the least by minimising the many aggravations which are already suffered by the float needle in the carburettor.

With regard to the laminated springs, these should preferably have a long period—this can be obtained either by the use of long laminations or by three-quarter or complete elliptical blades. The combination of such springs with those of a short period is the most satisfactory for track work, although one cannot state any experience of pneumatic suspension at Brooklands. It is to be hoped that we shall be able to witness the performance of a pneumatically suspended car on the track before very long.

I have already referred in passing to the friction and possible want of balance in the propeller shaft and universal joints. The amount of the losses occuring here depends upon the design of this part of the transmission gear, the amount of sliding which takes place in the square couplings, the sizes of the surfaces in contact, and the efficiency of the lubrication. Naturally, in what one would consider to be a well designed propeller shaft arrangement, the amount of sliding motion will be exceedingly small, but in other types there is no doubt that considerable losses occur. A case in point was that of a certain well-known car which would not hold the road, and this fault was attributed to the telescopic universal joints. The theory propounded was, that when the axle dropped at the time that the wheel entered a depression, telescopic motion occurred in the universal joint, evidently on account of the radius of the axle from the centre of its universal joint not exactly corresponding in length with the radius to which the axle itself was anchored. It was supposed that the pressure on the driving faces in the telescopic joints was so great that the lubricant was squeezed out from between the faces, causing them to bind. If these circumstances occurred, the joint itself would tend to retard the axle from returning to its normal position, and in doing so would undoubtedly cause the wheels to roll badly, i.e., they would not hold down on the surface of the road. The above explanation given by an official of the manufacturing company appears to be a somewhat risky one, but I am informed that when the telescopic joints were greatly increased in size a marked improvement was noticed. Personally, I have never experienced trouble of this kind, as in the first place I prefer to use an axle that is suspended as nearly as possible with true radial motion.

There may be losses due to unnecessary friction, and want of balance in the universal joint through the tail of the direct shaft in the gearbox becoming bent. This is more likely to occur in that type of live axle arrangement where a torque tube surrounds the propeller shaft and a universal joint, upon which the torque stresses are impressed, couples this shaft to the tail of the direct shaft in the gearbox, which latter supports the footbrake drum also. A permanent set can easily be given to both this gear shaft and to the end of the propeller shaft by a fierce application of the brake. The stress thus impressed is much greater than any driving stress, and I have known shafts to be slightly bent in this way, thus throwing the transmission badly out of balance.

It is unnecessary to draw attention to the importance of keeping the brake shoes clear of the drums, and to the numerous other well-known small points in the elimination of the friction on the road wheels.

In conclusion, however, judging by the comparatively large number of cars which fall out at Brooklands from small causes which are easily avoidable, it appears that several of the common failures are not sufficiently known and appreciated by a number of men who use the track.

Broken petrol pipes are one source of trouble and personally I take the precaution of annealing this pipe and frequently examining the unions, sometimes having them resoldered or brazed. The petrol pipe and oil pipes should be carefully cleaned out as well as the tanks to which they are attached. Small ignition troubles also are numerous, and all wires, in addition to having their terminals screwed up tightly, should be double nutted and have their ends bound with string or wire. Sparking plugs also should be carefully examined, and the insulators tested. I also examine the water

circulation and take the pump adrift in addition to going round all the water joints and renewing doubtful ones. All the nuts and bolts on the car, especially those on the engine, gearbox, backaxle and steering gear should be carefully gone over and tightened up. Finally, it is a wise precaution to tie down the radiator cap and the bonnet, as more than one accident has occurred through these parts coming adrift, and so flying upwards.

With reference to the radiator itself, this for its size and shape depends so much upon each individual engine. Some difficulty may be experienced by the use of a hooded radiator for a car which starts at or near scratch, especially when the engine has to be kept running for a long time, as is fequently the case, before starting. At least ten minutes or a quarter of an hour elapses between the time that the cars are marshalled in the paddock and the signal to start is given to the scratch man. As no fan is fitted, the heat generated and transmitted to the water has not much opportunity of escaping. Great importance, therefore, ataches to the question of sufficient radiation and the facility for throttling the engine down to a slow speed.

CORRESPONDENCE.

No responsibility is accepted by the Editor for the opinions of correspondents, and the right is reserved to omit any portion of a letter if it is considered desirable to do so. Letters to be inserted with a nom de plume must be accompanied by the name and address of the writer. If publication in the next issue is desired letters should reach the Editor in no case later than the 20th of the preceding month.

THE INSTITUTION OF AUTOMOBILE ENGINEERS.

Sir,—I notice in your August issue a letter signed by "Member" to which I am authorised to reply.

With regard to the constitution of the Council I need only point out that if your correspondent is dissatisfied, he can himself, or in consultation with his friends, nominate any number of members whom he considers more suitable, and these nominations would be submitted to the ballot of the members in accordance with the rules of the Institution; thus the constitution of the Council rests entirely with the members themselves. I may say that half the Council retires annually.

Only one other point I think needs nnotice. "Member" says that "the question of the training of engineers appears to have been settled off-hand by a Committee appointed by the Council." If he would have taken the trouble to have obtained accurate information before rushing into print with so ridiculous a statement, he would be aware that the I.A.E. have made no attempt to settle the question of the training of engineers; all that this Institution has done in that direction is to collect information from various works as to their facilities and vacancies for apprentices, and to place this information at the disposal of those interested. A very large number of parents and others have already availed themselves of this offer, and it has undoubtedly proved of great service to many who are not in direct touch with the industry.

I regret that it should be necessary to remind "Member" of the engagement entered into on the proposal form. There is a proper course open to members who are dissatisfied or discontented, but members, so long as they remain members, are bound by their undertaking. A member does not "advance the objects of the Institution so far as in his power" by writing anonymous letters to the press, especially of the present type.

BASIL H. JOY, Secretary.

WORM DRIVING.

Sir,—We have read with great interest the paper by Mr. E. R. Whitney, also the criticism by Mr. H. Kerr Thomas on the efficiency of worm gear which appears in your August issue.

We think the latter gentleman sums up the situation very clearly, and his remarks coincide with our ideas on this subject.

We are afraid sufficient stress is not laid upon such important points as lubrication, rubbing speed, end and side thrusts, selection of materials, etc., all of which are matters of great importance.

We do not agree that the Hindley or Concave worm is superior to the straight; at first sight the area of bearing surface appears to be greater, but this is not borne out by actual fact, and if you will consider the angles at the centre compared with the outside, the difference owing to varying diameters is considerable, and longer the worm greater this becomes. Further, a concave worm has many disadvantages compared with a straight one, but we do not propose to enter into this, except to say that a straight worm can be ground after hardening, which is a great factor towards efficiency.

For lubrication we prefer a good grade heavy cylinder oil, but the pressure on the teeth and the rubbing speed must be such that lubrication actually occurs, because cases are not unknown where these have been so high as to interfere seriously with it.

Rubbing speed and pressure must of necessity enter largely into the design of any worm gear, because only by taking these into careful consideration can we obtain the dimensions which will give the best results, but unfortunately the information available varies with different designers, and only those which have had a reallv large experience are able to give this with any degree of reliablity, and the results of any experiments which have been carried out are naturally kept by the people who have conducted them.

In designing a worm driven back axle not only the end thrust of the worm, but also the side thrust of the wheel must be provided for, and in our opinion it is wise in this particular to err in the direction of liberality. There are cases where worm gears have proved an utter failure simply on account of the inadequacy and disposition of thrust bearings.

As regards materials, we use a special alloy of phosphor bronze for the wheel, and a high grade steel, casehardened, and giving a glass-like surface on the thread for the worm. This must afterwards be ground on the thread to eliminate distortion due to hardening and inaaccuracy from machining.

We have not the slightest hesitation in saying that in our opinion worm gear for motor vehicles for either pleasure or heavy commercial work is the right thing, ,but great attention must be given to the design, and also to the manufacture, but if this is done, we believe, the day is not far distant when all makers of importance will adopt it.

DAVID BROWN AND SONS (HUDDFD.), LTD.

LEAD COATED SHEET.

Sir,—In the last *Automobile Engineer* there was a letter referring to the various materials which are used by modern constructors in body building, and in that letter it was said that the lead coated sheets were used in very considerable quantities in the trade in question. I am writing to ask whether, in connection with these sheets, there has ever cropped up a peculiar little difficulty which once came within the province of my own experience. The sheets in question were worked up into curves for body work by hand and not by machinery. The trouble made itself felt only when the sheets in question had been finished, painted, and varnished, and the car was ready for the road. Shortly after this a peculiar blister would become apparent on the paint, and eventually the latter would flake off and destroy the entire finish. On examination of certain of the new sheets sent in by the suppliers, very little trouble was apparent on the surface, but when a considerable amount of manual labour had been put into the sheets, peculiar hard spots were noticeable on the surface, these being extremely hard, if not altogether impossible, to remove, and apparently growing in size according to the amount of work put upon them. Eventually, the trouble grew to more alarming proportions and a considerable number of finished bodies were seriously interfered with. As a general rule the manufacturers seemed unable to give any particular reason for the occurrence, and were utterly unable to provide any suggestion which would lead to better results in the future. A peculiar part of the whole affair was the non-appearance of these blisters in some consignments, while in others they were utterly unuseable because of them.

I should like to know whether any of your readers are in a position to give a logical explanation of this fact, or to suggest what steps may be taken which might remedy the same on receipt of the sheet so afflicted.

O. T. GOWER.

THE VALUE OF RACING.

A short time ago we expressed the opinion that racing, especially track racing, was of real value to the manufacturers who indulged in it. In the light of this the following is interesting, being the opinion of Mr. Enise, of the Lozier Co., one of the highest-class firms in America :—

"Our positive conviction that in no manner can a manufacturer of motor cars obtain so much valuable information that will enable him to build an absolutely safe and indestructible automobile as by entering his stock cars in long distance races. We have raced with stock cars in practically all the big road races held since 1907, and we attribute our success entirely to the fact that from the moment we started racing until the present time every Lozier car at the conclusion of the race has been carefully inspected and examined in all of its parts, and in every case we have detected evidences of weaknesses or imperfections which would eventually have caused trouble. We have immediately taken steps to correct these troubles, with the result that we believe the Lozier car to-day to be inconceivably superior to any car we could have developed without the assistance of our racing experience.

Naturally the matter of steering knuckles and tie rods was one of the first to receive our attention. While there is not a single case of a broken steering knuckle on a Lozier car in service, we know that the steering knuckles which we made a number of years ago would eventually all of them been liable to breakage, for we discovered in races that the best material obtainable would crystallize in time; merely making the parts stronger would not overcome the trouble, and will not to-day. We found it was necessary to secure absolutely perfect alloy steels. Experiments have been constantly carried on with various alloys of Vanadium, Chromium, Nickel, and Tungston; numerous chemical tests have been made of these alloy steels and an endless amount of experimenting with various heat treating processes has been carried on. The size, shape, and methods of fastenings have received the most careful attention on the part of our designers. We have produced steering knuckles that the most severe form of tests devised in a laboratory result in their being declared perfect, yet racing experience has taught us that the millions of vibratory shocks which these parts receive in racing will eventually result in a crystallization of the material. How to overcome this seemingly eventual result is a problem which has had the attention of our engineers over a year past, and we believe that it has nearly been solved. Without our racing experience to guide us, we believe it would have been impossible to have found a solution to the problem.

The above is simply one concrete example as to how racing develops the perfect car. The Indianapolis race taught us another valuable lesson which had been learned from no other race. Never in the history of the world has there been a 500-mile race held on a course under three miles in length. The Lozier car running at a speed of about 80 miles per hour, circling the track to the right, making four turns every 2½ miles, subjected the differential to most unusual strains; strains which probably no differential had been called upon to endure. While no difficulty was encountered during the race, an examination of these parts proved that had the car continued at this speed and in the same direction for several thousand miles, differential trouble would have resulted. From the lesson learned on examination of these parts, a slight change was made in the construction which, in the opinion of our engineers, would render impossible any difficulty with these parts under thousands of miles of running under the same conditions as obtained in the Indianapolis race.

This instance can be multiplied from our past racing experience, and it is folly for any one to argue that racing under these conditions does not furnish a manufacturer with most valuable data which a non-contestant is unable to obtain.

LONG ADDENDUM GEARS.*

By E. W. Weaver.

The bevel driving gears in the rear axle have probably given more trouble to automobile makers and users than any other two gears used on a car. For that reason any system of gear tooth design that tends to quieter running, greater strength or durability is deserving of consideration.

The system described in this payer is not new, although none of the authors of the standard gear books have deemed it worthy of more than a passing comment. It is, of course, understood that the tooth is of true involute or of octoid form, depending upon whether it is produced for a spur or bevel gear. The special feature of it is the lengthening of the addendum of the pinion tooth, with a corresponding shortening of addendum of the gear tooth—the whole depth remaining the same as in the standard tooth.

This is in direct opposition to the advocates of the stub tooth gear, the strong point claimed for that being the absence of sliding contact of the meshing gear teeth, it being as near to full rolling contact as is possible with fixed teeth. However, with the fully proved high efficiencies of properly cut worm or spiral gearing, the action of which is wholly sliding, the loss due to the greater sliding of the long addendum tooth may safely be set down as having been exaggerated.

With the addendum or face of the driver lengthened, the arc of approach of the gear tooth action is lessened and the arc of recess is increased — becoming all recess and no approach when the driver has only faces and the driven only flanks. This gives particularly smooth-running gears—almost equal, in fact, to spiral gears. As is well known, the friction of the arc of approach is much greater than that of the arc of recess—something on the principle of a man dragging a stick after him or of pushing it ahead of him.

Another advantage of this system is the very great improvement in the shape of the tooth when the pinion has a small number of teeth. Outlines of several size pinions and the mating gear are shown that comparisons may be made. It is readily seen that the pinion teeth with the long addendum are fully as strong as the gear teeth, while with the standard tooth they are not. This being the case, it is possible in designing a rear axle drive to select a smaller number of teeth for the pinion than one would otherwise wish to select — for instance, if the number of teeth previously used had been 17 and 54, the combination of 15 and 48 would give the same ratio, and would be fully as strong. With $5\frac{1}{2}$ diameter pitch tooth, the outside diameter of the large gear would be decreased something over an inch, thereby making the case that much smaller and lighter with all its inherent advantages.

Another disadvantage of using a small number of teeth in a pinion, with the standard tooth, has been the small amount of stock left between the bore for the shaft and the bottom of the tooth spaces. I have seen pinions in which, in my opinion, the keyway weakened the pinion seriously. This is got away from to a large extent by the decrease of the dedendum of the pinion tooth.

So much for its advantages. To look at the other side—its field of application is limited to gear sets having a large difference in the number of teeth in the gears and pinions on account of the gear tooth becoming weaker as the number of teeth decreases. The gear having the lengthened addendum must at all times be the driver, as in reversing the application of power the arc of recess becomes the arc of approach with its greater friction. This is of no account in the driving gears of a car, as in coasting no power is transmitted, except when there is a propeller-shaft brake. It may also be said in objection that it is not a standard tooth and that introducing another system is to be deprecated.

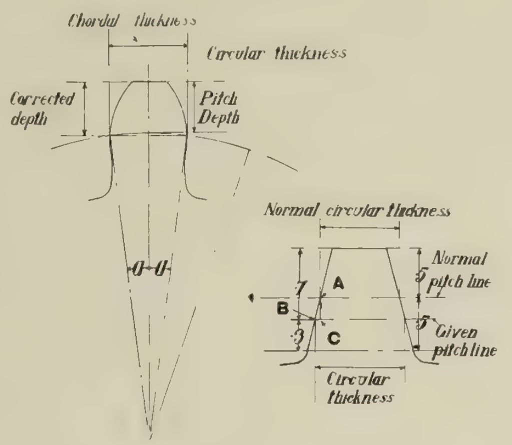


Fig. I.

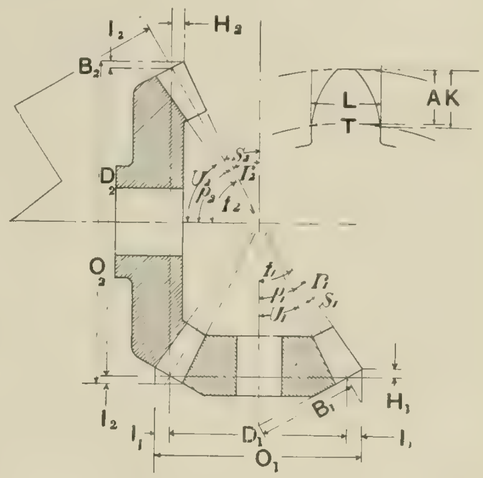
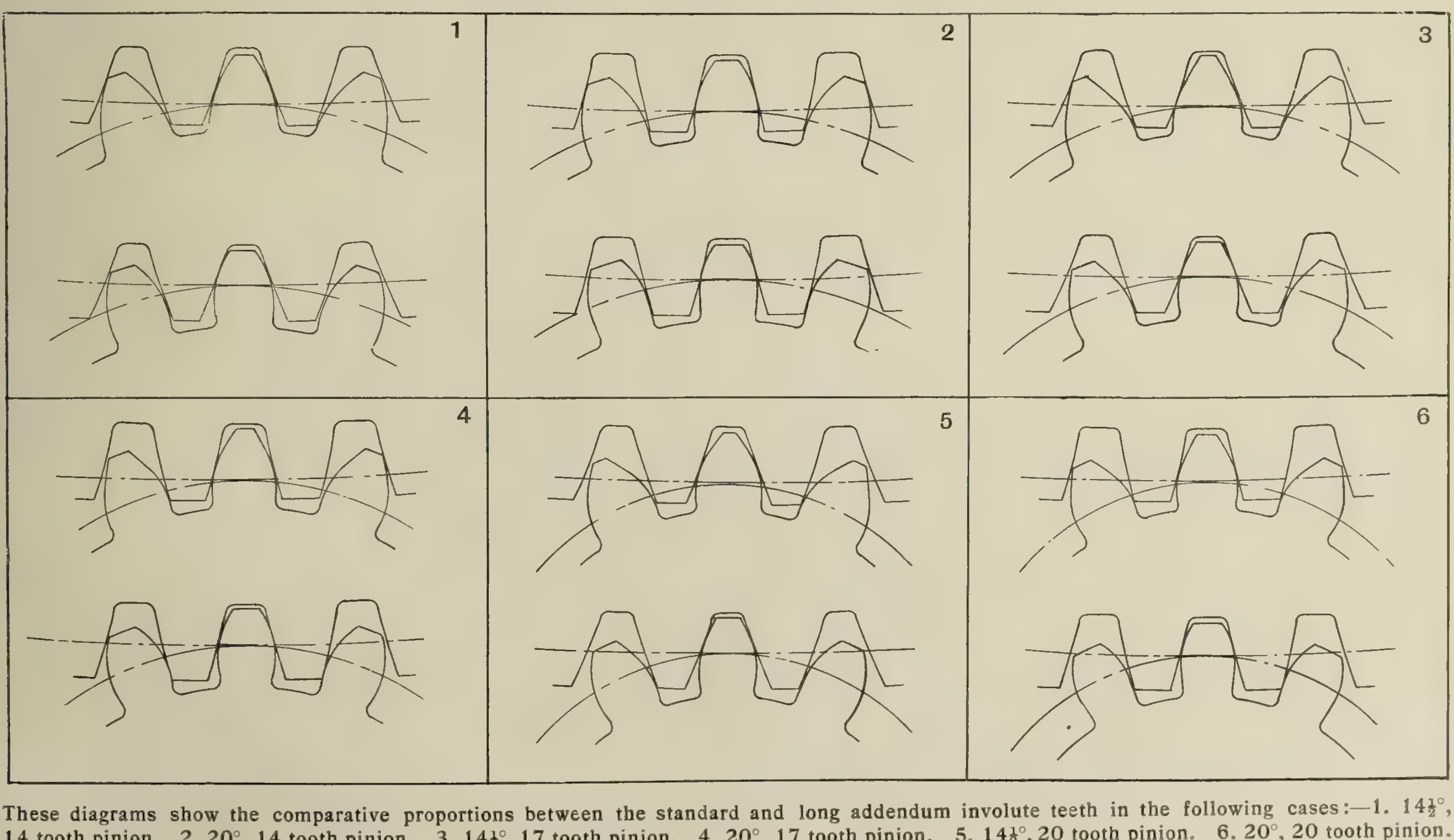


Fig. II.

One point which has not been touched on is the efficiency, and consequent life, of a gear of this system as compared with one of the ordinary or of the stub form of tooth. No exhaustive scientific tests to determine this have been made, to my knowledge, so it is a matter for further demonstration. However, so far as I have been able to learn, it compares favourably in this particular with either of the other types of teeth. In the design of the tooth form it is necessary to fix on some definite ratio between the length of the addendum and dedendum. This ratio theoretically should vary with the angle of pressure which is being used.

*Paper given to the Society of Automobile Engineers, June, 1911.



These diagrams show the comparative proportions between the standard and long addendum involute teeth in the following cases:—1. $14\frac{1}{2}°$, 14 tooth pinion. 2. 20°, 14 tooth pinion. 3. $14\frac{1}{2}°$, 17 tooth pinion. 4. 20°, 17 tooth pinion. 5. $14\frac{1}{2}°$, 20 tooth pinion. 6. 20°, 20 tooth pinion.

The addendum of the pinion tooth as further described in this paper is arbitrarily taken as .7 of its working depth, and .3 for the gear tooth for both 14½ and 20 degree pressure angle. To find the circular thickness of the tooth, at the pitch line, for these depths, multiply the circular pitch by .5659 for the pinion, and by .4341 for the gear for 14½ degree pressure angle. For 20 degree pressure angle multiply the circular pitch by .5927 and .4073 respectively for the pinion and gear.

This is most easily seen from the rack tooth (Fig. 1), which, being straight-sided, and the sides normal to the pressure angle, requires merely the solving of the triangle for the side *BC* and the adding or subtracting twice that amount from the normal circular thickness, depending upon whether the given pitch depth is greater or less than the normal depth.

Let a = normal pitch depth.
b = given pitch depth.
c = difference between a and b.
d = line BC (Fig. I.) = c x tangent of angle.

Required circular thickness equals one-half circular pitch plus or minus twice d.

The impossibility of getting accurate circular measurements necessitates the calculation of the chordal thickness and corrected pitch depth. Referring to Fig. I. let R equal the pitch radius for spur gear or the back cone distance for a bevel gear, angle a equals :—

$$\frac{360 \text{ Degrees}}{\dfrac{2 \times R \times 3.1416}{\frac{1}{2} \text{ circular thickness.}}}$$

Chordal thickness equals 2 x sine angle a x R.

Corrected pitch depth equals versed sine angle a x R plus the given pitch depth.

I have tabulated these values for gears with as large a range of teeth numbers as the system is applicable to with advantage, in my opinion.

The chordal thickness of teeth for spur gears of 1 diametral pitch and special pitch depth is given hereunder :—

To obtain chordal thickness of teeth and corrected pitch depth for any diametral pitch other than 1 divide figures in table by diametral pitch required.

20 Pressure Angle—For Pinions Addendum—Equal 7/10 Working Depth.

Number of Teeth.	Chordal Thickness.	Corrected Pitch Depth.
12	1.8545	1.4720
13	1.8554	1.4665
14	1.8567	1.4618
15-16	1.8573	1.4559
17-18	1.8584	1.4495
19-20	1.8592	1.4442
21-22	1.8597	1.4402
23-25	1.8601	1.4361
26-29	1.8606	1.4316
30-34	1.8609	1.4272
For Gears Addendum—3/10 Working Depth.		
35-41	1.2792	.6107
42-54	1.2793	.6085
55-79	1.2794	.6060
80-134	1.2795	.6040
134	1.2795	.6030

Note.—For bevel gears, find chordal thickness of tooth and corrected pitch depth of gear with the same number of teeth as a spur gear having a diameter equal to twice the back cone distance.

I have also arranged the formulæ in the logical routine order for all necessary calculations for a pair of bevel gears of this system ; symbols are as indicated in Fig. II.

As some firms are using the metric pitch or "Module" system, the conversion table below is given for their convenience.

Diametral Pitch—Standard Teeth.

Diametral Pitch.	Circular Pitch.	Nearest Metric Pitch or "Module."	Diametral Equivalent of "Module."	Circular Pitch Corresponding to Module.
2¼	1.3962	11	2.309	1.3607
2½	1.2566	10	2.540	1.2370
2¾	1.1424	9	2.822	1.1133
3	1.0472	8	3.175	.9896
3½	.8976	7	3.628	.8659
4	.7854	6	4.233	.7422
4½	.6981	5.5	4.618	.6803
5	.6283	5	5.080	.6185
5½	.5712	4.5	5.644	.5566
6	.5236	4	6.350	.4948

Formulæ for Long Addendum Bevel Gears.

Name.	Symbol. Pinion	Symbol. Gear	Formula. Pinion	Formula. Gear
Number Teeth........	N_1	N_2	$N_1 = P_d \times D_1$	$N_2 = P_d \times D_2$
Diametral Pitch	P_d		$P_d = \frac{N_1}{D_1} = \frac{N_2}{D_2}$	
Circular Pitch........	P_c		Table No. 1.	
Pitch Diameter in Inches..............	D_1	D_2	$D_1 = \frac{N_1}{P_d}$	$D_2 = \frac{N_2}{P_d}$
Pitch Angle..........	P_1	P_2	$\tan p_1 = \frac{N_1}{N_2}$	$\tan p_2 = \frac{N_2}{N_1}$
Working Depth	W		$\frac{2}{P_d}$	
Addendum............	A_1	A_2	$A_1 = .7 \times W$	$A_2 = .3 \times W$
Dedendum............	E_1	E_2	$E_1 = A_2 + G$	$E_2 = A_1 + G$
Clearance	G		$G = P_c \times .05$	
Full Depth	F		$F = W + G$	
One-half Diameter Increment	I_1	I_2	$I_1 = A_1 \times \cos p_1$	$I_2 = A_2 \times \cos p_2$
Outside Diameter	O_1	O_2	$O_1 = D_1 + 2I_1$	$O_2 = D_2 + 2I_2$
Circular Thickness....	T_1	T_2	$T_1 = P_c \times .5927$	$T_2 = P_c - T_1$
Pitch Cone Distance..	C		$C = \frac{D_1}{2 \times \sin p_1}$	
Back Cone Distance ..	B_1	B_2	$B_1 = C \times \tan p_1$	$B_2 = C \times \tan p_2$
Addendum Angle	r_1	r_2	$\tan r_1 = \frac{A_1}{C}$	$\tan r_2 = \frac{A_2}{C}$
Dedendum Angle......	s_1	s_2	$\tan s_1 = \frac{E_1}{C}$	$\tan s_2 = \frac{E_2}{C}$
Face Angle	t_1	t_2	$t_1 - p_1 + r_1$	$t_2 = p_2 + r_2$
Cutting Angle........	v_1	v_2	$v_1 = p_1 - s_1$	$v_2 = p_2 - s_2$
Distance from Crown to Pitch Line	H_1	H_2	$H_1 = A_1 \times \sin p_1$	$H_2 = A_2 \times \sin p_2$
No Teeth in Spur Gear Having Diameter Equal to Twice the Back Cone Distance.	S_1	S_2	$S_1 = 2 \times P_d \times B_1$	$S_2 = 2 \times P_d \times B_2$
Chordal Thickness....	L_1	L_2	Table No. 2	Table No. 2
Corrected Pitch Depth	K_1	K_2	Table No. 2	Table No. 2
Angle of Tool Slides..	x		$\tan x = \frac{\frac{T_1}{2} + (E_1 \times \tan 20^\circ)}{C}$	

SOME POINTS ON THE DESIGN OF ALUMINIUM CASTINGS.

By H. W. Gillett.*

THE lightness, beauty, and resistance to corrosion of cast aluminium, combined with its good strength in comparison with its weight, are bringing it into vastly increased use in the best modern cars. All automobile castings must, of course, be designed primarily with a view to the fitness for the particular use intended. Usually, however, there are several ways of designing an aluminium casting, any one of which would allow it to serve its purpose well, keep within the weight limit and still be a good proposition from the point of view of the machine shop.

The problem is then to select the best possible design so that the casting may be produced in the best and cheapest way. Luckily the design that allows a casting to be easily and cheaply made is usually one that gives us a casting of far greater strength and reliability than one so designed that it is a difficult one to handle in the foundry. The designer is usually closely in touch with the machine shop and well acquainted with its problems, but seldom, indeed, is he a foundry expert ; could he realize the extra labour, and hence the extra cost and delay in production, a seemingly slight point in his design may make in the foundry, he would more often consult with the pattern maker and with the foundryman before completing his design.

Owing to certain physical properties of aluminium, such as its high contraction on cooling and its weakness when just solidified—that is, its hot shortness—aluminium castings require more careful design than almost any other casting metal. If one examines the defective casting records of the individual patterns in a large aluminium foundry one is struck at once by the vast difference in the results from different patterns. Some patterns give uniformly good results, while others differing but slightly from them are extremely troublesome. Speaking broadly, almost fifty per cent. of the discrepancies between the number of moulds put up and the number of good castings made may be traced to the door of the designer in one way or another, and about twenty-five per cent. more may be traced to the door of the pattern maker.

The designer and pattern maker are keen to see that what they can do to aid the foundry without interference with the requirements which the casting must meet, will come back to them many fold in lower cost and in regularity of delivery. But it so often happens that the automobile engineer will say : "We know that particular design is a poor foundry proposition and that pattern is not made in the best possible way, but we cannot change that model now nor can we tie up production of our cars long enough to allow changing the pattern," that it is worth while to remind you that the time to go into these points is when a new model exists in the mind and on the drawing paper of the engineer, rather than when the car is being assembled.

In passing from the molten to the solid state aluminium contracts a good deal ; when a heavy and a thin section come next to each other, the thin place will freeze first. If the thin section is so situated as to lie between a heavy section and a gate or riser, the supply of metal is thereby cut off from the molten mass in what is to be the heavy part of the casting. The contraction on freezing has to take place, and instead of taking

*Paper given at the June, 1911, meeting of the Society of Automobile Engineers.

place uniformly over this heavy part and maintaining the exact shape of the mould, it will often draw away from a corner and produce a "shrink." We can induce the heavy portion to freeze more quickly by placing a chill in the mould at that point (a chill being a piece of some material, usually a metal having a higher heat conductivity than the pot of the mould). It is difficult to accomplish the end completely by this method; it greatly increases the time required to

Fig. I.

Fig. II.

put up the mould and produces unsightly chill marks on the casting.

The ideal casting, therefore, is one of as nearly uniform a section throughout as is practical, since that means that the whole casting solidifies at the same time, so that contraction is uniform. On account of the hot shortness of aluminium the shrinkage strains set up when a heavy section joins a thin one often causes the metal to give away entirely at that point, and a crack appears.

If it is inevitable that light and heavy sections come together, the cooling strain should be distributed by joining the sections by a smooth curve, that is, a liberal fillet. This is on account of another physical property common to all molten metals. Suppose we have a sharp corner, as at the vertex of a right dihedral angle. The metal, of course, freezes first at the edge, crystals being deposited which tend to grow inward as shown in Fig. I. Each succeeding crystal finds it easier to attach itself to the end of the one previously formed, so that soon a line of crystals has grown by bisecting the angles. Now crystals in regular lines do not form as strong a mass as when they lie interlaced in helter-skelter fashion. It is easier to pull apart a pile of nails lying side by side than one which has been stirred up. On a smooth curve, as in Fig. II. there is no adventurous crystal which comes out first and to which the rest attach themselves like a swarm of bees. The crystals along the curve surface come out more nearly at the same time, and the next lot is deposited lying between the first ones in helter-skelter fashion instead of coming out in a line with the "follow-my-leader" effect. In other words, a sharp angle tends toward the formation of a definite line of crystals, which forms a sort of cleavage plane and is a source of weakness.

A break across a square bar of almost any metal will show distinct lines connecting the opposite corners, showing the arrangement of the crystals in this fashion, while a round bar shows no such lines. From this we can see the necessity for a liberal fillet. The sharper the angle at a change of direction, or the greater the difference of thickness at a change of section, the greater should be the radius of curvature of the fillet. If a second heavy part has a large boss and is near a first, both being joined by a thin section, the fillet should be more liberal that if only one were present.

There is no one factor in foundry practice that more gravely affects the strength of the casting than the pouring temperature. The reason for this is again the speed of crystallisation. The cooler the metal can be poured into the mould the more quickly it solidifies and the less time the crystals have to grow or arrange themselves, and the result is a mass of closely interlocking crystals forming a strong fine-grained material.

The effect of pouring temperature may be well shown by a set of test bars, all of which are cast from the same pot of metal with exactly similar moulds, the only variable being the pouring temperature. Such an experiment will show the rough surface and coarse fracture of the bars cast at the higher temperatures, as well as their low strength. The average results obtained from such a series of tests are given by the curve in Fig. III., which shows that the lower the pouring temperature is, the stronger is the casting.

Sectional thickness has a distinct bearing on design, since the lowest temperature at which a casting can be poured is that to which the thinnest section will just escape a misrun. If the casting is so designed that this crucial section forces you to pour hot, all the thicker parts will freeze too slowly and will be weaker than they should be. By slightly increasing the section of the thinnest parts, a casting could often be poured 100 degrees colder and the strength of the whole casting be increased at least 10 per cent. This is a matter entirely dependent upon the designer. If the bulk of the casting is from a quarter to half inch thick, one little part one-eighth of an inch thick will give us a resultant casting, on account of the high pouring temperature required, with average strength of about 16,000 pounds per square inch, instead of 18,000 pounds or over. The call for lightness has led many designers to overlook this vital point.

The very great influence of the pouring temperature is the reason why separately cast test bars show only the quality of the ingot metal and nothing at all as to the strength of the corresponding casting, even though the test bar and casting may be poured from the same pot of metal. The Standards Sub-Committee has wisely specified that aluminium test bars shall be made on castings. Were this stipulation not made the foundryman who wished could pour the casting as hot as he pleased, allow his metal to cool down, and then pour separate test bars which would then show an utterly fictitious strength in comparison with the casting.

The general lack of attention to pouring temperatures, not only in commercial practice, but in most of the investigations on aluminium, vitiates a great deal of the published data on aluminium alloys and accounts for a great many irregularities and seeming contradictions in the results. In comparing the different aluminium alloys, really comparable results can only be obtained by pouring at the same number of degrees above the melting point of the particular alloy in question in all cases, thus allowing the same time for crystallization and producing an analogous condition. Such an experiment led many designers to overlook this vital point.

After the designer has done his work it is the duty of the pattern maker to decide how the pattern shall be constructed. First of all, the pattern should be so made as to allow the use of moulding machines wherever practical. If a patternmaker is not too much bound down by tradition he can often simplify matters greatly. For instance, in a crank-case with several projecting pieces on the side, which would make it impossible to draw the pattern from the sand without the use of core work at the sides, by the simple expedient of making the pattern hollow and putting in a lever by which the pieces are drawn into the body of the pattern and the pattern then lifted out, a large amount of core work was eliminated.

Core work always means trouble. It takes time to set cores in the mould correctly, and if a lot of small cores are used the danger of shifts is greatly increased. If on the other hand large cores are used they must be made hard enough to allow handling them and setting them in the mould, which requires not only a solid core but one reinforced by iron rods and wires. This makes them hard to crush, and, with large cores inside of thin walls of metal, introduces danger of cracking. When we have a core completely surrounded by walls of metal it is a question whether the tensile strength of the metal as it solidifies is greater than the compressive strength of the core. Let the core be ever so slightly too hard and your casting is inevitably scrap. We have seen patterns requiring large and complex cores within walls as thin as 3/16 of an inch, from which patterns it was absolutely impossible to make castings, since any core strong enough to place in the mould would have too great a compressive strength for the metal to crush without cracking the casting.

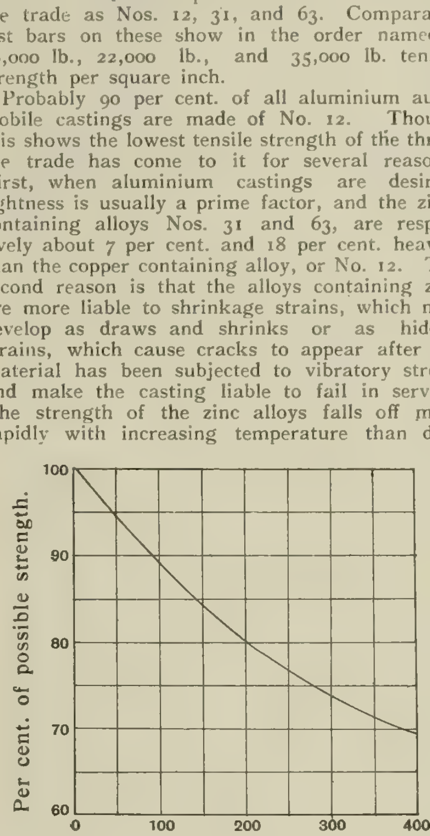
If cores must be used, the core prints should be large and deep so as to anchor the cores firmly without the use of chaplets to hold the cores in place, since it is impossible for the molten metal to fuse a chaplet into the body of the casting, without pouring at a temperature far above that necessary to give the greatest strength. When a job requires cores, the first question that should be asked by the pattern maker is if that pattern cannot be made so as to allow the use of green sand core, or at least a green sand half. Green sand will crush and give away when the casting contracts on cooling, where a hard, dry sand core will not crush and will crack the casting.

The foundry is called upon to make castings of ever greater complexity, and the core work required has increased amazingly in the last couple of years. Core rooms once fully adequate for a given moulding room are now being overcrowded and unable to cope with the demand for cores. This situation could be greatly improved if the pattern maker would put green sand halves in all castings where it is possible to use them; and the results would be both cheaper and better.

If a considerable number of castings are to be made, wooden patterns and core boxes are an abomination. They warp and swell in use, dry out and crack apart in storage, and wear out by abrasion from the sand and by the constant rapping necessary to allow the pattern to be withdrawn from the mould or the core box from the core. Good metal patterns and core boxes are an economy if many castings are to be made, or if a few are to be made from the same pattern in successive years.

With proper design and a well made pattern the engineer who wants an aluminium casting must next face the problem as to what aluminium alloy he shall specify his casting to be made from. He has a choice of practically three alloys—one containing 8 per cent. copper, one with 3 per cent. copper and 15 per cent. zinc, and one with 35 per cent. zinc, designated respectively as alloys Nos. 1, 2 and 3 in the Standards Sub-committee specifications and known to the trade as Nos. 12, 31, and 63. Comparable test bars on these show in the order named—18,000 lb., 22,000 lb., and 35,000 lb. tensile strength per square inch.

Probably 90 per cent. of all aluminium automobile castings are made of No. 12. Though this shows the lowest tensile strength of the three, the trade has come to it for several reasons. First, when aluminium castings are desired, lightness is usually a prime factor, and the zinc-containing alloys Nos. 31 and 63, are respectively about 7 per cent. and 18 per cent. heavier than the copper containing alloy, or No. 12. The second reason is that the alloys containing zinc are more liable to shrinkage strains, which may develop as draws and shrinks or as hidden strains, which cause cracks to appear after the material has been subjected to vibratory stress, and make the casting liable to fail in service. The strength of the zinc alloys falls off more rapidly with increasing temperature than does



Degrees above lowest pouring temperature.

Fig. III.

that of No. 12, so that the alloys which are stronger at ordinary temperatures, when subjected to the heat developed about a motor, may lose their seeming advantage.

The zinc alloys, moreover, are more brittle and less ductile than the No. 12, as well as more likely to break down in vibration or under repeated impact. Although so eminent an authority as Mr. Souther disagrees with this, yet there must be a reason for the very marked abandonment of the zinc alloys for the copper alloy throughout the industry. Mr. Souther's figures are the same he published several years ago, and on looking over the fuller data in this earlier paper it seems to us he must have drawn conclusions from too few tests and probably from tests on bars not necessarily cast at corresponding temperatures.

We expect to go into this matter of endurance of different aluminium alloys more fully in the near future and plan to report the results at a future meeting of the Society. A large number of tests so far have shown that on the White-Souther Endurance Machine the different alloys run about as follows:—

No. 12—over a million revolutions
No. 31—about 600,000 revolutions
No. 63—about 500,000 revolutions

before fracture. These are averages of a large number of bars. You will note that the resistance to vibratory stress is in opposite order from that of the tensile strength, but exactly in the order of the ductilities.

On another type of endurance machine where the bar is subjected to repeated blows, we understand the No. 12 will stand about a million blows

before rupture, while No. 31 will only stand 1,500 to 2,000.

The brittleness and unreliability of No. 31 and No. 63 is usually laid to the presence of zinc. On the other hand it is more likely due to the absence of aluminium—that is, alloys high in aluminium are resistant to vibratory stresses, while alloys low in aluminium are less resistant. An alloy containing 88 per cent. aluminium and the rest zinc gives nearly as good results in the endurance test as does No. 12 with 92 per cent. of aluminium, but since the tensile strength is decidedly inferior to No. 12, and it is a heavier alloy, this is no longer a commercial alloy.

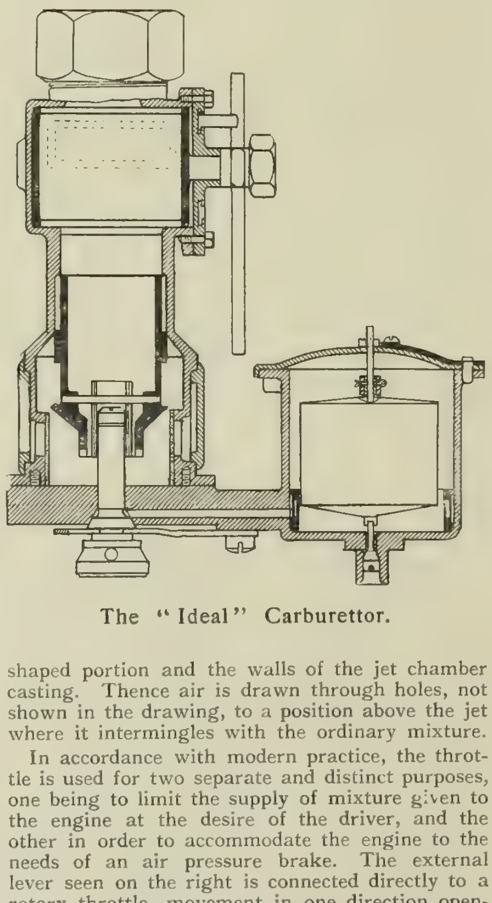
Magnalium, or 94 per cent. aluminium and 6 per cent. magnesium, is about as strong in tension and has about the same resistance to vibratory stress as has No. 12. To get strength enough to pay for the added weight where zinc is used as the chief alloying metal, we are forced to use so much zinc that the ductility and resistance to vibratory stress are cut far below the figure for No. 12. We really have three factors, strength, lightness, and resistance to vibratory stress. If we give each factor what we might call a co-efficient of importance and multiply the three together, the alloy which gives the highest product will be the most valuable. The automobile engineer has practically made this calculation and he has answered the question by specifying the 8 per cent. copper alloy in the vast majority of cases.

The engineer who wishes to get good reliable aluminium castings and at a reasonable price must then realize that the problem is not entirely one for the foundryman to whom he entrusts the handling, but that it is his problem as well. The responsibility for design is solely his, and if he will so proportion his castings as to avoid great inequalities in section or sharp changes in direction, and if he will use liberal fillets, if he will keep away from such thin sections that the metal must be poured too hot to get strong castings, if he will instruct his pattern maker to go as far as he can in avoiding dry sand cores, if he will provide good substantial patterns, will specify the proper alloy and then see that the casting is made with proper care and with proper control of pouring temperatures, he will get castings which will allow him to utilize to the fullest extent all the many advantages which aluminium has for his purposes. But he should not for one moment forget that unless the design is right the best efforts of the foundryman cannot produce a casting which will have in the fullest degree those advantages which the perfect aluminium casting should have.

THE "IDEAL" CARBURETTOR.

The majority of modern carburettors are open to the criticism of inaccessibility, especially as regards those parts which are likely to need adjustment or attention during running or on the part of the private motorist. It is refreshing to notice that a great number of the latest designs in carburettors embody features which are likely to render them particularly accessible in every portion one could desire. Herewith is a sectional diagram of the "Ideal Carburettor" which has quite recently been placed upon the market. Probably this carburettor has as many novel points as any which has heretofore been set before the public. Taking its component parts in the order in which the petrol progresses to the engine, one finds that particular attention has been paid to the rapidity and ease with which the float can be removed from the float chamber. This is accomplished by simply removing the lid of the chamber, which has a bayonet joint and is locked by a spring controlled peg seen at the top of the drawing, and immediately on the removal of the cover, a split pin is withdrawn from the top of the float, whereupon the float and its needle can be removed immediately. Certain adjustments for carburettor level can be effected by means of the collar through which the split pin is placed. Contrary to usual custom an efficient filter is provided before the float chamber petrol leaves on its way to the jet. This takes the form of a small gunmetal casting slotted and fitted with a gauze screen of the usual type, the whole attachment sliding down to the bottom of the float chamber until the petrol can only reach the jet pipe via the wire gauze. Instantly, of course, the gauze can be removed, without any tools, for inspection and cleaning, a point which in itself would be worth a considerable amount to the average private user. In the jet passage there is nothing uncommon, but the jet itself is distinctly unusual. In the first place it will be seen that a species of cone fastening is used to secure the jet to the bottom of the carburettor casting. A flat steel spring, fastened to the casting by a set screw, sockets into the flange on the jet itself, and sufficient spring action is obtained to force the jet always against its taper. To remove the jet all that is necessary is the sideways displacement of the flat spring, whereupon the jet will come away in the hand. One would be led to imagine that such a device, accessible as it may be and commendable on the ground of simplicity, might eventually lead to trouble through leakage at the petrol tight seating, but we are informed that this trouble has never occurred. At all events, no jet can be removed with such ease and rapidity. Another characteristic of the jet is that the base is provided with a scale which can be set against a fixed point provided on the non-movable part of the jet. Rotation of the scale corresponds exactly to the jet opening above, and in this way it is possible to set the area of the jet with the greatest ease and simplicity. In the drawing the slot immediately below the head of the jet is the actual petrol exit, and this slot can be closed altogether or widened in accordance with the movements of the aforesaid scale. As will be seen in the diagram the carburettor embodies the shrouded jet principle concerning which much has already been said. The shrouding is held in place in the slide by means of a longitudinal peg, socketted in the brass casting on each side, and this brass casting is used to obtain a sufficient supply of air varying more or less directly with the engine suction. At ordinary or at slow running speeds, air is taken past the jet at the bottom of the casting, and through the space between the shroud and the jet.

As a further instance of the care which has been taken to prevent the ingress of foreign matter to the engine, another wire gauze filter is provided for the air which reaches the jet. This is secured in position by two spring steel rings which can readily be removed without trouble. Before reaching the gauze, air traverses completely round the jet chamber inside a swivel casting seen in section on the drawing. The idea of allowing a swivel motion to the casting is in order that the air pipe can be placed in any position which is most suitable for the engine to which the carburettor is fitted. When once in position it is locked by a couple of set screws, which contract the split coupling. On an increase of engine suction the whole of the inner casting containing the shroud, and sectioned in black on the drawing, moves upwards, and thereby increases the gap between the cone



The "Ideal" Carburettor.

shaped portion and the walls of the jet chamber casting. Thence air is drawn through holes, not shown in the drawing, to a position above the jet where it intermingles with the ordinary mixture.

In accordance with modern practice, the throttle is used for two separate and distinct purposes, one being to limit the supply of mixture given to the engine at the desire of the driver, and the other in order to accommodate the engine to the needs of an air pressure brake. The external lever seen on the right is connected directly to a rotary throttle, movement in one direction opening the throttle until the full area of the inlet pipe comes into operation. Further movement in the same direction throttles down the supply of mixture at the lowest point of the throttle, while opening the air port, seen in dotted lines, to the upper part connecting directly with the engine. A reversional movement shuts off the air supply to the engine and re-opens the jet pipe. Two set screws are provided in the base of the carburettor which can be adjusted in slots so that the throttle lever will face in any direction suitable to the mechanism to which it is to be attached. It is probable that very few carburettors have so many novel, interesting or useful points as this particular one. Very few are so accessible at every point, and the provision of large filters both for the air and petrol within the carburettor itself cannot be too highly recommended.

MISCELLANEOUS.

ELECTRIC LIGHTING.—We have received a copy of a pamphlet on the Remy Magneto Light, from the Remy Electrical Co. The book is interesting on account of the full description of a lighting and ignition system which is contained therein. With the Remy magneto a direct current ignition is obtained of high voltage, while an arrangement is fitted by which the lighting circuit comes from the same machine and is automatically governed as to pressure. A full explanation in non-technical language is embodied in the book, coupled with a series of half-tones in order to elucidate the harder points. Sectional drawings of the full machine are also provided in order that the full scheme may be plain.

CABLES.—W. T. Henley and Co. have issued a large and complete new catalogue dealing with every wire and cable manufactured by that firm. Commencing with electric supply mains, tables are used setting forth every particular which could possibly interest the user, and enable him to order his particular requirements as easily as possible. Standard Association conductors are then listed in every size and in every possible form manufactured. The whole list is carefully and well got up, setting forth its information in a useful form and, without giving undue prominence to any one type of cable, yet makes the choosing of some particular brand an easy matter.

SHAFTING CLUTCHES.—The Unbreakable Pulley and Mill-gearing Co., has issued a book dealing exclusively with a type of frictional clutch known as the Benn, and suitable for line shaft drives in machine shops. Throughout the whole booklet there are many extremely interesting diagrams of the whole sectional arrangements and applications of this clutch, while the latter part includes power transmitting tables to enable a customer to choose his particular type.

IN REGARD TO THE INFORMATION which the Institution of Automobile Engineers has collected as to the practice of the various firms in the matter of apprenticeship, we are requested to announce that the Council wish to make it clear this information, whilst it has been tabulated for convenient reference by the staff of the Institution, is not in such a form that it can be circulated to the public generally, an impression which seems to have been conveyed by the previous announcement of the scheme. It is therefore desirable that parents and others should call at the offices of the Institution by appointment, in order to avail themselves of the information, although, if this is not possible, advice will be given by letter.

BINDING COPIES.

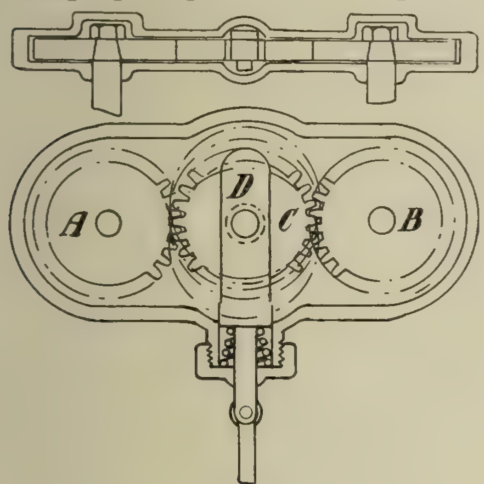
We take this opportunity of informing readers that we have only a very few copies of the first two issues left, but these will be supplied to any desirous of keeping a file for binding purposes, although it has been necessary to fix special prices (which may be obtained on application) for certain of the issues which are most scarce.

RECENT AUTOMOBILE PATENTS.

By Eric W. Walford, F.C.I.P.A.

Magneto Timing.

It seems curious that provision is not generally made for varying the timing of the magneto and at the same time ensuring the spark taking place when the armature is in its best electrical position. Constructions have been suggested for adjusting the armature spindle relative to its driving shaft and the invention now described is of this nature. The armature spindle is shown at A and is driven through gearing from the driving shaft

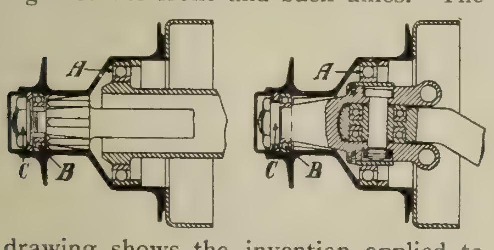


B, the gearing taking the form of three spur wheels of some ordinary type. The central spur wheel C is movable vertically by sliding its carrier D by means of the control lever. As the wheel C is moved the angular position of the driving and driven shafts A and B is varied and consequently the magneto can be advanced and retarded without the resulting igniting spark being affected. However, the variation in tooth clearances brought about by this action would seem to provide possibilities for noise and rapid wear.

No. 20,531/10. B. R. Raggett, A. D. Maclean, and C. Murray.

A Detachable Wheel.

In this construction of detachable wheel the wheel bearings are removable with the detachable wheel element, so that in the event of bearing damage the change can be easily effected. It will be understood that the bearing arrangements on the front and back wheels differ considerably, so that it is a matter of some difficulty to provide a construction of detachable wheel which will provide interchangeable bearings for the front and back axles. The



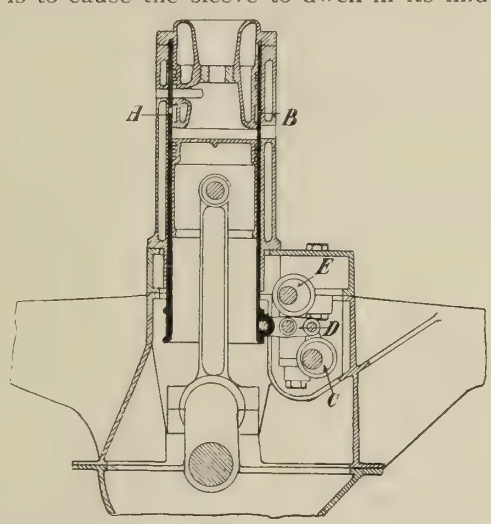
drawing shows the invention applied to both front and back axles, and it will be seen that the inner bearing A fits on to both axles, but that the outer bearing B is only used as a bearing when the wheel is fitted to the front axle. When fitted to the back axle the bearing is still carried by the detachable wheel hub, but it does not fit on the driving axle, and does no work. The securing nuts C are different in the two cases. In the case of the back wheel it forces the hub into place, whilst in the construction for the front axle the securing nut forces the outer bearing B on to the steering axle. This design does not appear to lend itself to the inclusion of a thrust bearing and might be criticised on this score.

No. 18,630/10. F. W. Lanchester.

A Single Sleeve Valve Motion.

The sleeve is provided at A with an inlet port and at B with an exhaust aperture, these communicating with the respective inlet and outlet passages. The ports normally lie at or about the positions illustrated, and it will be understood that the sleeve is raised to move the port A into line with the inlet passage and lowered so that the port B comes opposite the exhaust outlet.

This movement is obtained by the use of two eccentrics, one of which, C, running at engine speed, actuates the free end of a rocking lever D, the other end of which is pivoted to the sleeve, whilst the intermediate portion is acted on by an eccentric E, running at half engine speed. The action of the two eccentrics is to cause the sleeve to dwell in its mid

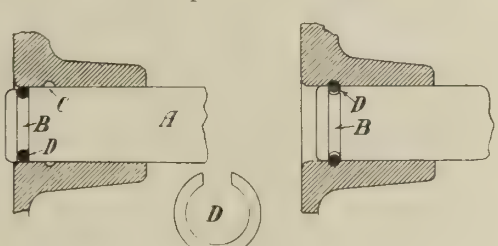


position during the compression and firing strokes, and then to rapidly move downwards and upwards.

No. 14,700. F. W. Lanchester.

Gudgeon Pin Attachment.

The gudgeon pin A is provided with a groove B, as also is the piston lug at C. Before the gudgeon pin is pushed quite into place a split keeper ring D is inserted into the groove B and the pin driven home into its normal position as shown in the

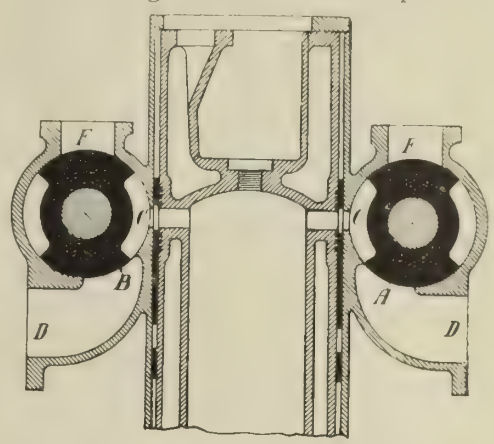


right-hand drawing. When in position the copper ring springs outwards so as to engage the groove C in the position. As will be seen, it lies half in each groove and effectually prevents accidental endwise movement of the gudgeon pin. To remove the pin the other end is given a sharp blow, causing the split ring D to be forced out of the groove in the piston.

No. 21,435/10. Birmingham Small Arms Co., Ltd., and R. Nicholls.

Rotary Valve Engine

This engine is provided with rotary valve members A and B on each side, these having recesses C which put the

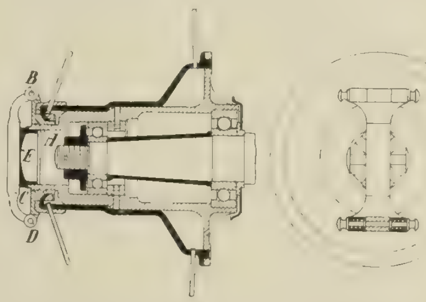


cylinder ports into communication either with the exhaust outlets D or the inlet passages E. It will be understood that inlet or exhaust takes place at both sides of the cylinder. To relieve the valve members A of pressure during the firing and compression strokes a single sleeve valve is employed, this being provided with ports at two different levels so that the cylinder passages are opened at the right times, the sleeve valve being used for timing and the outer rotary valves as distributors. This engine somewhat resembles the Henriot, on which it seems to be an improvement.

No. 15,840/10. M. E. Tunscombe.

A Detachable Wheel-Locking Device.

In the construction of wheel illustrated the detachable hub is forced on or off the fixed hub by screwing or unscrewing the collar A. In this case this has pivoted to it at B a hand lever C, which can be raised so that it forms a handle for unscrewing the sleeve A, or it can be lowered into the position shown. In



the lower position the free end is locked to the sleeve A by a spring bolt at D, whilst the middle portion engages a groove in the hub cap E, thus locking the lever C and preventing accidental rotation of the sleeve A. Spring bolts are provided at both points B and D so that either end of the lever can be freed by hand and the wheel removed, attached and locked without the use of tools.

No. 5,590/11. L. Giradot.

THE DEFIANCE FLYWHEEL BALANCING MACHINE.

In the article on "American Touring Car Design" which appears on pages 454 to 456, reference is made to the machine illustrated below and its value is discussed briefly there. It is by the courtesy of the Austrian Daimler Motor Company that we are able to reproduce this photograph, which was taken in the Vienna works, but the machine is precisely similar to that used in the Daimler shop at Coventry and in many American works. It will be seen that the whole apparatus is simple, and it is as easy to use as it is free from complication in design.

The Defiance Rotational Balancing Machine.

The vertical spindle is driven by friction bevels and terminates in a point, which supports the weight of the flywheel to be balanced, rotation being given to the wheel by means of the driving pins seen projecting upwards from the face plate. In operation the flywheel is first speeded up and when running at a fair rate the drive is cut off and the machine allowed to revolve by its own inertia. It will then usually be found that there is an oscillation of the flywheel owing to lack of balance. The instructions for the use of the tool are as follows:—

Before rotating, locate near the centre of the interior of the rim a weight of the size necessary to produce a standing balance, then rotate, and mark upon the edge of the rim with a pencil of moistened clay. If the mark occurs within one-fourth of the circumference of the wheel, from the weight and in the direction of rotation, raise the weight and at the same time advance it toward the mark. If the mark occurs at more than one-fourth the circumference from the weight, depress the weight and retreat it from the mark. If, by these steps, the edge of the wheel be reached, and the wheel still runs out, increase the weight and place a counter-weight diametrically and transversely opposite. Care must be taken, at every adjustment of the weights that a standing balance is not violated. If, upon the readjustment of the first weight, the marks occurs at one-fourth of the circumference of the wheel from the weight, nothing more can be done with this weight, but an additional weight must be placed at the mark and, if this second weight disturbs the standing balance, locate a counter-weight, as in the first described condition.

In a case in which it is not permissible to attach weights, but in which excess of material must be cut away, it is obvious that an inversion of the above reasoning must often be pursued, and frequently the use of trial weights facilitates the solution.

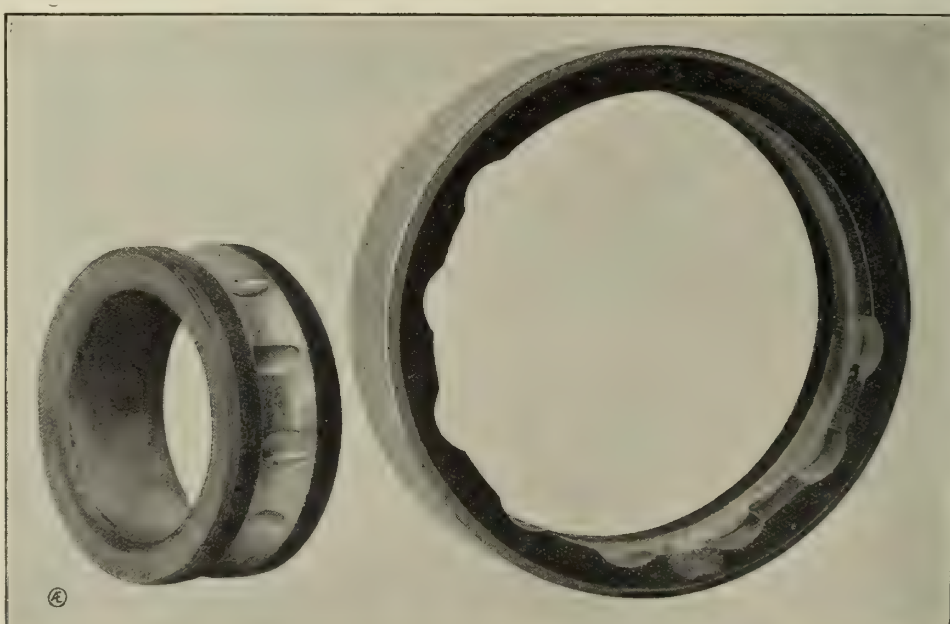
In practice it is found that a workman soon learns where to drill away metal, and the procedure of attaching weights by means of dabs of clay need not be followed.

A CASE OF EXTRAORDINARY BALL BEARING WEAR.

The accompanying illustration shews the inner and outer races of a large ball bearing which has been used for crankshaft work. It is perhaps scarcely credible, but the extraordinary grooving of the inner race occurred while the bearing was in use, and it was of course on account of the grooving that it came to be discarded. The grooves were at their deepest on the side of the inner ring seen in the illustration, but markings could be seen all the way round the ring corresponding in distance apart to the distance between the balls.

It is perhaps not too easy to see how this peculiar form of wear could have taken place, and it should first be pointed out that the bearing was theoretically quite large enough for the work it had to do. Indeed the race surfaces are in quite good condition except at the grooved points. Really the illustration is a fine object lesson regarding the importance of careful and intelligent fitting when ball bearings are to be used for heavy loads. A moment's reflection shews that the inner ring and outer ring would always be in the same relative positions on the instant of ignition, within the limits of range of spark which are, of course, extremely small. Obviously too the grooves could only be cut by the balls owing to the latter moving sideways across the race, which action can only take place by the balls coming out sideways into the slots in the outer ring through which they were first inserted. Supposing, therefore, that the outer ring was fitted slot downwards in the crankcase, then every time the balls happened to be just opposite the slots at the moment of ignition, they must have been driven out sideways with incredible rapidity, springing back into place almost before any revolution had taken place. That this was so can be observed by examining the edges and grooves, which are quite sharp, it being necessary to examine them with a lens to notice any rounding.

Several lessons are perhaps to be learnt from this photograph. Firstly, it gives striking indication of the immense pressure between ball and race during the instant of explosion, when a ball bearing is fitted to a crankshaft. Secondly, it shews that a slotted ring bearing ought to be mounted slots uppermost, and, though it would not be fair to say that this type of bearing is unsuitable for crankshaft work, there will appear to be good reason to prefer the use of a different pattern; if indeed ball bearings are to be used for crankshafts at all. Thirdly, the fact that such wear could take place without the total destruction of the bearing is proof of the quality of material used in its manufacture, and it should be added this particular bearing is of the casehardened or carbonised variety. Fourthly, a closer examination of the bearing shews not only the grooves on the near side, but behind are corresponding faint dull marks shewing that slight wearing away had begun there also. To render this possible the inner ring must have tilted slightly inside the outer, and it is therefore to be presumed that very considerable crankshaft whipping took place, in fact crankshaft whipping, by tending to twist the inner ring relative to the outer, would tend to shoot the balls out into the groove in the precise manner in which they obviously were shot.



Extraordinary Ball Bearing Wear.

THE

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CONTENTS.

THE FUTURE OF THE SMALL CAR.

NOT very long ago the trend of affairs appeared to indicate that touring cars will become sub-divided into three main classes, these being large four and six cylinder cars giving anything over about 30 brake horse power, smaller four cylinder cars running down to about 20 actual horse power, and quite small cars giving up to 12 or 15 horse power with either single or two-cylinder engines. Since this time, however, the demand for the really large cars has fallen off at a most surprising rate, the medium-sized car having proved its capability to do almost as much as the large car. At the same time, the middle class of car has, so to speak, grown downwards, and next season will disclose a market almost entirely filled by four-cylinder cars, with a cylinder bore of from 60 to 90 millimetres. The prices of the smallest four-cylinder machines are now so low that the two-cylinder car can show but little advantage, and even some of the old and more successful amongst the single-cylinder machines cost almost as much as a modern diminutive four-cylinder.

Running through a list of European makers, it is to be noticed that extremely few are able to supply any kind of complete car for less than about £250. Between this sum and the £80 or so, which is the average price of a really good passenger motor-cycle, there is an enormous gap. Men who have the ability to lay out from £100 to £200 on the purchase of a car have an extremely narrow choice, more especially if they limit themselves to British or European vehicles. Of course many attempts have been made to produce the much-talked-of £100 car, but none of them can be said to have encountered much success. Experience shows that a single-cylinder car is even more susceptible to the evil influence of cheap construction than a four-cylinder. The only single-cylinder cars which have given their owners any lasting satisfaction have been a few which were thoroughly well made, and therefore commanded prices in keeping with their quality. Now there is no denying the fact that a four-cylinder car, even if it is constructed cheaply, runs when new with much greater smoothness, quietness and general comfort than the best of single or even of twin-cylinder chassis. Thus, as four-cylinder cars are obtainable for smaller and smaller sums of money, it appears that the demand for single and two-cylinder types would dwindle almost to vanishing point. There is every probability that a really well-made single-cylinder car, developing about 8 or 9 h.p., would outlast a small four-cylinder vehicle of equivalent price, but the difference in the cost of manufacture is by no means so great as might be expected.

The intermittent impulse and the lack of balance in the single-cylinder re-act upon the whole of the rest of the mechanism, which must be strengthened accordingly if it is not gradually to develop a looseness at every connection. The advantages of the single-cylinder engine, apart from cost of manufacture, are its economy of fuel and its economy of oil. Undoubtedly a sixty millimeter four-cylinder engine is not highly efficient, but there is no comparison between its smoothness in running and that of an equal-powered single-cylinder. *If*, however, a four-cylinder car can be made and sold profitably for £200 or thereabouts, it is to be questioned whether fresh competitors in this particular field are not in error in using too small engines. The difference in cost between a sixty millimeter and an eighty millimeter engine is quite out of proportion to the difference in efficiency and power. The cheap American car has been produced almost universally by first ensuring the sufficiency of motive power, and then cheapening the chassis by an elaborate process of elimination, every detail that can be dispensed with being cut away, but always so as to leave the essential power and driving units amply large and amply strong.

The American is, of course, assisted very greatly by the large numbers of chassis he is able to manufacture, and few people will be ready to believe that a good four-cylinder car could be sold profitably for less than £200, or even so little as this amount, made in quantities suitable for our home market only. It is more than doubtful whether a demand for cars which are not good, but merely passable, is worth serious attention from manufacturers who have already good reputations and good businesses, but this is a question that our home manufacturers will have to consider very seriously. Cheap foreign four-cylinder cars, by reason of their power and smooth running when new, have opened up a new market that British makers have never secured successfully with single and twin-cylinder little cars, though they have been trying for years.

Of course, it may be found that a soundly-made British 60 mm. car is preferred to a rougher and cheaper 90 or 100 mm. foreign production, but the fact must be faced that the larger reserve of power in the latter cannot fail to have its effect on many purchasers. Still, if our own makers have serious regard for their lasting fame, they will probably be wise enough to let cheap trade alone. The cheap machine proved to be anything but a good speculation in the cycle trade, and unless motor manufacturers go very warily they will drop into the same pitfall as those which ensnared their colleagues of the bicycle industry. Foreign competition with cheap goods is always best met by retaliation with a less pretentious and more durable article—in the long run, and it is the long run that matters with such articles as motor cars.

AMERICAN MANUFACTURING.

Being the third of a series of articles on the American industry by a member of "The Automobile Engineer" staff who has recently made an extended tour of the North Eastern United States.

BEFORE commencing to consider American methods of manufacture in any sort of detail, it is necessary, firstly, to grasp the wide difference between the American industry and our own. The American manufacturer caters for quite a different sort of market to that which the European maker supplies, this being partly owing to the different conditions of life in America, and partly owing to the temperamental peculiarities of the British and American people. In my previous articles on American touring car design, direct comparison has been made between European and American practice, and the conclusions were perhaps rather to the disadvantage of the American car. It must not be forgotten, however, that I am writing entirely from a British standpoint, and it is doubtful whether an American could ever see the matter from quite the same point of view, however wide his knowledge might be concerning European work. First and foremost, of course, nearly all American machinery is finished only roughly where finish serves no useful purpose. In the ordinary American car no time is wasted by sand-blasting the outsides of castings, so as to give a smooth surface, or in polishing any parts which subsequently will be painted. Instead of the highly polished piping usually employed in Europe for water or gas conduction, one finds rough castings, and this sort of thing is irritating even to a man who is perfectly well aware that finish at such points is of no practical utility. On the other hand, however, the average American appears actually to prefer rough finish. Just as an American machine tends to give a British engineer the impression that its creator took no pride in its appearance, so does a highly finished British article impress an American, not so much with its intrinsic beauty as with the amount of time which must have been wasted on unnecessary scraping and polishing. Commercially, there is, of course, a good deal of common sense in the American view, and this point has been elaborated merely because it ought more clearly to be understood that American manufacturers do without finish, not because by avoiding it they can cut down the price a few dollars, but because the American user does not demand finish. Therefore one finds many American cars of very third-rate appearance made of first-class material by first-class workmanship.

Again, another point which affects the nature of manufacturing is the enormous size of the average American maker's output. We here are accustomed to seeing in print estimates of American production so huge that they are generally looked upon as wild exaggeration, but whatever the totals may be the fact remains that outputs of less than five thousand cars per year are regarded in America as quite small business. Undoubtedly many firms in Detroit are, for the greater part of the year, regularly turning out anything from twenty to sixty cars a day, and in some cases figures much higher than these are quoted on very good authority. It is the writer's belief that from the beginning of 1910, up to the end of the present year, the total number of cars that have been made, and will be made, in Detroit alone is at least equivalent to the output of *the whole of the rest of the world* during the corresponding period. While this statement may be a little difficult to believe it has not been arrived at without very careful estimation, and by Detroit the country in the immediate neighbourhood is, of course, included, so as to take in the large plants at Pontiac, and a variety of smaller factories not actually in the city of Detroit.

Classification of the Industry.

Broadly, the American trade may be divided into three classes. First of all, there is a comparatively small number of firms making large and expensive cars, almost invariably with six cylinders capable actually of developing from 40 to 120 h.p. It is in this class that the best American work is to be found and the highest finish, while the importance of this section of the trade may be gauged by the number of men employed in the production of large cars only. Thus the Pierce Arrow Company, the Locomobile Company, the Lozier Company, and the Peerless Company, mentioning only four out of a large number who are specialising on large car business, would employ between them at least 20,000 men. The average price of the cars turned out would be, at a very rough estimate, £800.

It might perhaps cause surprise that such large quantities of such costly vehicles can be disposed of even in America. However, there is one point that affects American outputs enormously, and it is one which I have never seen mentioned before, this is the difference in the value of money in the two countries. Not long since our own Government published a book dealing with the incomes and expenditure of the American working class, and, speaking from memory, I believe the calculations arrived at were that the average income was nearly three times as great as it is here, while the average expenditure on necessaries was about two and a half times as great. This seems to be a fair proportion, not only for the working classes, but for ordinary professional men and business men. A man who would be earning or making £500 a year here would probably make £1,500 in America, and certainly would not make less than £1,200. While this is so, the price of cars is about the same as here on actual rates of exchange, so if we look at the matter another way and assume that the American dollar is only worth 1s. 6d., instead of its nominal 4s. 2d., a 1,500 dollar (£300) car actually sells for £112. A moments' reflection will show that if exactly the same cars that are now made for £300 could be sold here for £112, an enormously greater number of them could be disposed of. In America the home population, together with that of Canada and Mexico, is at least one hundred and twenty million, even excluding districts which, by reason of their wildness, may be left out of the calculation. Probably ten times as great a percentage of the population there are practical possible purchasers of cars, so it is quite conceivable that it will still be some considerable time before the American industry begins to over produce.

However, to return to the sections of the trade, after the large car makers there come a quite small number of firms making really first-class four-cylinder cars, and here, of course, the Packard Company is much the largest and best known. The undoubtedly great reputation of this business has been built up on the four-inch 30 h.p. car, more or less. One or two of the big six-cylinder makers have models to compete with the 30 h.p. Packard, and there are a few smaller firms also catering for the same section of the public. So far, all the makers down to this point do not give very much consideration to the price of their cars. They aim to supply the most satisfactory vehicle possible with all the comforts and conveniences yet known to the American user, and the price is fixed to give a reasonable profit over and above the cost of production. So far, however, as I was able to ascertain, there are no small cars whatever made on this principle, the small car always being of the very cheapest class. However, after the Packard class of makers one comes to the biggest section of all; making four cylinder, usually 4 in. bore cars, and paying varying degrees of attention to quality and price. It it not very easy to pick out typical examples in this class. The Cadillac perhaps is one of the best cars which is made with an eye to the cost, and the Cadillac output is amongst the largest in Detroit. Then, again, come a few more names known in this country. There are such firms as the Chalmers, the Hudson, the Speedwell, the Stoddart-Dayton, and a host of others far too numerous to mention. Last of all come the firms who consider price above everything, and in this category it would perhaps be well within the capabilities of my readers to think of some examples.

Manufacturing, therefore, or rather perhaps design, divides itself into three sections. The first is high-class design, in which the car is drawn out to be as good as possible. The second class is where the design is made to be thoroughly sound, but with due consideration to the cost of producing each part. The third class is what might be called machine shop design, in which cheap manufacturing is the first consideration, and quality receives only enough consideration to enable the car to make a fairly good showing in use. In some ways the last class is most interesting, the dodges resorted to for cutting down costs being frequently extremely ingenious. It would not serve much useful purpose to cite these at any length, because it is to be hoped that the British trade will never undertake a class of work from which no lasting reputation can possibly be obtained, but just one instance perhaps may be given to show the sort of way in which money can be saved. In a certain cheap four-cylinder car, the upper half

of the base chamber is cast iron, in one piece with the cylinders. The camshaft has three bearings, and these are supported by lugs on the casting. These lugs have cored holes and, when the rest of the engine is assembled, the camshaft is held in place by a jig, while white metal is poured in round it. This forms the camshaft bearings *in situ,* and is accomplished in a few minutes at a cost of next to nothing over and above the white metal, of which naturally, only a very small quantity is used. These bearings seem to be quite satisfactory for a long time, but of course, to renew them would be a decidedly inconvenient undertaking, while if the camshaft itself sustains any damage the method of its removal would be a matter for considerable deliberation.

American Companies.

Turning to quite a different phase of the subject, one finds, too, that the constitution of American companies is generally quite different to that of English concerns. Speaking quite generally, of course, one finds that the number of shareholders in an American automobile concern is quite limited, the capital often being divided up amongst a very few men, all of whom take a more or less active interest in the conduct of the business. Thus the Pierce Arrow Company—one of the largest in America—is said to have only six shareholders, and the president of the company holds nearly a million pounds' worth of stock. The chief difference which this sort of constitution makes is that the principal officials of the company are more vitally interested in the financial success of the concern than managing directors and general managers usually are in this country. Here it is not unusual to find a managing director with very considerable power, whose salary amounts to much more than any possible interest from his holding in the company. In America, more often the president, who decides the policy, will own a very considerable portion in the company and will therefore be very anxious to make it pay as well as possible.

The responsible staff in an automobile factory are generally more or less as follows :—The treasurer, who corresponds to the English chairman of the board of directors, but who does considerably more active work. The president and vice-president, who are really managing directors, dividing between them the duties of general manager and secretary, the treasurer, by the way, doing a certain amount of the work of this latter functionary. In a very large works probably there will be a general manager coming next in order of power to the vice-president, and really acting as presidential deputy, but, in a small works, the vice-president attends to all the details of general management.

This completes the list of company officials, and below the management the responsibility begins to be sub-divided. The sales department exists very much as it does here, and the advertising department likewise. The actual production, however, is controlled differently and by two entirely separate departments, one being under the Chief Engineer, and the other under the Factory Superintendent. The Chief Engineer is responsible for the whole of the designing of the cars, and usually has an extremely free hand. The factory superintendent controls the whole of the manufacturing, and is responsible for making the chief engineer's designs in the best possible way, and at the lowest possible cost. Of course, both chief engineer and factory superintendent often take considerable interest in each other's work; in fact, I should say that as a rule they work together fairly closely, but it is a very usual thing in smaller works to find that the chief engineer is practically the superior of the superintendent, assuming responsibility for both design and production, and particularly is this the case in works where a good deal of stuff is bought finished. I do not, however, propose to consider this class of establishment until I have said a little more concerning the fully-equipped factories, but in America it is not easy to find a car that is made entirely by its nominal producers. Even the Packard Company, with their eight or nine thousand hands, their 8,000 h.p. power plant and their extensive stamping and foundry department, still find it economical to buy a few quite important parts.

However, in a large works the system followed is first for the chief engineer to produce a design—and a first design will usually be made with two or three alternatives. These cars will then be produced perhaps in the general factory, and perhaps in a separate department set aside for the exclusive use of the chief engineer for experimental work. When they are completed, extensive road tests will be conducted and the designs altered until the performance of the cars is satisfactory. Then, when the chief engineer is satisfied, the drawings are systematised and the materials ordered. From this point onwards the system followed is practically identical with that used in any really good European works, though the organisation is generally brought to a much higher perfection. As soon as a design has thus been accepted by the factory, the chief engineer proceeds to consider the next model, the drawing office superintendent, who acts under the chief engineer's orders, being responsible for the supply of blue prints, the numbering system, and so on. It will be noticed at once that this scheme assumes the chief engineer to be a thoroughly practical motorist, and does not expect him to have a very intimate knowledge of machine shop methods. Consequently, the American car, although it may be rough, is usually fairly convenient and fairly comfortable. There is not much risk of really unpractical designs being placed on the market because the "model" cars, as they are called, are tested so thoroughly by the chief engineer and his staff that faults are likely to be found before the actual manufacturing is commenced.

It appears that there is generally ample time between deciding on a new model and the commencement of its manufacture. The cars which will appear in the January automobile shows in America were mostly being tested on the road last June. Engineering staffs are now at work getting out new models to bear the date of 1913, and actually to be on sale about six months hence. Of course, this means that the American car is dated practically six months ahead of our own. The majority of new cars which will be found at the Olympia Show are probably only just on the road, while a good many of them will never have been tried at all, and these are 1912 models. By the time European makers are supplying customers with 1912 cars the American 1913 models will be undergoing their tests. Of course, this helps to make American cars seem old-fashioned; the cars which are now being designed are very different in many cases to those in the markets.

This is digressing somewhat from the point so, returning to the actual manufacture, something may be said concerning the buildings themselves. Nearly all modern works in America are constructed on the same plan. There is an office building of two or three stories containing the whole of the clerical staff, the drawing office, the staff restaurant (an almost invariable portion of American works), and perhaps a trade showroom. This building occupies the road frontage, and most frequently stands well back with a good sweep of lawn in front of it. It is rectangular in shape, the long side, of course, being towards the road, and the works themselves are preferably somewhat similar in general plan, but with the narrow ends towards the road. The works buildings may have two, three, or four stories, and a common proportion would be three or four works blocks to one office block, the works blocks often extending backwards from the office building to a depth of a quarter of a mile or more. Reinforced concrete is far the commonest building material, and there is a lavish supply of window. The walls indeed are often almost entirely glass, only sufficient concrete being left in pillar form adequately to support the weight of the upper floors. Between the buildings there will be sufficient spaces to give ample light, while the power house, the testing sheds, and the foundry are almost always quite separate, and may be some distance away. One finds very few cramped factories, land appearing to be cheap on the outskirts of the towns, and, with plenty of space, all kinds of systems can be worked to the best advantage. One thing which strikes a British visitor most curiously is that every shop is floored with wood, narrow tongued and grooved boards being used, which are forced together by a sufficient pressure to make them quite free from cracks. Wood also is used much more for internal fittings than it is with us, but the fire protections are extraordinary, especially considering that the far less rigorous laws of the United States do not impose very heavy responsibility on employers of labour.

The comfort of workmen is cared for much more elaborately than here. The wood floors are, of course, a concession to the labourer. Lavatories, with copious supplies of hot as well as cold water, and even baths, are attached to nearly every works, while metal wardrobes or lockers are usually supplied to every man, and it is quite a usual thing for mechanics to keep their working clothes in the factory and never appear outside it otherwise than perfectly clean, neat and tidy. Except in the few instances where factories are in the heart of a town, there is a catering department with a big dining room, where really good meals are served at cost price, but the capital involved in such undertakings is partially compensated for by the shorter interruptions in the working day, half-an-hour from 12 o'clock often being the only break in a ten-hour day. Neither is the

Saturday half holiday by any means a regular thing in America. Some of the works close on Saturdays during the slack season of the year, and some on alternate Saturdays all the year round. Just one or two follow the English custom, and it seems likely that their numbers will increase.

It is not proposed to say much concerning the actual equipment of machine shops, the subject being far too vast for treatment in article form, nor could sufficient material for an authoritative statement be gathered without giving a great deal more time to the collection than could be made in a reasonable visit, but, speaking generally, there is very little that is superior to the equipment of a first-class British works. The only really striking difference between an American and a British machine shop is the entire absence of old tools in the former. Every sort of machine employed is the latest and most efficient of its kind, and old tools are scrapped quite ruthlessly. This is one of the ways in which the big production acts to the advantage of the American. Here, often the manufacturer hesitates to buy a special tool for—say milling the cylinder faces—because such tool would finish sufficient cylinders in a few weeks to last him for a year. In an American factory one special machine often does not go very far. Often one sees in the gear cutting department, for instance, banks of twenty and thirty machines of one type, and if a new pattern saves a few cents on each piece turned out, it pays for itself very rapidly, because it can be kept busy all the time. The Pierce Arrow Company, for example, have a gang miller with a long traversing bed on which a row of cylinder castings can be mounted as soon as the bottom flanges have been faced. The cutters then finish every other facing simultaneously, the top, the inlet and exhaust pipe flanges, and the water pipe flanges, needing only boring to complete the cylinder.

In another works the rear axle used is slightly arched, and consists of flanged tubes bolted up to a central casing. To make a simple turning job of the sleeves, the casing is faced on a special enormous machine which carries two boring bars at the requisite angle, and operates on both sides of the case at once. The material being cast steel, the rigidity necessary will make the massiveness of the machine and its cost quite obvious.

Another interesting tool, too, possessing a capacity in excess of the requirements of most European works, I saw in the Hupmobile factory, this being an automatic machine for grinding in all the valves at once—and making a thoroughly good job of them. The most elaborate special tool of all, however, was in use in the Cadillac factory in the erecting shop, and was designed with the idea of minimising the amount of labour in lining up engines and gearboxes to their frames. Rivetted to the chassis are some seven or eight brackets or feet, on which the engine and gearbox arms rest and to which they are bolted. Owing to the inaccuracy of pressed steel work, it is naturally not easy so to assemble a frame that all these brackets shall lie in a single plane. Ordinarily, the crankcase or gearbox would first be attached and the other member got into position by use of packing pieces. The Cadillac arrangement, however, is to have a travelling crane running the whole length of the erecting shop and carrying a rigid frame which bears an electric motor driving a number of spindles, corresponding in position to the crankcase and gearbox brackets. Milling cutters are mounted on these spindles, and all are driven together, so that when a chassis frame is mounted on its springs and axles the tool can be run along the traveller and lowered down. As it descends, the milling cutters come into operation on the brackets, and the feed is continued until all the brackets are faced up. Naturally, the cutters themselves being fixed in a rigid plane, they attack the highest bracket first and the lowest last, so as soon as every bracket shows evidence of having been cut, it is known that the whole set are at the same level, and that the gearbox and engine castings can be dropped into place, with the certainty that no vertical lining up will be required.

Assembling Producers.

A very large class of manufacturers who assemble, instead of actually making their cars, are worth a little special consideration, because the results are in some ways surprising. Over here the manufacturer who buys finished parts and puts them together has generally been regarded somewhat contemptuously, and it is usual to assume that cars made in this way cannot be really good. American car assembling, however, is a different scheme of things entirely. First of all, one usually finds that the system already described of getting out "model" cars prevails in works of every class. The chief engineer and his department design the cars, and these cars are actually made on the premises in a small machine shop, while they are tested exactly as the products of a manufacturing company. When the design has been approved, and after getting out drawings, instead of passing them to the machine shop superintendent, various general engineering firms (of which there are an enormous number throughout the United States) will be asked to quote for each individual part, a very rigid material specification being the rule. Then, when a tender has been accepted, the purchasing concern sends a kind of clerk of the works to the supplier's plant, his business being to act as an inspector of the work done there. In some cases even quite a large staff of inspectors are kept going dotted about in various works.

One finds, of course, that there are large concerns specialising on the production of automobile parts, but the various axle or gear companies, several of whose names will be familiar to the majority of my readers, do not only get out standard patterns of their own, but are ready to make to their customers' designs. Often this accounts for an apparent similarity between the axles on two cars, but the details will differ. The Metal Products Company, of Detroit, supply axles to both the Hudson and the Chalmers Company for instance. This concern possess dies from which they can prepare a rather neat pressed steel casing, but the dies are costly and, therefore, one finds that both the Hudson and Chalmers have the same axle case. Coming down to details, however, there are quite a number of points somewhat dissimiliar so, although the two axles are very much alike outwardly, they are really separate designs, the convenience of using the standard stamping for the case really counting for no more and no less than the convenience of buying standard axle case stamping from Messrs. Rubery Owen (for example) does here. At last Olympia Show several cars had axles with the Rubery Owen case, but only the initiated would perhaps have known it. In America when they say that such-and-such a car has a "Sheldon" or a "Metal Products" axle, it does not mean much more than it would to say that the Thornycroft and the Armstrong, Whitworth here, had a Rubery Owen axle. It is, of course, possible to buy completely finished engines, axles and gearboxes, and a few firms do this, putting them together in much the same way as some of the mushroom companies in England and France put together the products of some of the component makers round about 1905, but these firms do not occupy an important position in the American industry.

Of course a firm who possess their own machine shops take a higher standing in America than do an assembling company, but there is no stigma in buying many or even all finished parts. Commercially it is that the advantage of the assembling scheme is to be found, and it is financial foresight that has led to the evolution of so many "erecting shop factories." Take, for instance, the case of a firm who believe they are able to sell five thousand cars during 1912. Certain men already in the industry get hold of a rough design which they believe will sell well. Between them they determine to float a new company, and to put a new car on the market. Now, in a matter of a few weeks probably enough money can be found to build a works, in six months a "factory" will be ready, and, if the parts are to be bought out, assembling can be begun at once. On the other hand, if it is wished to make every part, or most of the parts, the equipment of the factory will take a long time, and the profit earning possibilities of the company delayed. Not only this too, for the factory would need to be larger to accommodate the tools, and both the increased building itself and the increased staff necessary to work the business would load the new company with a very heavy capital expenditure. Thus the assembling system enables a new design to be put on the market on a large scale in the minimum of time, and with a minimum of capital outlay.

It is stated that firms such as the Hudson Company (who turn out five thousand or so chassis a year) have been able actually to pay for the cost of their buildings, out of profits and within twelve months of entering into possession. This, of course, would be impossible by any other system, but it means that the shareholders of some of the youngest American firms are (when they have large holdings) not only making good incomes, but are actually amassing fortunes of no mean proportions.

Here the absence of a sufficiency of general "contracting" mechanical engineering firms and the different class of car demanded by the public, make the position quite different, but it does not follow by any means that the policy is not a thoroughly good one from the point of view of its upholders.

(To be continued.)

STEEL.

By Walter Rosenhain, B.A., D.Sc.

(Continued from page 459.)

HAVING recognised, by the aid of the microscope, the various constituents of slowly-cooled carbon steels—ferrite, cementite, and pearlite, our next step is to enquire as to the nature and mode of origin of these various bodies. To follow up this enquiry it is necessary to go beyond the microscope and to make careful observations on the process of cooling which steel undergoes in passing from the liquid, molten state at very high temperatures down to the final condition which we have seen under the microscope. In this way each step in the genesis of the micro-constituents can be followed and their mode of formation and the various changes to which they are subject can be understood. Unfortunately, in the case of steel these changes are exceptionally complex, and it is therefore necessary to begin by considering a simpler case.

First as to the observations themselves and the means by which they are obtained. All that is required is some method of determining at short intervals of time the exact temperature of the sample of steel under observation—this sample may be a small ingot cooling in a crucible or it may be a small cylinder of the metal being cooled or heated in a tube-furnace. In either case the most convenient means of taking the temperatures is a thermo-electric couple, inserted into the mass of steel and suitably protected from injury; this couple is connected to some form of electrical apparatus which gives exact readings of the temperature of the couple at any instant. The cooling—or heating—process is watched by noting the time occupied by successive equal changes of temperature. If the rate of cooling remains uniform or only changes gradually, these time intervals will remain the same or will only change gradually, but if for any reason the rate of cooling is either retarded or accelerated, then these observed time-intervals will suddenly became markedly longer or shorter. The best means of showing such changes is to plot the observations in the form of a curve, the time intervals being plotted horizontally, while the actual temperature at which each interval was observed is plotted vertically.

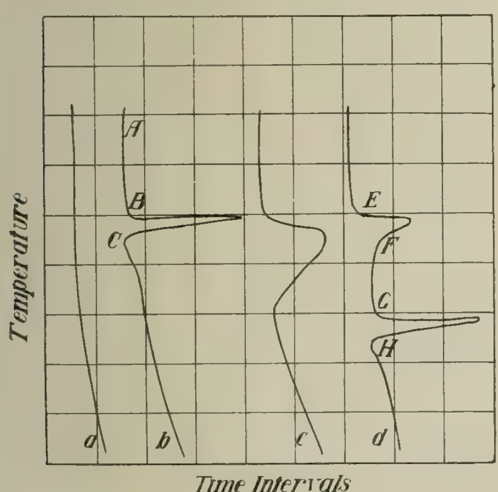


Fig. I.

The resulting curve is a "cooling curve" —or in the case of heating, a "heating curve" of the sample of steel (or other metal) which has been under observation.

If we take the cooling-curve of a perfectly inert body over a range of temperature which does not approach its melting-point, we obtain a curve like (a), Fig. I., which is a smooth curve not very far removed from a vertical straight line; the curve as plotted might represent the cooling of a piece of platinum from 1,600° C. to 200° C. If, now, we take a sample of copper and follow its cooling

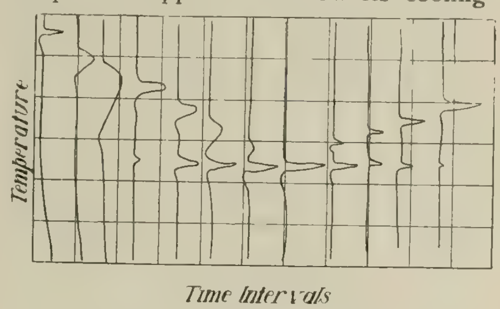


Fig. II.

over the same range of temperature, we obtain a totally different form of curve, like that shown at (b), Fig. I. The topmost portion of this curve, from A to B, is very similar to the corresponding part of curve (a)—here we have molten copper cooling steadily without undergoing any special change. At the point B, however, the curve suddenly sweeps out to the right—thus indicating that the time-intervals occupied by the metal in cooling a given amount have suddenly become very much longer—i.e., that the cooling of the metal has been very much retarded, or even entirely stopped, for a time. Now the metal is losing heat at much the same rate all the time, and if its fall of temperature has been stopped or retarded this can only be due to the fact that some heat is being supplied to the metal sufficient more or less completely to balance the outgoing heat for a time. In the present case this supply of heat is derived from the "latent heat of fusion" of the copper; the copper is undergoing solidification at this temperature and the energy stored in the molecules of molten copper is being given out in the form of heat. Hence the "arrest point" or point of arrested or retarded cooling on the cooling curve, an arrest which lasts until the whole of the copper has solidified. The peak of the cooling curve at B represents the duration of this arrest; when the copper has completely solidified, however, the supply of heat ceases and cooling is at once resumed; it must, however, be noticed that, while the temperature of the copper itself has remained stationary during the freezing process, the crucible and furnace containing the copper have continued to cool, so that the freshly-solidified copper finds itself in surroundings which are considerably colder than itself. For a time, therefore, the copper cools more rapidly than its "normal" rate, so that the curve sweeps slightly to the left after returning from the peak of the arrest-point. Subsequently. as the temperature of the copper gradually catches up to that of the furnace, the normal rate of cooling is gradually resumed, and the lower part of the curve is very similar to curve (a).

The curve (b) is typical of the cooling-curve of any pure metal taken from a temperature above its melting-point, but when we deal with alloys, very different types of cooling-curves are met with, although it is possible to group alloys together into a few large classes having similar types of cooling-curves. This classification depends upon the manner in which the various constituents of the alloys behave towards one another as the alloy cools and solidifies. In the liquid, molten condition we may safely regard alloys as being solutions of metals in one another, closely resembling the ordinary solutions of salts in water with which we are familiar—the two molten metals are intimately mixed and may be regarded as being uniformly diffused through one another. Now the question arises, How does this intimately-mixed liquid behave when solidification takes place? There is a very large class of alloys—to which some of the iron-carbon alloys belong—in which the process of solidification produces no separation of the two constituents; the mass freezes and forms a solid crystalline aggregate in which the two constituents are still uniformly distributed—at all events, if the solidification is allowed to take place slowly. The resulting solid crystals consist of the two metals still as intimately intermixed, as completely diffused through one another as did the liquid solution—and consequently we term these solid bodies "solid solutions." Now the cooling-curve of

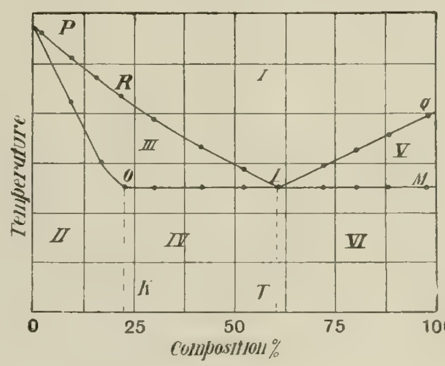


Fig. III.

an alloy of this class is very similar to that of a pure metal—there is again a single peak representing the process of solidification; the only difference is that, while with very pure metals the peak on the cooling-curve is very sharp and extends over only a very short range of temperature, in the case of solid solutions the peak is often very blunt and the freezing-process extends over a large range of temperature. This is shown in curve (c), Fig. I., which illustrates the fact that in such alloys the commencement and the end of solidification may be widely separated in temperature. The freezing process of a steel containing about 1 per cent. of carbon is of this type.

We have now to consider another class of alloys in which the state of mutual solution which exists in the liquid state

is not preserved, or only to a limited extent, after solidification. In these alloys solidification begins by the separation of that constituent which is present in excess, leaving behind a liquid which steadily increases in concentration of the second metal as the first is removed by solidification. This process continues until a certain limiting saturation is reached, and then the remaining liquid solidifies completely, forming a mass of juxta-posed crystals of the two constituent metals. The freezing-process of such alloys is both prolonged over a wide range of temperature and marked by two very definite stages—viz., that of the initial freezing of the predominant constituent and the final freezing of the residual liquid or "eutectic," as it is called. The cooling-curve of such an alloy is shown in Fig. (d) I. At E we have the peak due to the commencement of solidification of the metal present in excess (say, metal P). This peak ends at F, but the curve remains displaced to the right since more and more of the metal P continues to fall out of solution as the temperature drops; finally, at the point G the remaining liquid has reached both its limiting saturation and the lowest temperature at which it can remain liquid, and the whole now freezes, as indicated by the peak on the curve, forming an aggregate of crystals of P and the second metal (Q). It is important to notice, however, that for the whole series of alloys of P and Q, while the temperature of the point E varies with the composition, the temperature of the point G, the solidification of the saturated liquid, remains constant, while the composition of this saturated liquid is always the same for any given pair of metals. If the whole of the alloy has the composition of this saturated liquid, then the whole will remain liquid down to the temperature of the point G and the whole will then freeze on reaching that temperature. Such an alloy is known as the "eutectic" (most fusible) alloy of the series.

The manner in which the cooling-curve varies from one alloy to another of the same series is best shown by plotting the whole series of cooling-curves on a single diagram. If we plot such a series of curves next to one another, as in Fig. II., we may convert the series into a single diagram by supposing each cooling-curve to be erected vertically on a point corresponding to the composition of the alloy from which it was taken, and in doing this we may simplify matters by representing each "peak" by a simple dot or cross. This has been done in Fig. III., and it is then seen that these dots can be joined by a system of simple lines or curves, and a little consideration will show that these lines have a simple and important meaning. We have first the line P L Q joining all the points of initial freezing, so that above the line P L Q the alloys are entirely liquid, while below that line they are at least partly solid. From P to R this line represents the commencement of solidification in alloys which form solid solutions, and the end of the freezing process in these alloys is indicated by the points along P O—but it must be pointed out at once that these cannot be accurately found from cooling-curve observations, but must be determined in another way. From R to L we have the commencement of freezing of alloys which, along the line O L complete their solidification by the formation of eutectic, and this eutectic extends to L M, but while the alloys to the left of L begin to solidify by depositing crystals of P, or rather of solid P, containing as much Q as it can retain in solid solution, along Q L the alloys first deposit crystals of Q, the eutectic freezing along O M remaining the same in composition and nature throughout. It will be seen, now, that below the line P O M the alloys are all completely solid.

If to the lines of Fig. III. already indicated we add the two dotted lines O K and L T the figure becomes a diagram dividing up the whole system of alloys at all temperatures into six definite regions representing six different conditions of the alloys. In region I. they are entirely liquid. In region II. they are entirely solid and consist of one kind of crystals only—they show a micro-structure typical of a pure metal or of a homogeneous solid solution, such as that seen in Fig. IV., Plate II.—the crystals are either the pure metal P, or P containing Q in solid solution. In region III. the alloys consist of crystals of P embedded in the solidified residual liquid or eutectic and show micro-structures like those of Fig. V. and VI., Plate II., where the white crystals of P, and the dark duplex structure of the eutectic are clearly seen; the amount of eutectic increases as we move to the right in this region, until at the point L the whole of the metal consists of eutectic, having a structure like that seen in Fig. VII., Plate II. In region V. we have again a mixture of solid crystals—this time of Q—and liquid, while in region VI. we have crystals of Q embedded in the eutectic, as shown in Figs. IX. and X, Plate II. The illustrations of micro-structures given here have actually been taken from such alloys as those of lead and tin and of aluminium and zinc, but they are typical of any system of alloys, although special factors have to be taken into consideration in individual cases.

The diagram, whose formation and meaning we have studied at some length, is generally known as the "Equilibrium Diagram" of a system of alloys, since it describes all the conditions in which the alloys can exist in equilibrium at various temperatures and compositions. We have already seen the direct connection between this diagram and the micro-structure of the alloys, so that we must look to the equilibrium diagram of the iron-carbon alloys for the explanation of the micro-structures of steel.

The equilibrium diagram of the iron-carbon alloys, according to the best available data, is given in Fig. XI. It will be seen at once that the upper portion of this diagram, viz., the lines A B, B C, A D, D B, B E, closely resemble the ideal typical case given in Fig. III., and the interpretation of the lines and areas is exactly the same. There is, however, one modification to be made before the interpretation can be correctly given. In the ideal case of the alloys of P and Q we were dealing with the alloys of two metals, and our series of alloys extended from a composition of 100 per cent. of P to 100 per cent. of Q. In the iron-carbon system this is not possible—alloys with more than about 6 per cent. or carbon cannot be prepared—so that our series here really relates only to one end of the binary system. On the other hand, so far as steel is concerned, it is not strictly correct to speak of alloys of iron and carbon since the carbon does not enter into these alloys in the free state but only in the form of the iron-carbon compound Fe_3C, which has received the name of "Cementite." Our alloys are therefore really iron-cementite alloys, and since Cementite only contains 6.7 per cent. of carbon this accounts for the fact that a typical binary diagram is obtained for a series of alloys extending over a range of only 6 per cent. of composition. With this explanation we may now approach the diagram of the iron-carbon alloys, Fig. XI.

Along the line AB we have the commencement of solidification by the deposition of crystals of iron, or rather of iron containing as much cementite in solid solution as they are capable of carrying. Along BC we have solidification commencing by the formation of cementite crystals, while along DBE there is the freezing of the eutectic of iron, saturated with cementite and *of* cementite. Along AD we have the completion of the solidification of the solid solution crystals.

Up to this point the diagram is perfectly similar to Fig. III., and only two points of special importance need be mentioned. The first is that the lines AB, BC, give the temperatures which must be attained for complete fusion of steel of the corresponding grade. The second point is that the lines ADB indicate the temperatures at which steel of each particular grade commences to undergo fusion. Now it is generally admitted that heating steel to a point at which fusion begins will completely spoil the steel, this incipient fusion being in fact equivalent to "burning" the steel. The exact position of the line AD is therefore of particular practical importance, and the position of this line has been determined by special methods. It has already been indicated that from the cooling-curves it is not really possible to determine the temperature at which the solidification of a solid solution is quite completed—the end of a heat-evolution is never well-marked on a cooling-curve. The method which has been used to supplement the cooling-curves in this respect has been to heat small specimens of steel of various carbon-content to known high temperatures, and then to cool them very suddenly by quenching them in water. When such a quenched specimen is afterwards examined with the microscope, it is possible to tell very definitely whether at the moment of quenching any portion of the metal was actually melted. In this way it has been possible to see exactly at what temperature incipient fusion had occurred, and so to fix exactly the position of the line AD. It is certainly safe to say that in no circumstances must the working temperature of steel be allowed to exceed that of the line AD as given in Fig. XI.

If the equilibrium diagram of the iron-carbon system contained only the lines which have so far been discussed, we should find in our steels a series of micro-structures exactly similar to those of Figs. IV. to X., and indeed in white cast-iron we can actually see the structures to be anticipated from these considerations. But in the case of iron-carbon alloys containing up to about 2 per cent. of carbon we should in that case find nothing but a perfectly homogeneous structure of a simple solid solution, viz., the solid solution which completes its

solidification along the line AD. In practice this simple structure can only be found if we examine the steel at high temperatures (lying above the line FHID), or treat it in such a way as to suppress the changes which those lines indicate. A photo-micrograph of the structure of steel at such a high temperature is shown in Fig. XII., Plate II., obtained by heating a specimen of steel in vacuo to a temperature of 1,100° C., and etching it with hydrochloric acid gas. This photograph merely serves to show that at these high temperatures the steel really possesses the simple structure of a homogeneous solid solution. We have now to consider what happens to this solid solution when it cools at the ordinary rate.

To understand the process which takes place we must recur to the analogy which has already been pointed out between a solid and a liquid solution. We have seen how a liquid solution of two metals deposits crystals first of one of its constituents and then of a eutectic mixture of both, as described at length in connection with the ideal equilibrium diagram of the alloys of "P and Q." Now we find that a solid solution can undergo decomposition in precisely the same manner, and if we regard them from this point of view, the lower lines of the iron-carbon diagram (FHID, JIK) will be understood at once. Above the lines FI, ID, we have a homogeneous solid solution corresponding to the homogeneous liquid solution above the lines PLQ of Fig. III. Along FI this solid solution begins to deposit crystals of iron, and along the line DI crystals of cementite, while in each case the residual solid solution—exactly corresponding to the residual liquid in Fig. III.,—becomes more nearly saturated.

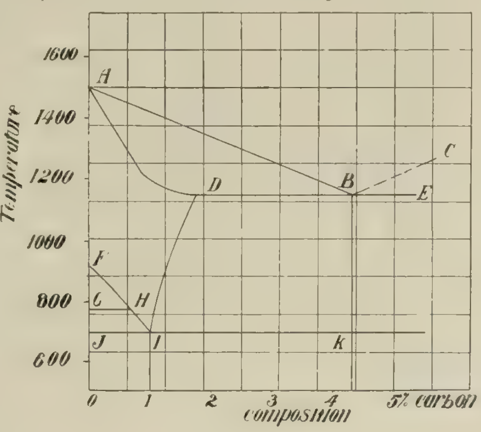


Fig. XI.

Finally, the composition and temperature of saturation of the residual solid solution are reached along the line JIK, and there the residual solid solution deposits juxtaposed crystals of iron and cementite forming a body strictly analogous to the eutectic of the alloys of "P and Q"—but for distinction this body has been termed a "eutectoid" instead of a "eutectic." The resulting micro-structures can now be readily understood. In the region below the line IJ we shall expect to find crystals of iron embedded in the eutectoid, and if we refer back to Figs. X., XI., and XII., Plate I., we shall recognise the iron in the "ferrite" of those photographs and the eutectoid in the duplex, laminated pearlite, whose structure closely resembles that of the eutectic illustrated in Fig. VII. Then to the right of I we shall look for crystals of cementite embedded in pearlite, and these we find in Figs. XVII. and XVIII. of Plate I. of the previous article. We have thus a complete explanation of the micro-structures of the slowly-cooled steels as a direct consequence of the lower lines of the equilibrium diagram of the iron-carbon system interpreted by the close analogy which exists between liquid and solid solutions. In our next article we shall utilise the equilibrium diagram in order to obtain an insight into the more complex structures which are obtained when steel is quenched and tempered, and incidentally we shall also have to consider the nature and meaning of the line GH, which has not so far been considered, although—like the rest of these lower lines—it represents definite observed retardations on the cooling-curves of these carbon steels.

(To be continued).

ROTARY VALVE ENGINES.

Two American designs which are in actual use in that country, and possess certain qualities combined with reasonable durability.

In these columns belief has several times been expressed that a rotary valve engine, of some as yet uninvented type, is likely to triumph ultimately over both poppet and slide valves. Speaking broadly, the rotary valve seems to be the most mechanical of the three, and would appear also to lend itself to the simplification of the internal combustion engine. Whether either of the designs described below have any lasting merit time alone can show, but it may at once be said that the experimental engines which are now running in actual use do not seem to have any special drawbacks, notwithstanding their apparent crudity of design. The Mead engine, which is perhaps the least promising on paper, has been run on the road for some twenty thousand miles in a car of moderate weight, and has earned the approbation of Mr. Henry Souther, the President of the Society of Automobile Engineers of America, who has had the engine under his personal observation during most of its life.

The first engine had a bore of four

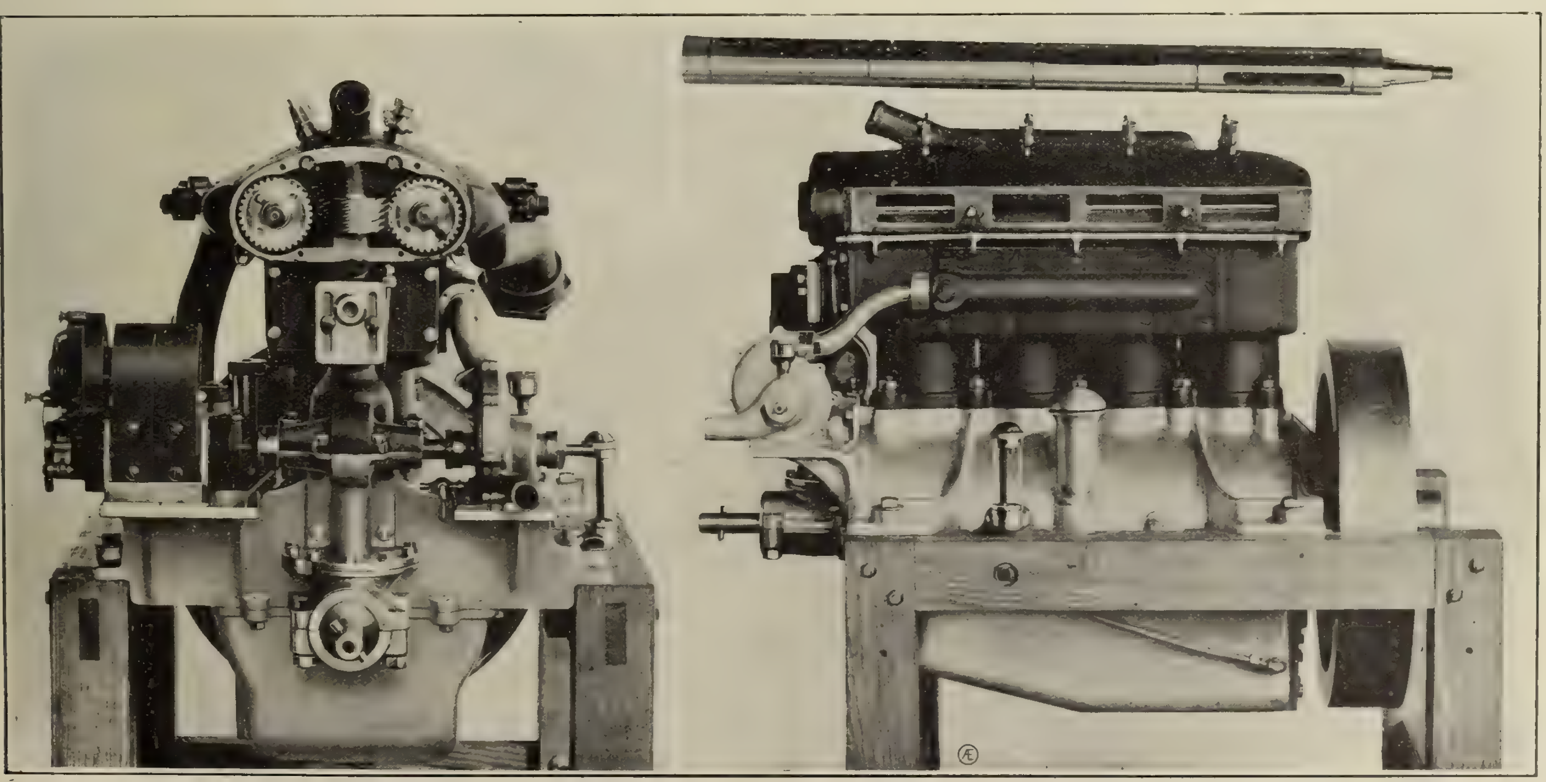


Fig. 1. The Mead Engine, showing the method of driving the valves; a view of one of the latter is inset.

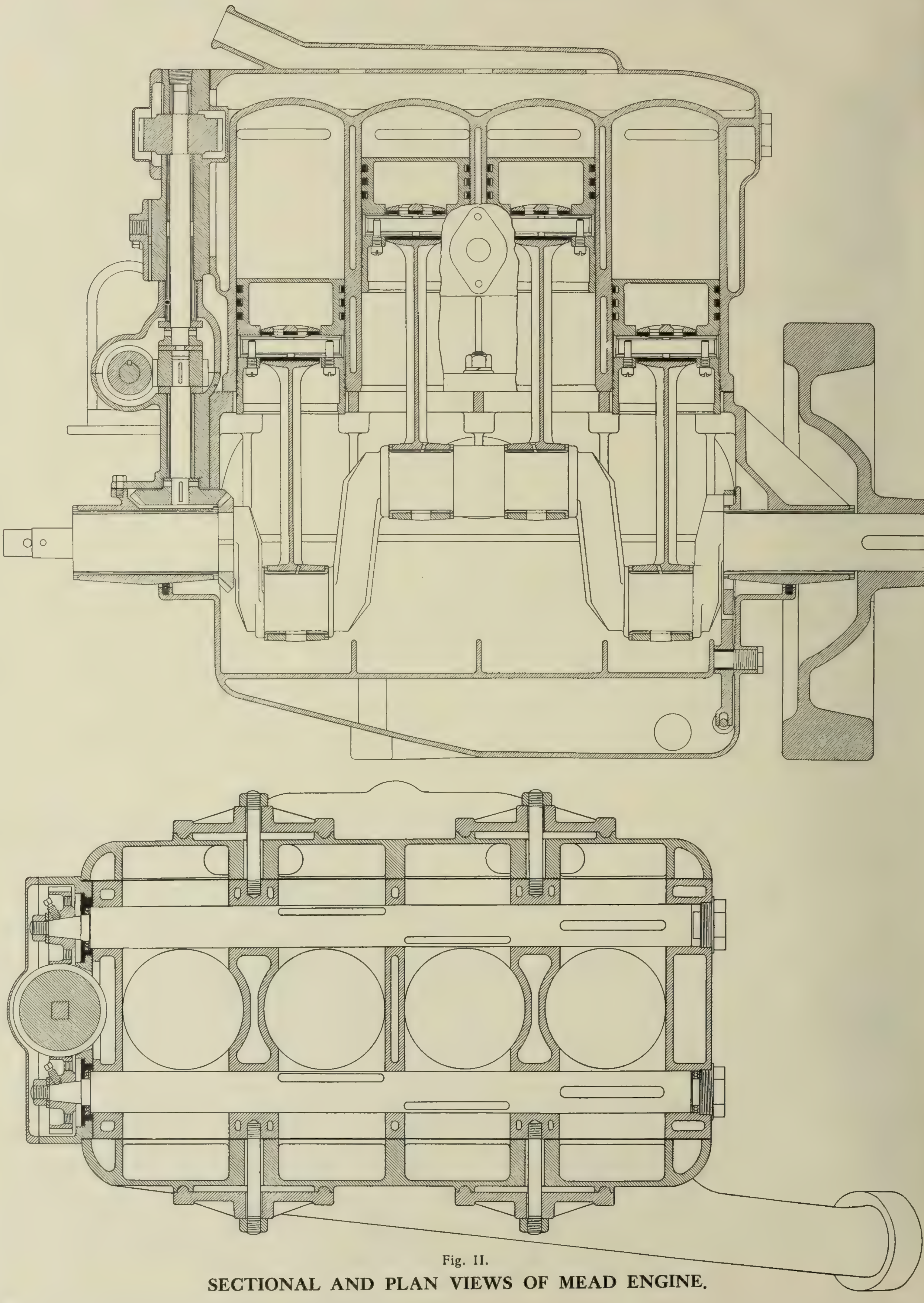


Fig. II.

SECTIONAL AND PLAN VIEWS OF MEAD ENGINE.

inches, and a stroke of very little more than the bore, the valve diameter being $1\frac{3}{8}$ ins. The cylinders are all cast together, and the valve chambers are cored, great care being taken to ensure their exact centring when commencing to bore them from the front end, but no special precaution being observed to maintain the same accuracy at the other end remote from the driving gears. Care is, of course, taken to ensure the correct slot-

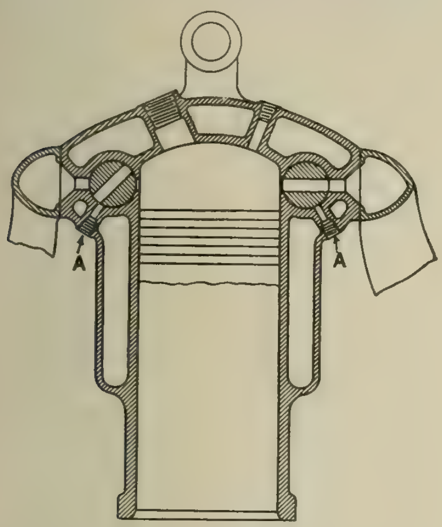


Fig. III. Diagramatic Section of the Mead Engine. A, A are lubricating nozzles.

ting of the ports in the cylinder walls, and the valves are cut in a milling machine, there being no difficulty in spacing the openings correctly. Cast iron is used for these valves, but the driving ends which carry the spiral gears are of steel, the two parts being screwed together. Fig. I. shows a side view of the engine with the exhaust pipe removed, and also an end view, which makes clear the method of driving. Separately in Fig. I there is shown one of the valves removed from an engine after use, but it is on a rather larger scale.

Naturally, the first thought on seeing this engine is that the compression would probably be very poor if the clearances were sufficient to prevent sticking, but this does not seem to be the fact. Firstly the clearance is very small, the valves being ground to only one and a half thousandths of an inch less in diameter than the bore, and also the pressure in the cylinders tends to keep the valves tight against the outer parts of their seatings. Next, it is assured that the valves do not score, because any roughnesses which they may originally possess very soon become filled up with carbon deposit, and in this connection an interesting experiment was tried. A valve was first finished as usual and then maltreated in different ways. One part was scratched deeply with a rough file, another rough ground longitudinally, another had a small flat made on it, and another was chipped lightly close to the edge of a port. After a few days of running all the abrasions had disappeared, leaving a highly polished surface of hard carbon.

To obtain different timing it is, of course, necessary to vary the dimensions of the ports, but with valve slots $\frac{3}{8}$ in. wide the inlet opens at $4\frac{1}{2}^{\circ}$ past the dead centre, and closes at 38° past the bottom of the stroke, while the exhaust opens 73° before the bottom, and closes exactly on the top dead centre. Experiments are still being made in the endeavour to find the most suitable setting, but in any case the opening is fairly rapid and the accuracy is maintained, as the edges of the slots do not appear to burn at all. With the first engine lubrication troubles were anticipated, and somewhat elaborate arrangements were made to feed oil to the whole length of each valve. These have now been simplified, and consist of a lead, from either a gravity or low pressure supply, to each of the five inter-opening bearing surfaces on both valves. Owing to the fact that the valves are surrounded by water they keep reasonably cool, and to guard against overheating, fins are cast to direct the cold water down the valve passages before it enters the cylinder jackets proper. It is conceivable that some trouble might be encountered if it was desired to use very high compression, but the rotary construction has many advantages over all other types of valve for high speeds of revolution. Fig. II. shows the experimental engine in greater detail, and Fig. III. is a diagramatic section in which the valve lubricating leads are shown.

The other engine is known as the Reynolds, and this has been on the market for about a year in a form suitable for motor boat work. Fig. IV. is a section showing the system, and at the same time disclosing the driving gears for the valves. After the cylinders are bored in the usual way they are faced at the top of the combustion space and finished to a very smooth surface with a hand operated tool made with a scraping cutter, and intended to do no more than to remove roughness. For the valves, phosphor bronze is used, this having been found to be more durable than other metals. The port openings are, of course, fan-shaped, the valves each bearing a single orifice which registers with the inlet and exhaust ports, slotted by jig in the correct positions in the cylinder heads. On the spindle of each valve a single helical gear is fixed by means of a Woodruff key and nuts, a few thousandths of vertical up and down play being allowed. No provision is made either to maintain the valve in contact with the cylinder head during the inlet stroke, or to relieve the pressure during the expansion stroke, but a small quantity of oil is fed by a pump to the centre of each valve spindle, whence it passes down as shown in Fig. IV.

At first some trouble was experienced through the valves warping, but it was found that this could be overcome by annealing after rough-turning and before taking the finishing cut. Each valve is ground in with emery or carborundum, and an examination of an engine which has been run for a considerable time showed that the scoring of the working surfaces was not very pronounced. Thus it seems that the actual valve mechanism is a practical possibility, and the chief disadvantage therefore is the elaborate driving gearing. This is now so proportioned that there is absolutely no backlash when first fitting up the engine, and the writer has been unable to examine the gears of a motor that had done much work. Certainly a new engine can run quite quietly and with very little vibration.

In the particular type of which Fig. IV. is the general arrangement, there are a number of peculiarities which do not affect the principle, and will, therefore, not be commented upon here, but a better designed example is now in course of construction by one of the larger firms in Detroit, and exhaustive experiments will be made with it in actual service in a car on the road. If really successful the gear would lend itself to much simplification, by the substitution of a chain-driven top shaft and skew gears to each valve for the spur wheels now employed.

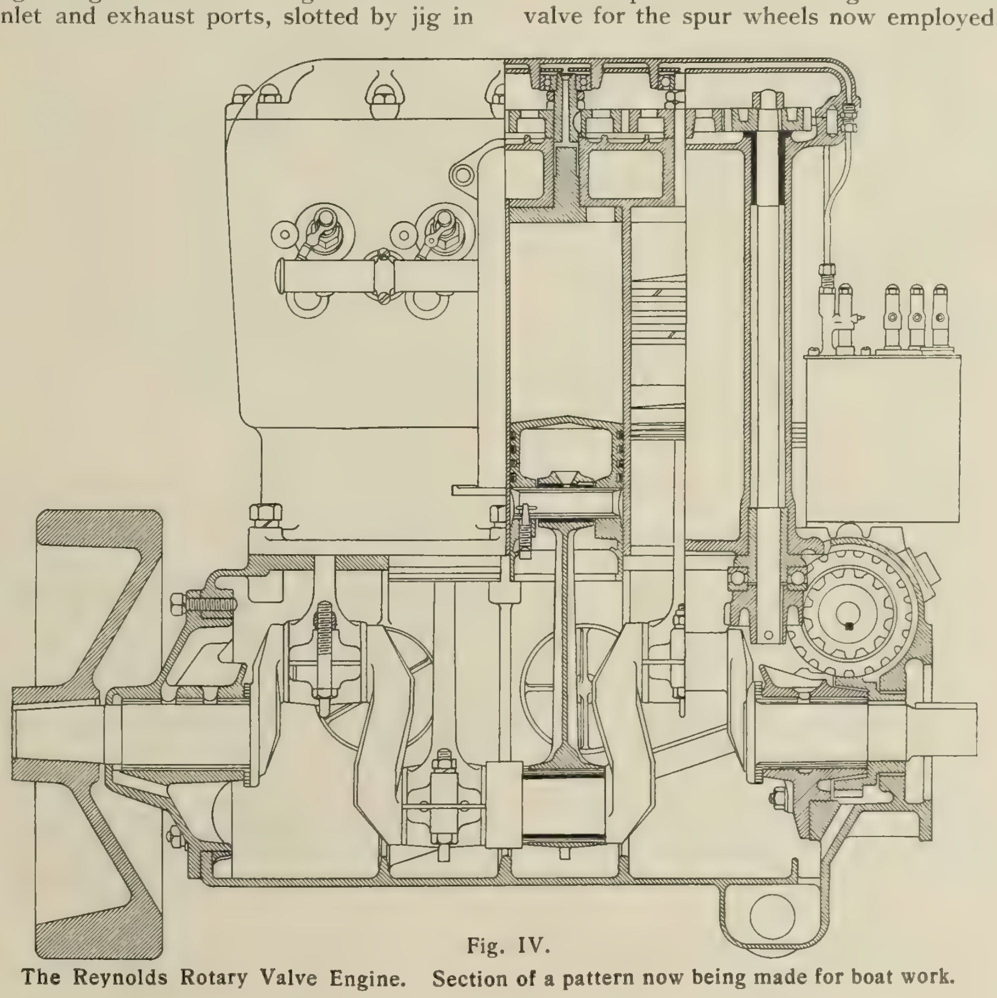


Fig. IV.

The Reynolds Rotary Valve Engine. Section of a pattern now being made for boat work.

MODERN MACHINE METHODS.

An Account of the Sunbeam Works Practice.

By Robert W. A. Brewer, A.M.I.C.E., M.I.M.E., M.I.A.E., F.S.E.

THE manufacture of the motor car complete with its various parts is now, after some years of experiment and financial difficulty, a matter of serious and sound business. It has become so in England owing to the adoption of a few standard sizes, following in this way American practice, the carrying out of sound designs and the most modern and up-to-date workshop practice.

As an instance of the enormous development which will result firstly by the selection of a good staff, secondly by giving them scope for their talents, thirdly by the provision of suitable machinery, and fourthly by the supply of adequate capital, the writer is able to place the following useful and interesting information before the readers of The Automobile Engineer.

In choosing an example of a progressive firm the Sunbeam Motor Car Co., Ltd., have been taken partly on account of the fact that during the past two years a sum of no less than £20,000 has been expended upon the works equipment, and a further sum of £20,000 is now being spent in the erection of a large new shop and in the addition of machine tools. During this time the profits earned by the works have increased so largely that it is evident that the manufacturing methods adopted must be such that the cost of production is an absolute minimum. This cost is not kept down by any sort of scamping of the work, as every detail is constructed most carefully, the workmanship throughout being of a very high order.

By the courtesy of the managing director, Mr. Cureton, and of the engineer and designer, Mr. Coatalen, the writer has made an extended visit to the works and was given exceptional facilities for examining, photographing, sketching, and gauging, the work in the factory. It was then possible to discover how such high-class work could be carried out at a cost of production such as to make a commercial success of motor car manufacture. It is a very difficult question for a firm of motor car manufacturers to decide which size of car is the most likely to be popular, as well as profitable, when they embark on their new season's model in an extensive manner. The Sunbeam directors decided wisely when they came to the conclusion that a high-class 80 mm. bore four-cylinder car was the most likely to satisfy the objects in view. The 1910-11 80 mm. bore engine has been manufactured with very great success, and this dimension of cylinder diameter is now one of their principal standard models.

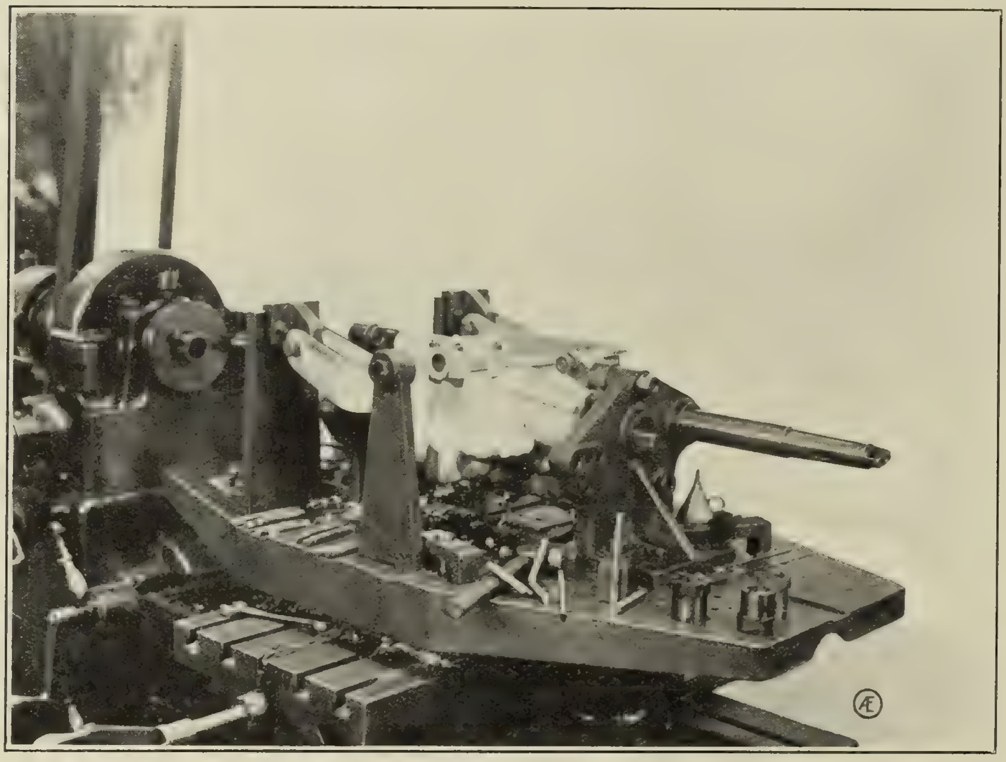


Fig. I.

Universal jig arranged for gearbox. New and old types of tools also shown.

The principal objects aimed at in the design of these cars are, simplicity of construction by eliminating all superfluous parts, and so arranging the necessary ones that they are able in certain respects to cope with the duties which might be required of certain eliminated parts. Lightness of certain details by utilising materials of the highest quality has been successfully attained, and simple and clean design is responsible for the great accessibility which is afforded.

Accuracy of manufacture accounts in a great measure for the efficiency and absence of wear of the complete machine, and how this accuracy is obtained will be gathered from the following remarks.

The wisest course that can be adopted in carrying out a successful design is in allowing the designer of the car also to design the jigs, simultaneously with each part to be constructed. A great measure of success in Mr. Coatalen's case was undoubtedly due to his having a free hand in this matter, and to his ability to design jigs and tools to suit each job, and at the same time to be adaptable to other jobs without much trouble. What is termed a "universal jig" was introduced by him to these works, and by means of this jig such parts as gearboxes and crank

Fig. II.

Universal jig as in Fig I. but turned 90° for holes on transverse axis.

chambers, not only of one size but of several sizes, can be machined with the greatest accuracy and convenience. The universal jig is shown in several of the accompanying photographs, and the first point that strikes one is its accessibility, adaptability, and the small size of such parts as require alteration or replacement to enable the jigs to be used for a different type of casting.

As distinct from the universal jig, the box jig usually employed is heavy, inaccessible and expensive, and it is only suitable for the particular casting for which it is designed. When the box jig has to be replaced by another one, as for instance when a certain machine is required to carry out another class of work, the removal of a jig of this weight involves the labour of two or more men and tackle.

The universal jig, on the other hand, consists of a large table of lozenge shape, fitted on a central spigot through which a bolt passes, and this is mounted to a mating spigot attached to the table of a horizontal boring machine. The jig table is free to be revolved through 90 degrees in either direction, and a second bolt, which works in a circumferential groove in the revolving table, affords the completion of the clamping down provisions.

The revolving table is located by means of a taper peg in either of its working positions. This fundamental table always remains on the boring machine, and upon it are located the necessary removable jig brackets for any particular job.

Along the centre of the rotating table is a wide groove, and there are three transverse grooves at right angles to it.

These grooves are used for locating the jig brackets, and the brackets themselves are each fitted with a flange or foot, provided with a steel tongue piece which mates in the grooves. The brackets are held in position by bolts and, as it is immaterial that the exact distance in the direction of the groove is maintained, these bolts do not form any part of the locating arrangement.

In order to build up a new jig for any particular job, the necessary brackets *only* are required, and these are located by means of tongue pieces somewhere near their required position. The holes in the brackets are then accurately bored in the same machine which is to do the job, and there can thus be no mistake about the location of the whole.

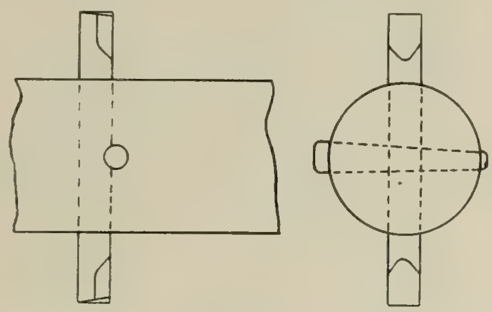


Fig. IV.
Arrangement of boring cutter and taper pin in bar.

Hard steel bushes are fitted into the holes of the jig brackets, the brackets are then marked, and can be removed if desired until they are required. Any number of other brackets to form jigs for other jobs can then be prepared in the same way, and it will be seen that the cost of production of a new jig of this type only involves the making of brackets in the way described. A study of the photographs will show the accessibility afforded by an open jig of this type, as the workmen can easily get to every side of it and see exactly how the work is progressing.

In the event of a batch of work, say crank cases, having been finished on the boring machine, and there being a press of work, say gearboxes, on the other machines the jig can be converted in a short space of time, by one man. In the event also of an alteration in design being found necessary, the alteration of the jig to suit the new type can be made very easily and cheaply, as it only involves drilling additional bolt holes or an alteration in possibly a single bracket. The universal jig is easily adaptable to a long or a short job, as for instance a four cylinder or six cylinder crankcase which is otherwise identical. In such a case the only alteration required is a spacing out of the jig brackets by sliding them along the grooves and drilling fresh holes in the rotary bed.

When all the holes are bored and faced in one direction the bed is rotated through 90 degrees in either direction by the removal of the locating taper pin and slacking the two holding down bolts of the rotating bed. This is clearly illustrated in Figs. I and II. In Fig. I. a gearbox will be observed with the bed arranged for machining the holes on the long axis. For the purpose of the photograph the job has been turned round so that the jig presents the plate containing guiding sleeves for the boring bar for the primary and layshaft, and also the holes for the selector rods. Referring now to Fig. II. the universal jig is turned through 90 degrees and the bar is in position for the selector shaft bearing hole to be bored. There are many other points on these jigs to which reference will be made later, but attention is drawn by these photographs to the accessibility of the job when supported by this type of jig.

A fine example of the universal jig is shown in Fig. III., where the same type of table is arranged for use in machining the top half of the crankcase. The actual boring machine has a double head; but the special jig employed for crankcase work converts this double head

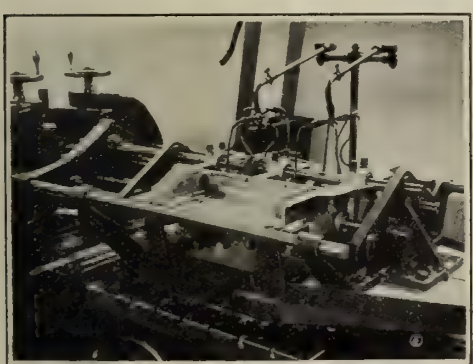


Fig. III.
Universal jig with five boring bar head for crankcase.

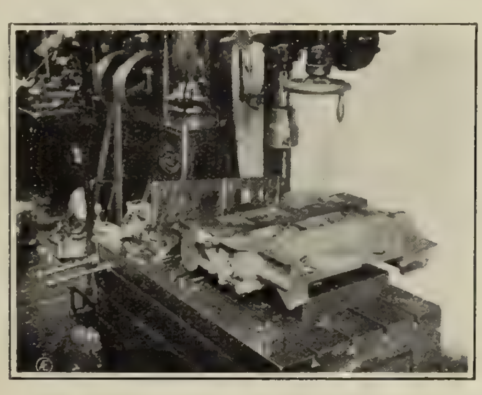


Fig. V.
Drilling jig for main bearing holes in top half of crankcase.

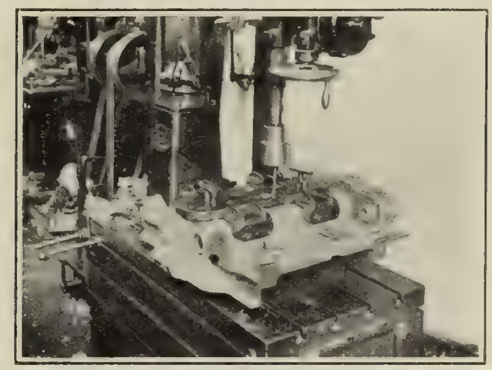


Fig. VI.
Drilling jig for the top half of crankcase. Holes for cylinder bolts, tappets and oil.

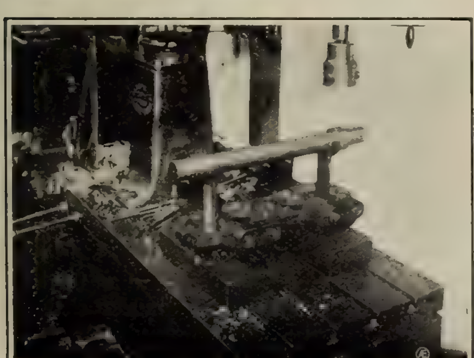


Fig. VII.
Showing jig Fig. VI. reversed, and locating method.

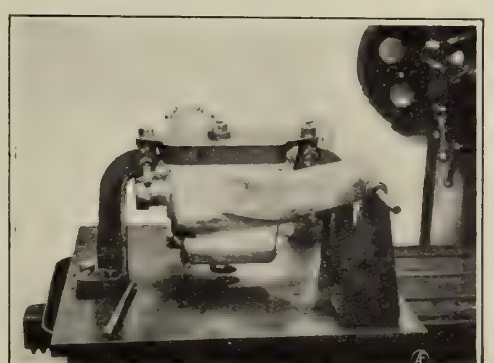


Fig. VIII.
Drilling jig for gearbox lug.

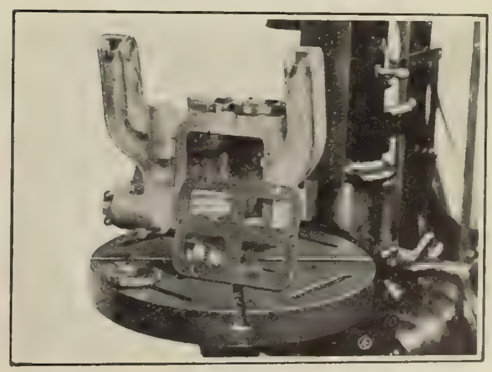


Fig. IX.
Drilling jig for gearbox bolt holes.

into one driving 5 boring bars by means of a train of gears.

The multiple head is shown on the left of the figure, and extension brackets will be seen from which project guiding bars, one on either side, these bars passing through two brackets on either side of the jig which form steady guides for the whole job. These bars are clearly shown in Fig. I. which figure also indicates very well one of the boring bars used in connexion with this work. This type of boring bar was introduced by Mr. Coatalen, and it possesses enormous advantages as compared with the old type of shell boring tools, of which some are shown for comparison at the right hand side of the table in Fig. I. The boring bar consists essentially of a hardened and ground steel bar, the hardening being necessary as otherwise the bar might seize in its bushes, and, when trued up again, the accuracy of the jig would be impaired.

Making the Boring Bar.

The boring bar is made in the following way. A steel bar of suitable length is taken and the correct positions of the holes for the tools are marked off from the drawings. These holes are so arranged that their distance from centre to centre is exactly equal to the distance from one face to the next, and so on. By this means the same hole which is used for the boring cutters is also used for the facing cutters, and it is only necessary for the work to traverse the length of the cut. The table can then be run back that distance and the boring cutters replaced by facing cutters: the set of faces is then trued up together. As distinct from the use of shell cutters, in this case it is not necessary to draw the boring bars from the work when the cutters are changed, and the act of changing only takes a few seconds.

When the location of the tools is marked on a bar, flats are ground where the holes are to be, and holes 10 mm. diameter are drilled and reamered dead to size. These holes are then filled up with soft steel preparatory to drilling the taper pin holes as shown on the sketch Fig. IV.

The pinholes are then drilled to take taper pins so that a small portion of each pin beds on a groove on its boring cutter. The cutters are then cut off 4 mm. longer than the finished length, each one has a groove filed in the middle of it, and is fitted to its taper pin with red marking. Each hole is then marked by a letter A, B, C, D, etc., and each cutter and pin is similarly marked. When all the cutters are in position and pinned, they are turned, or ground on a Norton grinder, and reduced to size within fifteen or twenty thousandths of an inch of their correct size.

They are then dismounted from the boring bar and the angle of cut is given by filing. The tools are then hardened and finally re-assembled in proper position; the tools being secured by the ground taper pins, are then ground dead to size. Each hole receives two cutters for boring, a roughing and a finishing one, the roughing cutter leaving forty to fifty thousandths of an inch for finishing, when working in aluminium. The same hole also receives facing cutters, where these are required, and as the holes have been drilled the exact distance apart it is only necessary to gauge for one facing cutter when a number of bosses are being faced by one bar. The great advantage of this system of cutters is the very low cost of manufacture as compared with shell cutters, and many cutters may be employed to work simultaneously, which is impossible with shell cutters on the same bar.

Operation and Location.

We will first consider the boring of a job as in Fig. III., the amount of work, the size of holes, and the time taken.

The matter of first importance in all jig work is the location of the job and the method of holding it down in the jig, and particularly in the latter case great precautions should be taken to prevent distorting the work, especially metal which suffers easily, such as aluminium. When a crank chamber top is received in the rough the upper face is rough milled on a facing miller, the amount of material removed being gauged from the crown of the main bearing seats; the job is then turned over and milled on the other faces and the bearer brackets, when the crank chamber is again reversed, and the upper face is finished off. The crankcase is located in the jig horizontally by resting on two parallel strips of the necessary thickness. Its vertical location is given by one of the bearers whose two bearing strips fit against two brackets on the jig, these bearing strips on the arms having previously been milled the correct distance from the centre of the case. The main bearing caps which have previously been milled on their faces and had their holes drilled in a plate jig, are bolted in position and the boring bars are mounted in the machine. Three of these bars, namely that for boring and facing the main bearings and those two for boring for the camshaft bearings, pass right through the casting, the tail ends of these bars being supported in a jig plate provided. The other two spindles, which are used for boring the holes for the bearings of the driving shafts of the pump on one side and the magneto on the other, are supported in one guide plate and two small brackets. It will be noticed that these boring bars are driven through universal joints.

When the job is thus mounted the cutters are put into operation and the table feeds the job to them, two ball bearings on the main boring spindles taking the thrust of the feed. In this operation twelve cutters are at work cutting the following holes:—Three main holes for the crankshaft 52 mm. diameter to take the die-cast white metal bearings. Six holes for each camshaft, these being 51 mm diameter to receive brasses which are pressed into them and which are finally reamered out in the jig. Two holes of 36 mm diameter are also bored to carry brass bushes, one on either side of the casting, for magneto shaft and pump shaft respectively.

Each of these holes is bored twice (roughing and finishing), and the main bearing holes are faced on both sides, the time allowed for the complete job being $4\frac{1}{2}$ hours.

Drilling the Crank Chamber.

The casting is then drilled, and in addition to the usual bolt holes, those oilways not already cored are drilled through the casting itself, no oil pipes being used.

The oil pump is bolted to the under face of the crank chamber casting, the oil being delivered through a vertical passage into a horizontal hole running the length of the casting. From this hole others at right angles, three in number, are drilled to a point over the main bearing brasses, and vertical holes are also drilled from this position to the brasses themselves. Fig. V. shows the drilling jig for the holding down bolts of the main bearing caps, these holes being drilled before the boring operations. The caps are spigoted, and the drilling jig is located in the spigot grooves clearly shown on Fig. V.

The same jig which serves for drilling the vertical oil passages is also used for drilling the holes for bolting down the cylinders, and for the tappet covers and their holding down bolts. Sufficient accuracy could not be obtained were this jig located by the main bearing capholes, and Fig. VII. therefore shows the jig reversed in order to indicate its method of location.

It will be noticed that this jig has a pair of projecting brackets each one terminating in the form of a bearing cup. A dummy shaft, fitted with two sliding collars or distance pieces, is fitted through the crankshaft bearings and the jig rests upon it as shown inverted in Fig. VII. The width of the two collars locates the jig longitudinally, and it naturally centres itself on the shaft, the steel bearing pieces shown on the half caps affording an accurate fit, and side location in the spigots for the bearing caps. A second plate jig not shown, gives the tappet holes and the four large holes for the pistons. In passing it will be noted that the cylinders are slightly offset; this accounts for the two camshafts not being placed symmetrically about the crankshaft.

It has already been noted that the gearbox is dealt with on the universal jig described, shown in Figs. I. and II., and we will therefore consider the gearbox machining before proceeding with the engine. Referring again therefore to Figs. I. and II. we see that the gearbox consists of a single aluminium casting without joints, except for its cover plate and the bearing housing plates.

When the casting is received in the rough the top face is milled on a vertical milling machine, the material being gauged from the edge of the holes in the casting.

The box is then removed to the jig shown in Fig. VIII. which is again of the universal type in that brackets are located to a flat table and the job is open round the sides. A flat face plate is used to locate the casting, and this is held at either end to the jig brackets shown, by means of substantial studs. The gearbox casting is bolted up against the plate, and is placed in such a position as will give the most convenient distribution of metal in the lugs, and so that the job is square on the jig. The three bolt holes in the supporting lugs are then drilled, these holes being used to locate the job in the future processes. Having now obtained means of locating the job for boring, it is mounted on the universal jig shown in Figs. I. and II. This jig is placed on a single spindle boring machine, the main bar boring for three ball race housings for the main gear shaft, these holes being 100 mm. diameter at the rear end and at the front end, and 80 mm. diameter at the outer end of the oil-retaining case where the shaft comes through. Each of these holes is in addition faced at both sides, the tools employed being of a similar nature

to those already described in connexion with the machining of the crankcase.

The table is then traversed across the machine and the layshaft holes, 80 mm. diameter at the front and 106 at the rear end of the box, are bored and faced. Also the holes for the reverse shaft are first drilled out and then bored to 30 mm. diameter finished size. This jig also locates the holes for the three selector rods on the same axis as the gearshaft. The table is then turned through 90 degrees and the hole for the gear change rod is bored, the position being shown in Fig. II. The drilling jig for the gearbox bolt holes jigs are shown in Fig. IX., and consist of a flat plate jig for the cover and circular jigs for three holes each for the end plates of the bearing housings: the flat plate jig is located on the centre line of the job. The complete gearbox is shown on Fig. X., it making a particularly neat job and one in which there is no opportunity for the oil to leak. There is a marked absence of oil-tight joints, and the gear operating shaft, entering at the top of the box, does not afford an opportunity for the oil to creep out.

The cylinders of the 1911 Sunbeam engine are cast in pairs, the engine complete being shown in Fig. XI., but the type for next season in the 80 mm. class consists of a monobloc casting. The drilling jigs of the two types are shown in Figs. XII. and XIII. (See next page). We will consider now the process of machining a pair of cylinders and the manner in which the castings are located. The first operation is to mill them over the tops on a horizontal miller; the casting is then turned round and the bottom face milled all over, there being no spigot left on the bottom face of the casting, the shape of which is shown in Fig. XIV. At the same setting in which the bottom face is milled, the two faces for the valve cover plates are also milled, the amount of metal removed being gauged from the centre line (Fig. XIV.), as these faces are used for locating the job in both the boring and grinding operations.

The casting is then ready for boring, and the jig shown in Fig. XIV. is used to locate the job for this process. The casting rests on a valve cover face upon a face plate on the machine and the cylinder top face is held against a face on the angle jig by means of two hook bolts passing into two core holes, one on either side of the casting. These bolts pass through the jig and clamp the casting to it, thus locating it on the long axis. The location on the small axis is given by the machine block on which the valve face rests, and two distance rods, shown in Fig. XIV. still further help to retain the

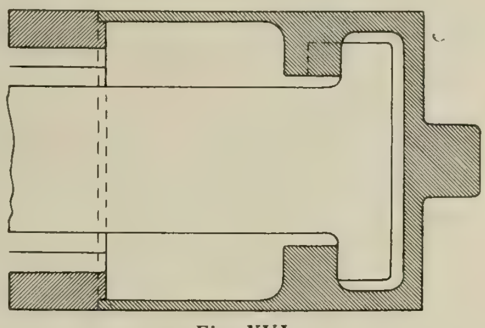


Fig. XVI.
Method of attaching piston for turning operations.

casting in the jig laterally. During the boring process a clamp is placed across the uppermost valve cover face, while long bolts pass downwards to the machine bed and firmly clamp the job down. The cylinder is then rough bored, but before taking the finishing cut the long vertical bolts are slacked back to eliminate any chance of their distorting the job.

The finishing cut is then taken with the job held as shown n Fig. XIV., fifteen to twenty thousandths of an inch being left for grinding. The casting is then removed to the drilling machine, and all the remaining work is done at one settng; the location of the job being given by the bottom face and by two bungs on which the cylinder bores fit. A pair of cylinders is shown in Fig. XII. mounted on the drilling jig which gives the necessary guide for boring the valve seats and facing them on their horizontal face. This jig locates the holes for the sparking plugs, which are fitted in the sides of the inlet valve pockets, and also the two stud holes on either side for attaching the flanges of the inlet and exhaust pipes respectively. There is also another hole immediately below these, through which the long bolt passes for securing the plates which fit over the boxes round the valve springs. On the tops of the cylinders six holes are drilled in each casting, these being for the studs fixing the top cover plate and water outlet pipe. The mating jig for this top water pipe is shown in Fig. XV., which will be referred to later. The valve seats are not mitred in the drilling machine, preference being given to direct hand cutting with a rosebit. The valve seats are only made about 2 mm. wide and at an angle of 30 degrees to the horizontal.

One point worth remarking is an important precaution taken in finishing the cylinder bores, as it is found that after the skin of the casting has been removed a molecular movement is set up in the cast iron. The cylinders of the Sunbeam engine are therefore placed, for two weeks, on the top of the case-hardening furnaces before they are finally ground to size. After this period the cylinders are replaced in the manner shown in Fig. XIV., and are ground to size on a Heald cylinder grinder.

Fig. XIII., to which reference has already been made, shows the box jig used in connection with drilling cylinder castings "en bloc," and this jig clearly shows

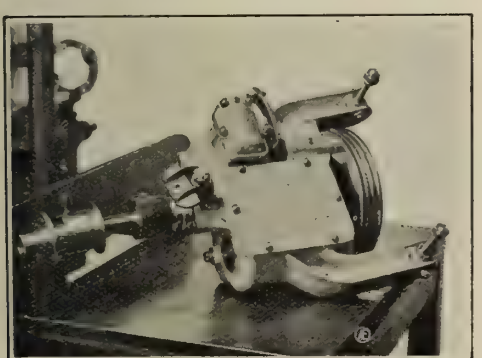


Fig. X.
Complete gearbox.

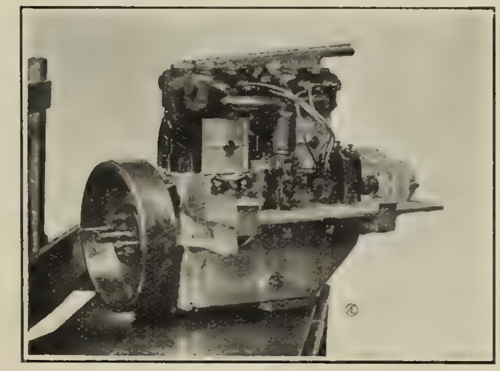


Fig. XI.
Complete engine with pair cast cylinders.

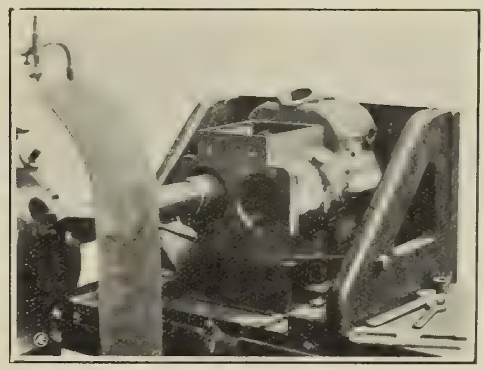


Fig. XIV.
Jig bracket for cylinder turning and grinding.

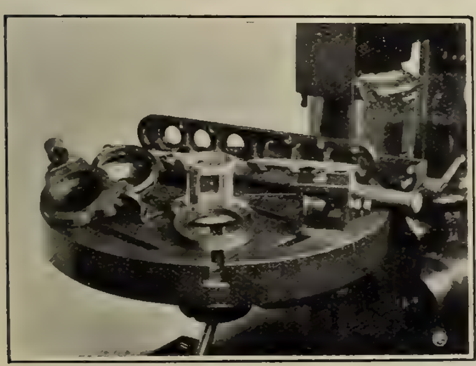


Fig. XV.
Drilling jig for top water pipe, oil strainer, water pump, and tappet box.

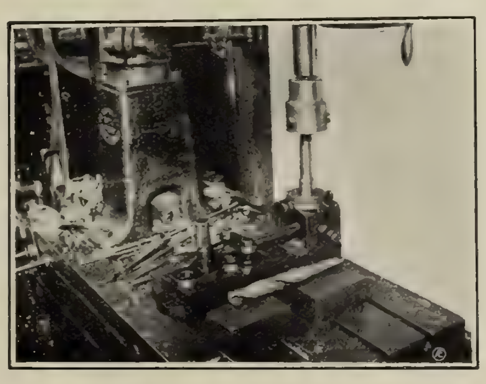


Fig. XVIII.
Jig for connecting rod machine.

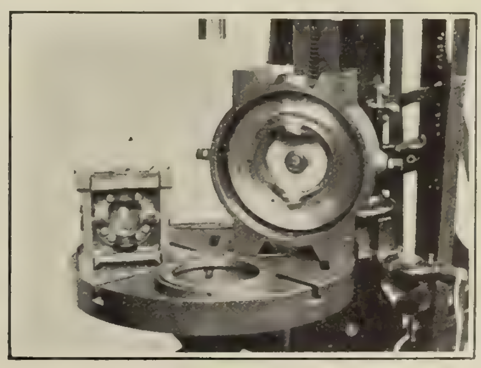


Fig. XX.
Drilling jig for universal joint foot brake drum.

the bungs upon which the two cylinders at the extremities of the casting are located.

Fig. XV., already referred to, also shows the drilling jig for the water pump casting, giving the holes for the cover plate and the inlet and outlet water pipes. The drilling jig for the oil strainer casting is also shown in this group.

The lantern pistons are made from malleable castings, and are first held in a three-jaw chuck to be bored to their true dimensions, and have the end of the trunk

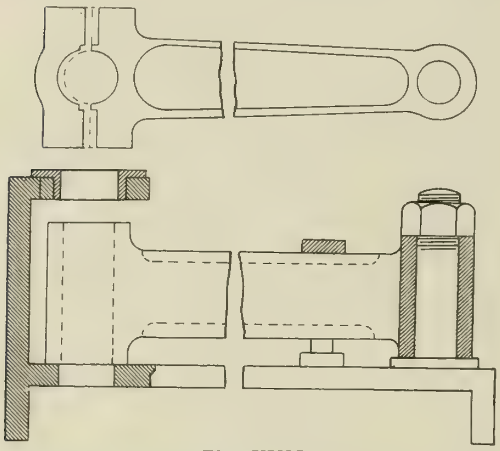


Fig. XVII.
Connecting rod, boring and facing jig. Method of constructing rod end from solid forging.

faced. This face is now held up against a bung on a special lathe fitted with a drawback arbor passing through the lathe spindle, Fig. XVI. This arbor is inserted into the piston and grips the gudgeon pin bosses, holding the piston firmly against the faceplate. A box tool is fitted on the cross slide of the lathe, carrying a tool for facing the top of the piston and also a compound tool for cutting out the ring grooves. Simultaneously with the cross traverse working, a tool fixed in the turret turns the cylindrical part of the piston to its diameter. The second operation is exactly the same, but with finishing tools, half a millimetre being left on the diameter of the cylindrical part for grinding. The boring and turning of a piston complete is priced at forty minutes. The piston is then fitted in a box jig where it is located endways for drilling the gudgeon pin holes and also, if required, the holes for lightening the piston; the gudgeon pin hole is then reamered out and the bosses faced.

The connecting rods are made of V.S.M. axle steel, drop forged with two solid eyes, one at either end, as shown in sketch, Fig. XVII. There is a decided saving in cost in dropping the big end complete and splitting it, as compared with manufacturing two separate forgings. The method of constructing a connecting rod is as follows. The small end is drilled and reamered dead to size, to be afterwards fitted with a casehardened bush. This hole is used for locating the job in the future processes, and a large drill is next put through the large end, and one side of the large end is faced, both sides of the small end having been machined similarly. The large end is then split, the location being made from the small end and, in the same jig which is used for this purpose, the spigots are faced by a gang of four milling cutters, traversing vertically. The cap is milled in the same way with the cutters reversed.

The bolt holes are then drilled in a plate jig and the cap bolted on the rod end, the jig shown in Fig. XVIII. being employed to hold the rod down and to locate the boring of the big ends. The jig, as will be seen, is suitable for rods of two lengths, the large end being located sideways so that the hole for the crankpin bush comes equidistantly between the spigots. A small block is fitted on which the rod is clamped down (Fig. XVII.), and the heights of the faces to which both the large and the small end of the rod bear are so arranged that the rod lies level in the jig. The

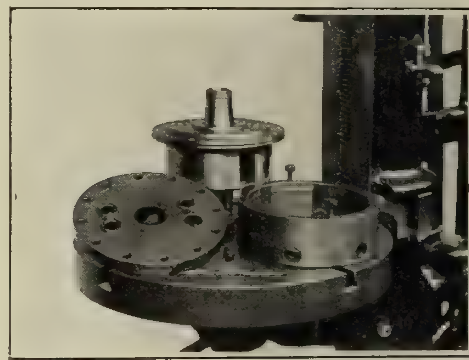


Fig. XXV.
Drilling jigs for differential casing.

large end is then bored out and, when the boring bush is removed, the same jig is used for the facing operation of the top side of the big end. The large end of the rod is then tinned and a white-metal run into it. The same jig as is shown on Fig. XVIII. is then used for machining the white metal. All these various jigs are carefully marked as to their purpose, including the number of the drawing, etc., and some of these markings may be discerned in the accompanying illustration.

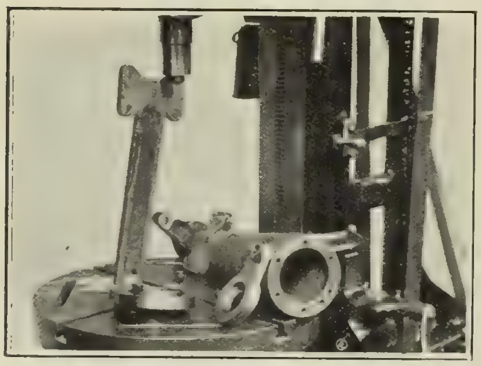


Fig. XXI.
Drilling jig for back axle tubes, brake hanger and springseat.

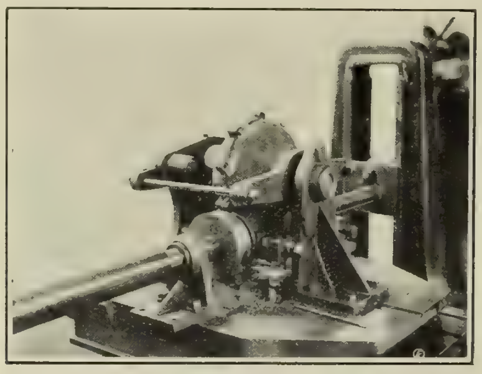


Fig. XXII.
Boring jig for worm drive axle casing.

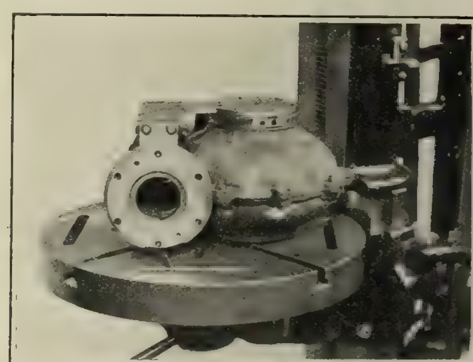


Fig. XXIII.
Drilling jig for worm drive axle casing.

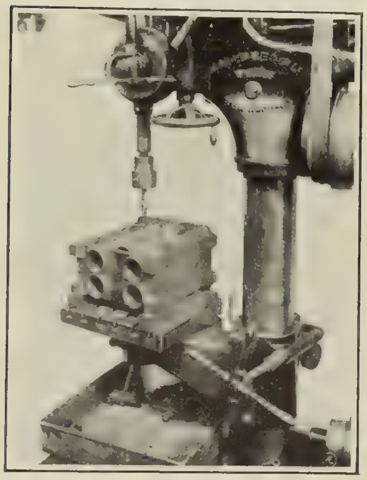


Fig. XII.
Drilling jig for cylinder pair.

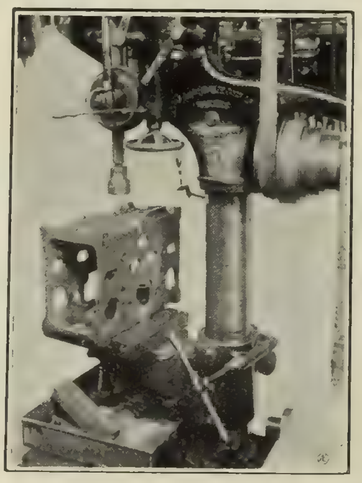


Fig. XIII.
Drilling jig for four cylinder block casting.

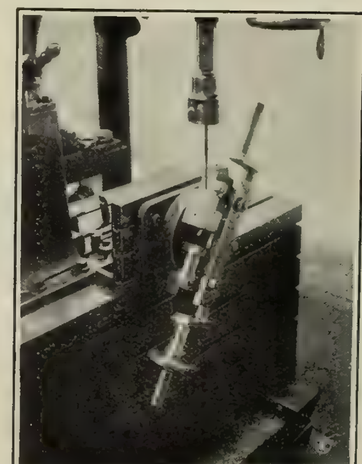


Fig. XIX.
Drilling jig for crankshaft.

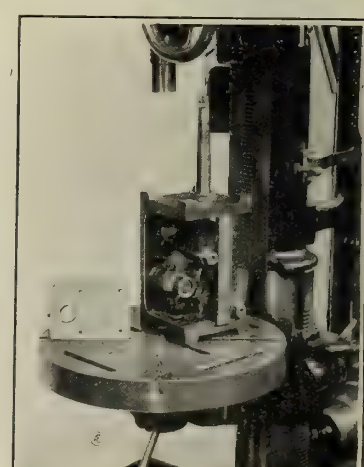


Fig. XXVII.
Drilling jig for steering box.

Mention has been made of the lubricating arrangements for the crankshaft, and Fig. XIX. shows the drilling jig used in connexion with the shaft itself; also the rotary carrier for the crank-

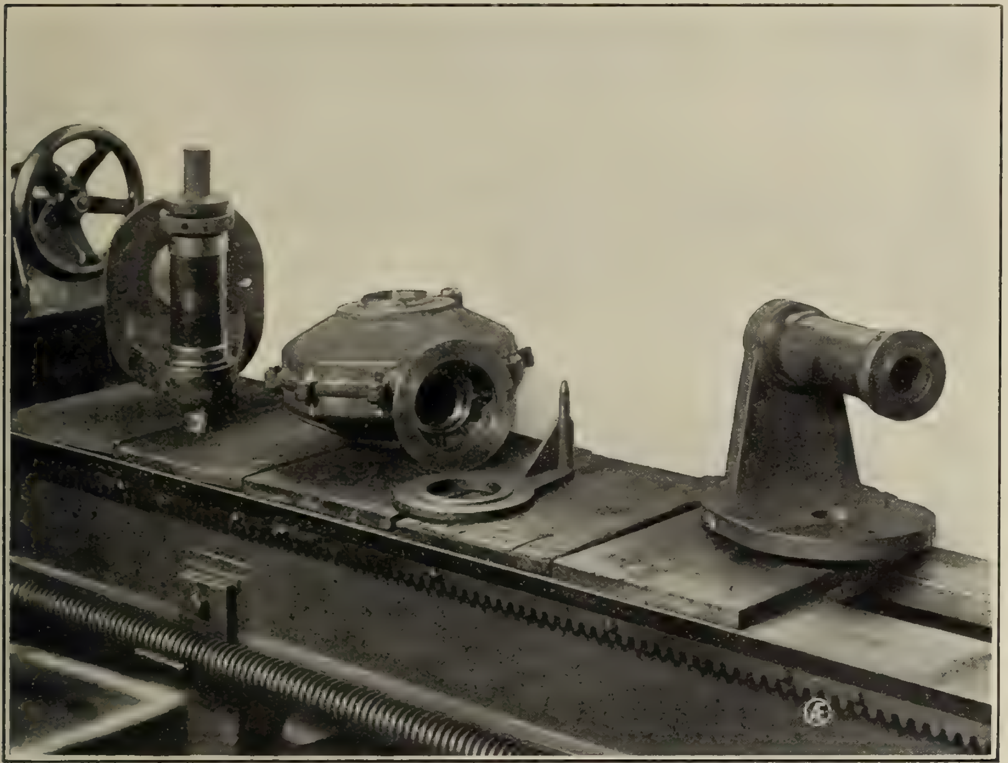


Fig. XXIV.
Boring jig for bevel drive axle casing.

shaft, which can be located at the desired angles by a locating peg. By means of the small bushed jig, oblique holes can be drilled through the crankpins for the oil, the jig being reversed when the holes in one direction have been drilled. The crankshaft is then rotated into a horizontal position and the oil holes drilled along the crank webs. It will be noticed that a locating pin fits into the revolving part of the crankshaft holder, and that the centre bearing of the shaft is located against this pin. A stout clamp holds the faces of the webs against the revolving table and so locates the shaft as regards its shorter dimension.

The universal joints are made of steel stampings, which are faced up in the lathe, and at the same time the four bosses which encircle the pins receive a light cut so as to provide a locating face. The joint halves are also turned round the outside to their correct diameter, and each half is fitted in a drilling jig, being located on this outside diameter. After the bolt holes have been drilled, as shown in Fig. XX., a pair are gripped by their bolts and placed in the jig, being located on the four faces previously referred to. The job is held in position by a star piece, and a bolt passing through the jig. The four holes for the bronze bushes are then drilled and reamered out, no other machining being done to this piece. On the right-hand side of Fig. XX. is shown the jig for drilling the foot brake drum, which is marked "drilling jig footbrake drum 80 x 120, drawing 206." The jig is for drilling and facing the holes for the pin of the universal joint, the work being first bored out and then located on a bung on this jig. The lugs are set vertically central, and the job is held by a central bolt and two adjustable set screws. Beside this jig is shown the drilling jig for the universal joint cover.

Passing now to the back axle, Fig. XXI. shows the drilling jig for the axle tubes, a combined jig giving the holes for fixing these tubes to the axle casings, and at the same time the holes for the spring shackle bolts. The method of using this jig is as follows:—

The malleable casting is first rough ground on the spring seats and the axle tube bored and faced where it fits up to the axle casing. It will be noted that this tube is flanged internally, so that there are no external bolts round this joint. The drilling jig has a bung upon it, which fits into the end of this axle tube, and is held firmly there by two hook bolts (see Fig. XXI.), the palm of the jig giving the necessary holes for the spring shackle bolts. The drilling jig for the brake hanger is also shown in this figure, the axle having been previously bored and faced at its outer end.

The axle casings are of two types, namely, for worm or for bevel drive. They are malleable castings, and the method of manufacture is as follows:—

A casting is first gripped on its back in the lathe chuck and the female spigot face is turned. The ball race housing is then bored dead true and the thrust spigot is turned to gauge. The male half of the casing is then machined similarly, and the castings are reversed on the face plate of the lathe, a bung being provided for the purpose of locating. In this position the backs of the castings are turned, and the holes on the various halves are marked off to suit the lugs in the castings. First, one half is marked and drilled, and then its mating half drilled to suit. Two halves are bolted together and rigged up in the jig shown in Fig. XXII. It will be seen that this jig is of the universal type previously described, and it is therefore unnecessary to go further into its details. The location of the job is on the two side brackets shown, and it is supported by set screws, two on either side.

Fig. XXIII. shows the drilling jig for this piece of work, for the bolt holes of the axle sleeves, and for the ball race housing. This jig is centralized on a bung in the axle tube hole.

Considering, now, the bevel axle casing, Fig. XXIV. shows a malleable casting in a finished state. The angle plate jig at the left-hand side of the figure is located on a bung on the face plate of the lathe, the two halves of the axle casing being bolted together, and fitting on the large vertical pin, which is hollow, and provided with a fixing nut. It will be seen that the top of this pin accommodates a male gauging pin, and when ma-

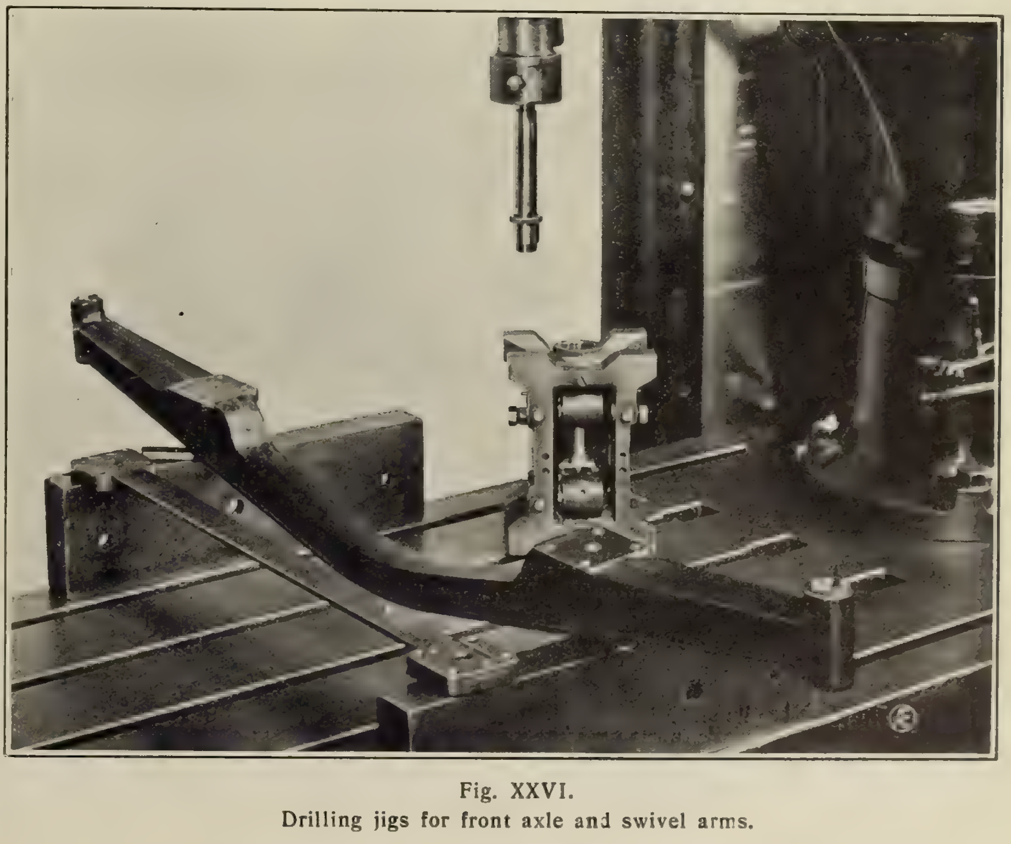


Fig. XXVI.
Drilling jigs for front axle and swivel arms.

chining the large flange for fixing the housing of the pinion shaft bearing, the pointer gauge is applied to this face, the distance being correctly given by the pointer coming in contact with the projecting portion of the male gauge before referred to. By the use of this jig a perfectly radial hole and true face are machined for the bearings of the bevel pinion.

At the right-hand side of the figure is shown a similar angle jig in another position.

The differential casing with its jigs are shown in Fig. XXV., and it consists of the box-shaped side, a malleable casting, and the flat plate side, which is a steel stamping. The method of working is as follows :—

One star pinion boss is centred and a pilot hole drilled. The flat side is then taken and drilled all over in the jig shown. Next, the box-shaped side has the four pin holes drilled for the spindles of the planet pinions, and locating planet pinions are fitted in position and the job assembled. The two halves are then coupled together and the bolt holes in the deep half are drilled through those in the shallow half, and any other holes for lightening the casing are then drilled round the sides of the deep half.

Fig. XXVI. shows the two jigs for the front axle spring clips and the swivel arms. The former is a flat drilling jig, and is placed on ground faces on the axle forging. The swivel, it will be noticed, is of the female type, and the box jig gives the vertical hole for the swivel pin. The swivel consists of a V.S.M. carbon steel stamping, oil hardened, and it is first turned on the stub axle, parallel in two places for the ball races, a tapered part separating the two. At the root of the stub axle a large radius is left, upon which fits a felt washer. This stamping is then milled on the faces of the lever bosses, and the stub axle is then ground where the ball races fit on. The holes for the steering arms are then marked off and drilled. The stamping is then placed in the drilling jig (Fig. XXVI.) and the vertical hole for the swivel pin bushes is then machined. This hole is of different sizes at the top and bottom and four faces have also to be machined in this jig. In the top fork fits a bush, which also comprises one-half of a ball thrust washer made of hard steel. In the lower fork is a hard steel bush, which protrudes through the bottom and terminates in a screw thread for attaching a brass oil cap.

A large brass oil cap also covers the nut and thrust washer at the upper end of the steering pin, this cap being screwed on the fixed half of the ball thrust washer before mentioned.

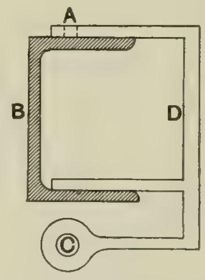


Fig. XXVIII.
Drilling jig for mounting gearbox in frame.

The steering box jig is shown in Fig. XXVII. The box consists of a malleable casting, the faces of the two halves being first ground true on a surface grinder. The holes in the flanges are then drilled in a drilling jig to accommodate the bolts for holding the two halves together. The work is located in this jig by inside set screws pressing against the cored part of the casting. The two halves are then bolted together, and the hole for the rocking arm pin is drilled with two different diameters and reamered with a double reamer. This hole locates the job in the jig for the main bore, a pin projecting from the jig fitting in the hole. The work is held against one of the ground flanges on the side of the steering box, there being one of these flanges on either side, as can be seen in Fig. XXVII. The casting is clamped down, as shown, and still further located by the two set screws shown in the figure. The work is now ready for the main bore, which is carried out by a boring tool of the type previously described, which cuts a hole of two sizes, namely, 45 mm. diameter by 97 mm. long, for the main bush and a pair of holes 38 mm. diameter for the bearings of the worm gear. In this jig also the two outside faces of these holes are machined. Attention is also drawn to the fact that the bearings for the worm gearing are slightly eccentric, thus giving an adjustment between the worm and the sector.

An interesting and useful type of small drilling jig is employed in connection with the erection of the gearbox in the chassis, and it will be necessary to describe how this is done. The gearbox is suspended in the chassis on three lugs, one in front and two behind. The front lug is bolted to its bracket and the bracket itself clamped to a cross member of the chassis, the rear end being held by two jigs shown in Fig. XXVIII. These jigs rest on a channel section cross member and support the box without clamping. They replace the eyebolts which are fitted in the finished job. The gearbox is now lined up to the flywheel by means of a scriber fixed to the universal joint by a small set screw provided for the purpose. When the box is thus got into line, the holes as given by the two jigs at the rear of the box are then marked off and drilled in the frame. The box is then permanently attached to the frame by the eyebolts before-mentioned, and re-set by the scriber, when the front hole is marked off from the bracket attached to the front lug of the gearbox.

It will be seen that the methods employed by the Sunbeam Motor Co. are very up-to-date, for the jigs which have been described are such that the work turned out requires a minimum amount of setting and fixing.

THE 33 H.P. HUDSON CHASSIS.

One of the most modern American designs.

IN discussing American touring car design generally, the fact was mentioned that there was a very large number of American cars having four-inch engines, and one of these is the 33 h.p. Hudson. In the design of this chassis, however, rather more attention has been given to engine work than is usual, with the result that the engine resembles European practice more closely than all but a very few other American designs. Of course, the proportions are not in accordance with present day ideas in this country, for the four-inch bore has a stroke of only $4\frac{1}{2}$ in., but, as reference to Fig. I will show, the cylinders are cast in single block, having the valves all on the near side, with cover plates. Moreover, there is an oil pump, driven from the camshaft, delivering to base chamber troughs, and the cross shaft in the front of the engine is driven by a skew gear instead of the usual bevel. One point, which perhaps is not made perfectly clear in the illustration, is that the crankshaft has two bearings only, it being claimed that the large diameter—2 in.—and the comparatively small length between bearings, render it stiff enough for all practical purposes. In fairness to the designers it must be allowed that there is no very noticeable vibration, unless the engine is forced considerably above its normal speed. Even then it is hard to say whether the vibration is not attributable as much to the very considerable reciprocating weight as to crankshaft oscillations. The latter are minimised by balancing the shafts rotationally on a Norton machine, and the pistons are, of course, weighed against each other. At the same time, there is no doubt that the engine would be improved greatly by the addition of at least a centre bearing and, if the manufacturers experiment with lighter reciprocating parts, no doubt the disadvantages of the two bearing crank will become more evident.

Both the main and big end bearings are white metal, the former being also bushed with phosphor bronze, though in the latter case the white metal is run directly into the connecting rod end. For the camshaft two phosphor bronze bushes are used, and the peculiar form of tappets should be noticed. It may perhaps be recollected that the valve diagram obtained from this mushroom-ended valve striker is slightly superior to that got from the ordinary roller end arrangement, and wear on the mushroom head is minimised by off-setting the cams slightly so that their contact tends to rotate the whole tappet in its guide. The length of the valve guides is a commendable feature, they being of steel and pressed in the cylinder casting, while the valve head diameter is satisfactory, this being an effective $1\frac{3}{4}$ inches. The peculiarities of the piston are very marked. Not only is the thickness of the wall exceptional, but its length is very great, and the gudgeon pin hardly commends itself to British ideas, although in practice it appears satisfactory. For the oil circulating system a plunger pump is used, driven by an eccentric on the camshaft, sucking oil from the sump and forcing it through a dashboard sight feed, whence it returns to the four troughs, the splash therefrom being relied upon to

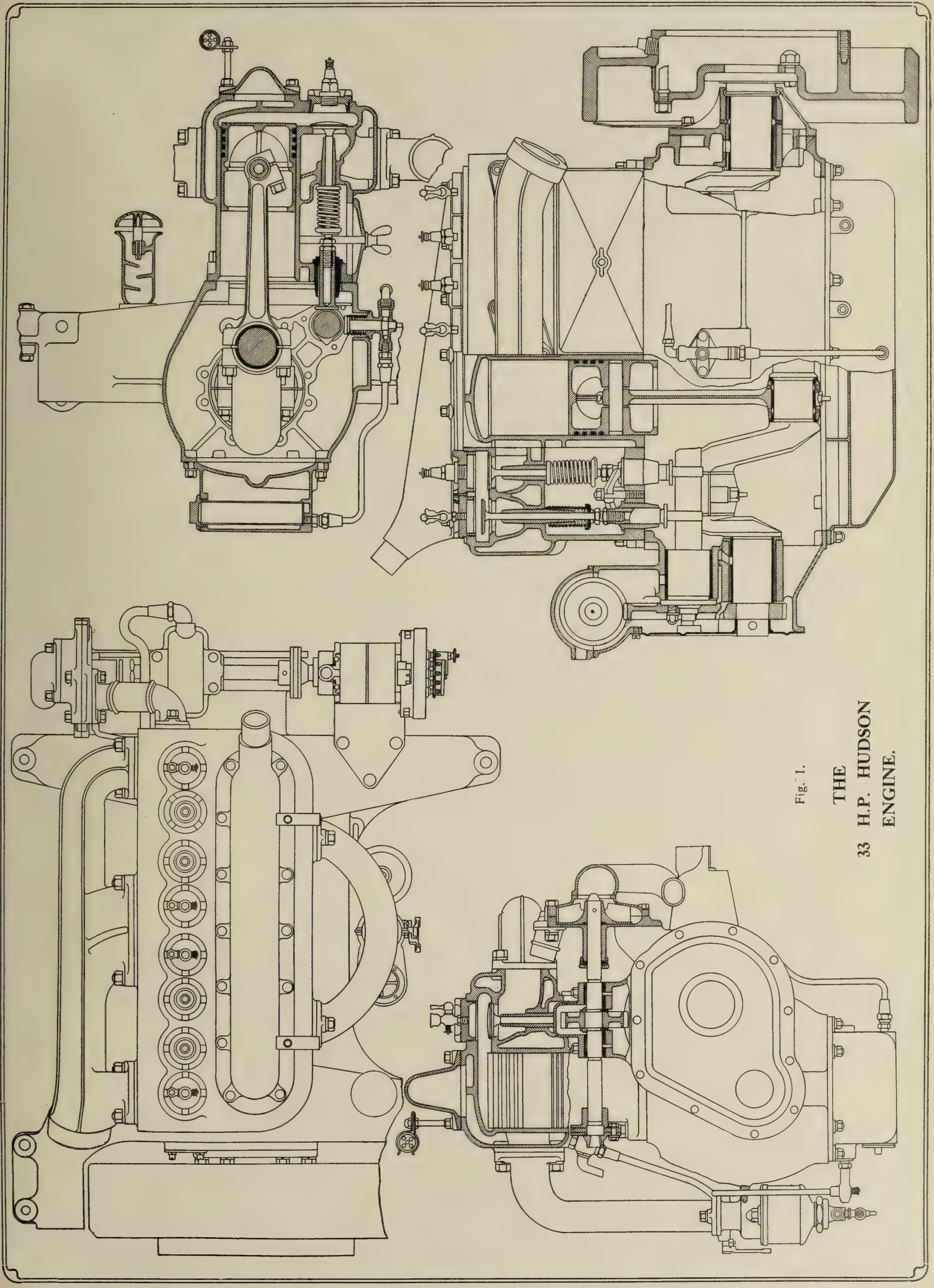


Fig. 1.

THE 33 H.P. HUDSON ENGINE.

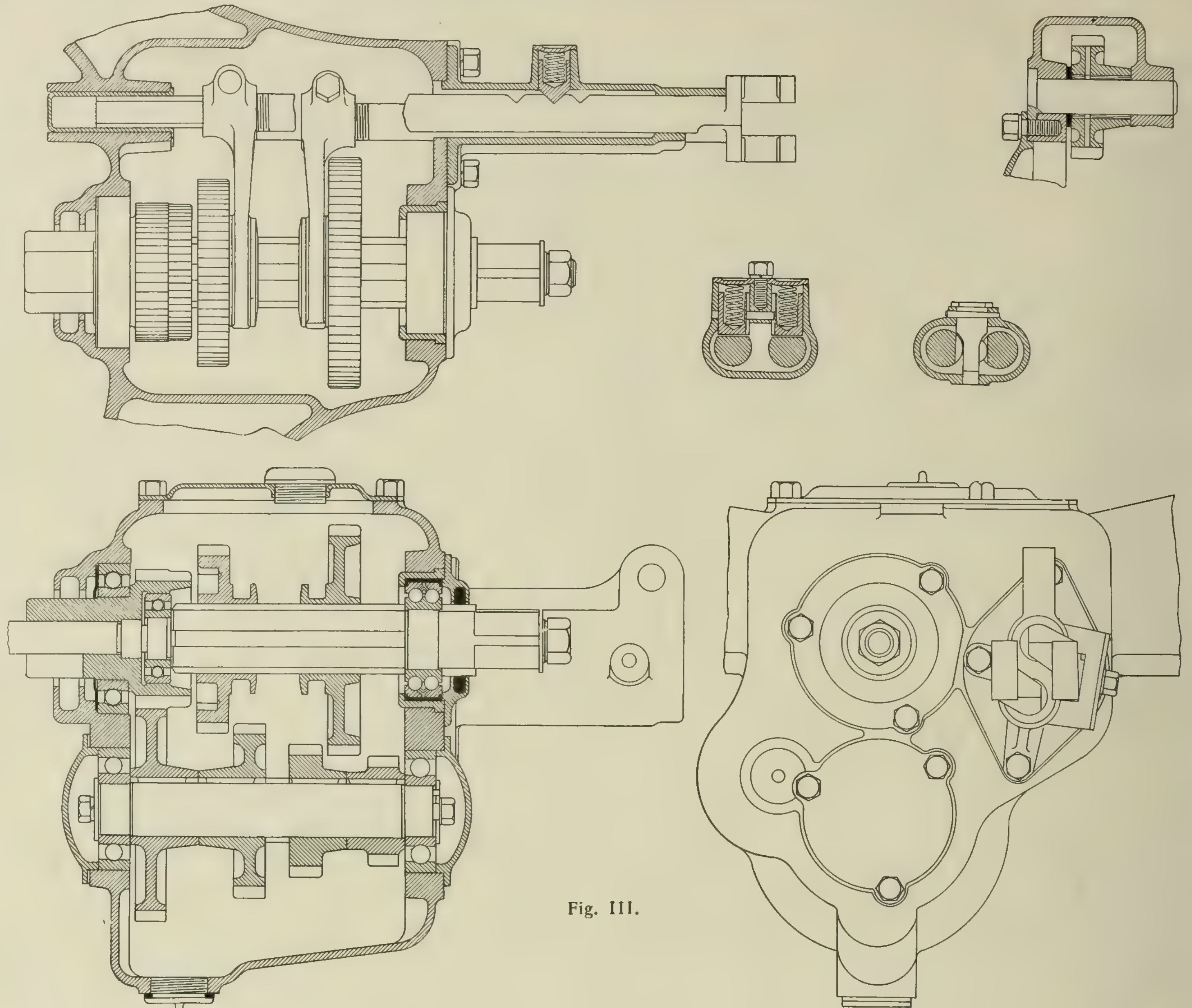


Fig. III.

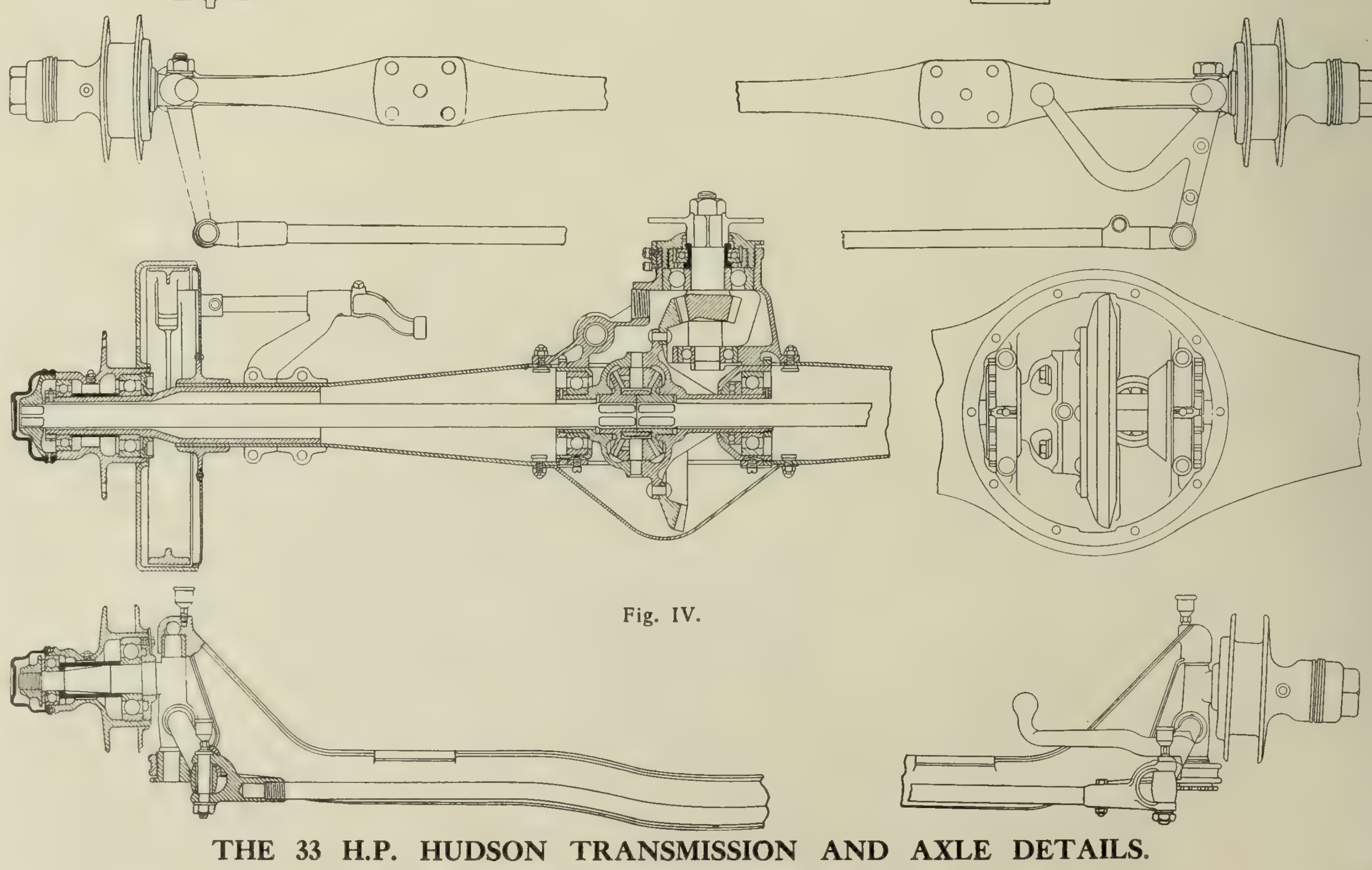


Fig. IV.

THE 33 H.P. HUDSON TRANSMISSION AND AXLE DETAILS.

lubricate all parts of the engine, with the exception of the cross shaft driving gear, to which a separate lead is run.

In the plan view the inlet arrangement can be observed, and also the magneto, which is not shown in the sectional view of the cross shaft, though the latter demonstrates the water pump. The use of the two-bearing shafts causes the upper half of the crankcase to become a very simple piece, and it comprises four arms which support the engine direct from the main side members of the frame, a rather unusual feature being the six bolts used to retain it in place. Passing on to the clutch, this is shown in Fig. II., and is entirely peculiar. The engine in this instance is on the right hand side, and the clutch spring is contained by a bored-out crankshaft end, while in connection with this a point particularly worthy of notice is the fact that the expansion of the clutch spring is limited by a small bolt (contained inside), furnished with a washer at each end. The length of this bolt is so proportioned that the spring cannot extend beyond the limit of travel of the other moving parts of the clutch, so that when it is inserted in the crankshaft the rest of the clutch can be assembled without the necessity for compressing the spring. None the less, the full power is available for the clutch actuation, and the rest of the construction may easily be followed from the drawing, it only being necessary to explain that the clutch has alternate discs of ground sheet steel and thicker steel discs, each of which latter is drilled to a honeycomb and filled with cork inserts.

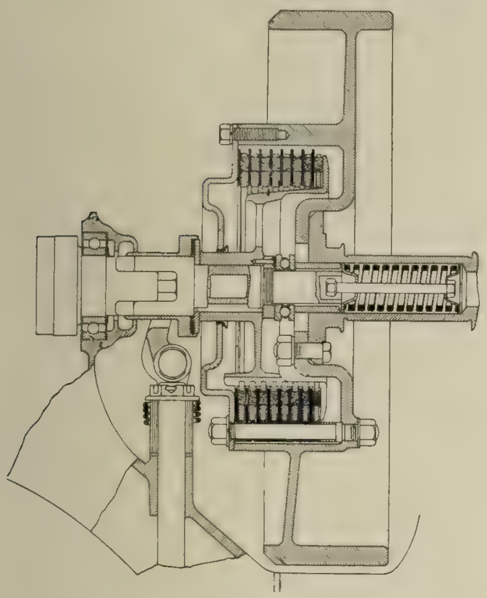


Fig. II.
The Hudson clutch.

Flywheel and clutch are also balanced, and from them the drive passes through a jaw coupling to the gearbox, which is shown in Fig. III. It will be observed that three forward speeds are provided with a splined shaft for the sliding gears, and a keyed lay-shaft. The method of securing the bearings to the latter with set screws and washers has a certain crudity, though it appears to be effective, and the bearings are of good size for the work they have to do, even the spigot being sufficiently large to withstand a good deal of rough usage. The purpose of the double ball race at the rear end of the box is to resist any accidental thrust from the propeller shaft. The control is quite straightforward, consisting of an ordinary gate with an interlocking striking mechanism, the details of which are made sufficiently clear in the figure, and an important point, which is not brought out very well in the illustration, is that both gearbox and engine form a single unit, the casing of the box having arms brought forward and ending in segments of a circle, which are faced and bolted to corresponding faces on the crankcase. The arrangement of the clutch is such that very little length is wasted, so there seems to be no lack of rigidity in the gearbox attachment. In this chassis, therefore, the only rigid length is that between the engine arms, while a good length of propeller shaft is obtained.

Two universal joints are used, these being of a patented variety turned out in very large numbers by one of the component firms, and allowing a sufficient amount of telescope, for there are no radius rods, although a torque stay is fitted. Both the front and rear axles, which are shown in Fig. IV., are examples of the work of the Metal Products Company, and therefore they are very similar in detail to those used by a number of other makers of cars similar to the Hudson. The central casing, with the sleeves, is a single pressing, the differential bevel pinion and the bearings therefor all being mounted on a steel casting which bolts up to the pressed steel. The rear axle ends are flanged steel tube, and these, together with the brake brackets and spring pads, are attached by rivets, being first of all a considerably tight fit. This back axle is very well provided with bearings, but it will be seen that these are all adjustable, this feature being introduced to allow the axle manufacturers to adjust the relative positions of the bevels until quiet running is obtained. An alternative design for the rear axle end, which may be employed for next year, is shown in Fig. VI., the difference being the substitution of a single roller bearing for the two ball races. Incidentally this figure shows the brake work in full, which is not given in Fig. IV. Both brakes are steel rings possessing some natural spring and lined with Raybestos, and both are operated through a toggle gear which is positive in both directions. Similarly with respect to the front axle the ball bearing pattern appears in Fig. IV., while in Fig. V. a modification is shown in which parallel roller bearings are used. Fig. V. also shows that no ball bearing is provided for the steering pivot, this having a plain phosphor bronze bushing, which is all the more curious because the Hudson is unusual, for America, in having a rather neatly designed steering box, with a good sized ball bearing on each side of the worm.

A neat detail in connection with the steering gear is a spring lid covering a really good sized orifice immediately above the worm. This is fashioned on exactly the same principle as the usual Bosch magneto bearing cover, so the replenishing of the grease in the steering box is quite an easy operation. No chassis view has been included, because there is very little detail on it not shown satisfactorily in the other illustrations. Briefly, it consists of two perfectly plain side members, with pressed cross members in the usual position, and the spring hangers are all stampings, in fact, there is scarcely a steel part throughout the chassis which is not stamped, from the crankshaft downwards. Semi-elliptic front springs are used with three-quarter elliptic springs at the back, and a feature of the latter, which incidentally gives testimony to the vileness of the American roads, is shown in Fig. VII., this being the design of the shackle with an arched top which prevents the shackle from reversing its position on a bad bump, a thing which it would otherwise be very liable to do.

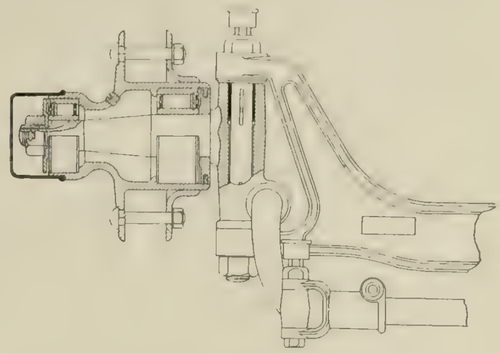


Fig. V.
Hudson front axle end, alternative design to Fig. IV.

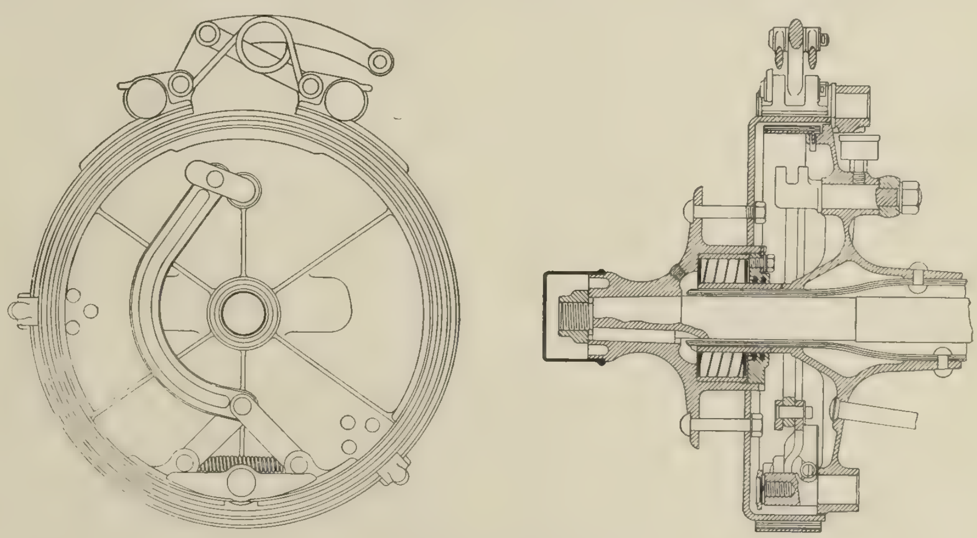


Fig. VI.
Hudson rear axle end, alternative design to Fig. IV.

This car is the product of one of those factories which have been mentioned else-

where as possessing practically no machinery. There is not a portion of the vehicle actually made in the Hudson Company's works, but similarly, every part of it is made to their design. The majority of the parts come into the works singly, and are assembled by Hudson employees, so the factory resolves itself really into an erecting and body finishing establishment. The pros and cons of this method are discussed elsewhere in this issue, and so no more concerning it need be said at present.

On the road the engine shows itself capable of developing at least as much as its nominal power, while it possesses good accelerating qualities. The gears are moderately silent, and having had occasion to observe a good many similar chassis undergoing road tests, there is

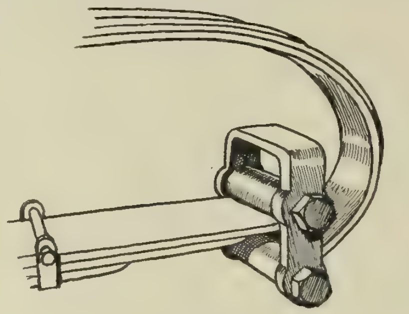


Fig. VII.
Rear spring shackle on Hudson.

no doubt that there is very little variation between one box and another. The comparatively large pitch, 5 and 6 inch diametral, renders the sound more of a growl than a high pitched note, and the layshaft drive is very quiet when the top speed is in engagement. Similarly, the engine is considerably more quiet than that of the majority of corresponding American cars, owing to the enclosed valves and the absence of small bevel gears, but the steel and phosphor bronze timing gears are quite audible, although provided with helical teeth. The clutch is particularly sweet in action, and the control generally is quite convenient, which can by no means be said of many cars originating on the other side of the Atlantic.

SOME FACTORS IN TYRE ECONOMY.

With some interesting comparisons of wood and wire wheels.

By J. A. Mackle.

It is satisfactory to note that recently there have been signs of awakened interest in the matter of tyre performance. The press correpondence on the subject of the relative life of tyres on wire and wood wheels, and also the suggestion for an extended endurance competition for standard size of tyre, all show that special attention is being devoted to the subject.

In these days of efficient and economical engines, the matter of tyre cost and upkeep has assumed a position of far greater relative importance than was formerly the case. Taking the average car of medium power as an example, it may be stated without much fear of contradiction, that the tyre item comprises sixty-six per cent. of the upkeep bill. One often hears the dictum "the pneumatic tyre made the motor car possible." It is probably more correct to say that the cost of upkeep of the pneumatic tyre has made the motor car impossible for many would-be owners whose purses are limited.

The subject is not one which admits of much discussion. The cost of tyres—both prime cost and upkeep — constitute the most serious item in the motorist's expense account, and therefore any feature or improvement which promises to effect an appreciable saving under this heading is deserving of the closest attention. Now, the recognition of the importance of tyre cost has brought forward many new protective devices during recent years. Most of these are in the nature of unpuncturable and unburstable coverings, and it will be agreed that, in almost every case, these are not preventors of wear as much as they are preventatives against sudden breakdowns on the road. The burstless band simply defers the necessity for changing the worn-out cover till the garage is reached. What is greatly needed is the means of reducing the normal wear of tyres; some method of postponing that evil day when the canvas interior begins to peer through the lacerated rubber strip and to threaten impending trouble on the road, and all the attention and study which has been devoted to this subject has resulted in the production of no device more worthy of the motorist's gratitude than the wire wheel.

When the design and construction of wire wheels was undertaken, many years ago, the primary advantages which seemed capable of realisation were greater strength and safety, and a considerable reduction in dead-weight. These features are fully recognised by all present-day car owners, but (not so well known except by those motorists who have had trial of both types of wheel), is the undoubted fact that the wire wheel offers a great advantage in the matter of reduced tyre wear. The experience of many observant owners of unquestionable motoring standing has conclusively proved that the tyre fitted to a wire wheel undoubtedly will give a greater mileage than a similar tyre on a wooden wheel.

Striking confirmation of these results is supplied by the recently published report of the manager of the Daimler company's hire department at London. Careful records have been kept of all tyre replacements and repairs, and the following figures show the mileage obtained (on heavy covered cars, by the way) from 100 non-skid 935 x 135 covers, half of which were fitted to wire and half to wooden wheels.

Total mileage obtained from 50 n/s 935 covers taken from wire wheels, 172,731 miles, average 3,454.

Total mileage obtained from 50 n/s 935 covers taken from wood wheels, 102,524 miles, average 2,050.

Average miles per cover, 3,454 on wire wheels, 2,050 on artillery wheels.

The reasons for this great feature of tyre economy are not difficult to understand. In the first place considering in particular the case of a heavy car, long before normal wear should have worn down the top layers of the cover, gradual disintegration of the tyre occurs, uneven wear is set up, and, without warning, a burst finally completes the ruin. Such is the case with the wooden wheel, and the reason is certainly that the heat set up by the friction of the tyre on the road causes deleterious physical changes in the condition of the rubber, and this begins to lose its wearing and homogeneous qualities far sooner than if the temperature had been kept as low as that of the atmosphere. Anyone who understands the qualities of rubber can realise how harmful it would be to leave a tyre closely exposed to the heat of a strong fire; the conditions encountered when fast speed is attempted on a rough road are very little different, as far as the heating and damage of the tyre are concerned.

With the wire wheel a strong current of air blows constantly over the rim, and the cooling of this comparatively thin metallic member keeps the interior (the vital part) of the tyre reasonably cool. In the case of a wire wheel, a thin steel rim lies between the tyre interior and the cooling air; the wooden wheel opposes a thick and heavy barrier of non-conducting material. The sceptic has only to feel the temperature of the cover and rim of a wire-wheeled car immediately after a long run, to appreciate the importance of this point. Hardly less important—in fact, closely bearing upon the above feature—is the relative difference in the moments of inertia of the two types of wheel. When the car encounters an obstacle on the road, this obstacle is struck by the revolving wheel. The wheel is slowed down, and the car rises over the obstacle. The slowing down of the wheel liberates a certain amount of energy; a small part of this is communicated to the obstacle, and the remainder has to be absorbed by the tyre, mainly by molecular movements in the material itself. It hardly needs emphasis that this action—inevitable, it may be agreed—is harmful to the tyre and eventually results in wear. Therefore, if this process can be reduced in magnitude, a step has been taken in the direction of lessened wear. Now, the inertia of a revolving wire wheel is but half, or very slightly less, than the inertia of a similar wooden wheel. Surely, then, the saving here is obvious.

There are many other matters of importance, though these are not so vital as the two chief reasons given above. The smaller weight of the wire wheel calls for the expenditure of less energy, when the wheel and axle are raised by the passage of the car over a bump in the road, while similarly there is less gyroscopic torque set in when the car is rounding a curve. It is also conceivable that the small but appreciable "springiness" of the wire wheel acts, in a measure, as a spring-drive when the car is being started, and in this way relieves the rear tyres of a little strain.

MOTOR CYCLE DESIGN.

An examination of current practice and some suggestions for future improvement.

By H. Linley Byrd.

SPEAKING generally, motor bicycles are designed in a more rule of thumb manner than cars and, as their designers do not pay so much attention to theory, the author will avoid mathematics as much as possible in the following. Before going into any detail it is necessary to consider the general principles on which the design is based. At present motor cycles are used chiefly for pleasure purposes, but in a few years' time they will be used for business, taking the place of the pedal bicycle. This being so, the design of the machine should be such that no time is wasted in starting the machine, filling the tanks, putting up the stand, putting the machine away, putting on overalls to keep clean or dry, cleaning the machine, etc. All these things are of very little importance when the machine is used for pleasure, as time is of very little object, but, for business purposes, for instance, five minutes saved on a five mile journey is equivalent to five miles per hour on the average speed, which is equivalent to ten miles per hour on the maximum speed. It is also necessary for the rider to be able to keep reasonably clean. The author proposes to deal with the different parts of a motor cycle in the order in which the petrol flows and is converted into work, viz.: Tanks, carburettor, engine, control, drive, frame, etc.

Tanks.

Some of the modern high power twin-cylinder machines consume much fuel, but in any case it is advisable to have a large petrol tank. As it is possible to go at least thirty or forty miles on half-a-gallon of petrol, there is no object in making the tank to hold more than two-and-a-half gallons, but anything under two-and-a-quarter gallons is rather inconvenient for the rider, for the following reasons:—As it is necessary to have always half a gallon in the tank for unexpected or prolonged journeys, with a one-and-a-quarter gallon tank this only leaves three-quarters available, so the tank has to be filled two and two-third times as often as a two-and-a-half gallon tank, and it takes quite as long to put in three-quarters of a gallon as two gallons, as it will be frequently necessary to fill the tank from two cans. In addition to this the owner, when away from home, frequently has to pay for two gallons when he only uses one, or has to pay an enhanced price for that single gallon. The wide tanks necessitated by the large capacity enable the rider to grip the machine with his knees, keep his trousers away from the belt and the hot engine, and at the same time give the designer plenty of room for large stoppers. A tank up to eight or nine inches is quite comfortable, and is not in the way, and tubes through it immediately over the valves make the latter quite accessible for grinding in. At the same time, the tank must be well gussetted and supported, or otherwise there is strong probability of it springing a leak.

After deciding the size of the tank the next most important thing is to be able to fill it quickly, preferably without a funnel, as this is a nuisance and hinders the flow of petrol. Two-inch stoppers are about the smallest size which will enable this to be done without spilling any petrol. To enable one's hand to be put into the tank three-and-a-quarter stoppers would be required, but these are really too big. Hinged stoppers with a fine strainer reaching to the bottom of the tank are preferable to a gauge glass in every way, for they do not leak, and they show the depth of the petrol in the middle of the tank and so are more accurate, while the strainer might be marked to show the amount of petrol in the tank. The stopper should be designed so that it does not leak if the machine falls over, and at the same time, it should have an air vent. The strainer stops the petrol from splashing through this when the tank is full, but there is some difficulty in making a hinged stopper petrol-tight, as well as quick to open and light in weight. A good piece of cork makes a fair washer for a short time, but cork composition might be better if the petrol did not disintegrate it.

There should be a three-way supply tap combined with a filter, one position being off, another opening to the lowest portion of the tank, which should be at the near end so that the carburettor is not starved when going up hill, the third should connect with the tank about an inch above the bottom, so that half a gallon of petrol is left in when the tap is in this position. If a petrol injecting device is fitted the drain tap should be large, say a quarter-inch bore, so that the tank can be emptied quickly, and should be so placed that when opened the petrol cannot get on the belt. It is even more important to have a large oil tank, say two-and-a-quarter quarts, as the right oil cannot always be obtained as easily as petrol. The tank stoppers should be as large as possible, and the oil pipes should be at least a quarter-inch internal diameter, as the oil used for air cooled engines is very thick.

Carburettor.

Ninety-nine out of every hundred motor cyclists drive on the exhaust valve lifter because if the throttle is moved the mixture is spoilt. Surely it must be admitted that this is rather a bad way of driving the machine, as it causes it to go along in a series of jerks, which must do harm to the engine, belt and tyres. Now, with a good automatic carburettor one can open the throttle suddenly to the full extent and the engine will give its full power without a falter. The design of an automatic carburettor for a motor bicycle is really a more difficult problem than for a car, as the range of speed is greater than for a car engine. The engine should respond to the throttle immediately and the carburettor should not restrict the power or the speed of the engine in the slightest degree. Therefore the semi-automatic carburettors fitted to most cars and a few motor cycles are not what is required. If the automatic air is worked by the throttle, the mixture cannot be correct while the engine is accelerating, as it is bound to be weak just at the time when it ought to be stronger, unless some means are taken, as in some carburettors, of reducing the inertia of the petrol to a minimum. With a floating choke tube or funnel, somewhat similar to the "Rover," it is possible, by increasing its inertia, actually to make the mixture richer while the engine is accelerating; some other carburettors which depend on the suction of the engine could be altered to obtain this effect by the same means. Most of these are unsuited for motor bicycles, as the moving parts are not balanced, and so are affected by the roughness of the road and the vibration of the engine. Some carburettors with the pistons moving horizontally might do if these moved either across the bicycle or backwards, to reduce the amount of air. Friction of the moving part should not be relied on entirely to stop the hunting, as it varies according to circumstances. A little grit will make the friction very much greater, which would perhaps cause the carburettor to give an incorrect mixture at all times. On the other hand a drop of oil might make the moving parts more free than they were intended to be.

It seems advisable in order to get the petrol well mixed with the air to take the whole of the air past the jet, but if this is done the area of the choke tube round the jet, as anywhere else, must increase in size as the speed of the engine increases, or the throttle is opened, so that the engine is not starved of gas. This is done to a limited extent in the "Rover," and in a most ingenious way in the "Polyrhoë." The same effect as regards the throttle only can be obtained by having two or more jets. Two or more jets are excellent for obtaining an increased range of speed, but they are not sufficient by themselves. No carburettor can be absolutely automatic unless it also takes into account the temperature and the humidity. The former might be allowed for to a certain extent by having a needle valve in the jet made of a different material to the rest of the jet, say nickel and brass, but it is hardly practical. These two adjustments should be made by hand preferably by an adjustable jet, which would also be a help in starting and would avoid flooding the carburettor.

The body of the carburettor should be easily detachable without disturbing the inlet pipe, control wires, or petrol pipe. If the jet is adjustable, and the body of the carburettor can be dismounted quickly, it is not necessary to be able to take the jet out without tools, although it should be accessible. It would be an advantage if the jet was visible while riding. Some simple and reliable means should be adopted for varying the level in the float chamber, for quickly detaching the choke tube, and for taking the air from some part of the machine where there is no dirt. Also it must be borne in mind that a manufacturer cannot afford to put an expensive carburettor on a motor bicycle, however good it might be.

Engine.

The majority of engines have single cylinders with a capacity just under 500 C.C., and a stroke not much in excess of the bore. Twin-cylinder engines of about the same cubic capacity are so rare that a large number of people do not know that they exist. This is probably due to the lack of reliability of the early twins of all sizes, as twin cylinder engines were not fitted with the high tension magneto until the accumulator had almost disappeared from the single cylinder machine. Their automatic inlet valves called for endless tuning up, and the inlet piping was often so connected to the engine, that one of the joints leaked and caused carburation troubles. Now that the makers see the folly of their ways, it is possible that we shall see more of the medium size twin, as at its best it is about 30 per cent. lighter than a single cylinder engine of equal capacity, it will run faster and slower, gives the same power, runs more smoothly, accelerates more quickly, and is not so severe on belts and tyres.

For side car work the general practice is to have an engine from one and a half times to twice the power of the above; as a 5 h.p. single cylinder engine is rather heavy and uncomfortable twin cylinders are almost invariably used for this work, but the power of the single cylinder is gradually increasing. A great many twin cylinder machines are now in use, but many of the well-known makers seem to avoid them as they do the plague. Why this is so seems inexplicable, as they must lose a large amount of trade in consequence. A twin cylinder of 650 to 700 CC. should not weigh any more than a 500 CC. single cylinder engine, and would only require the same fittings, except the magneto.

As the rating for all sorts of competitions is based on the capacity, it would seem at the first glance that the effect of this formula would have tended to shorten the stroke. That this is not so is probably due to the following reasons :—The short stroke engine is more liable to pre-ignition, and so it is necessary to have a lower compression; it overheats more readily, as the area of the combustion chamber is greater and the radiating surface less than that of the long stroke engine, and so the compression has to be still more reduced; lastly, owing to the time lag in firing the charge, the speed of revolution cannot be very much higher. The second of these defects could be avoided if water cooling was adopted; the last would be cured by having two sparking plugs in series placed in the combustion chamber. Several of the magneto firms are experimenting in this direction, and already some of them have machines on the market. Likewise a two pole pattern could be used as one of the plugs.

The lubrication of motor cycle engines is in a most primitive state. No doubt the question of expense has had a lot to do with the neglect of this question. The drip feed in conjunction with an ordinary pump and the pump with a spring to force the oil into the engine at a certain rate are improvements, but they are not by any means ideal. Probably the simplest way of getting a constant level of oil in the crank case would be to have a drip feed, giving an excessive amount of oil and an overflow into a sump, which might be placed under the valve gear case. The sump could then be emptied at intervals by the ordinary pump, which would deliver it back, through a filter, into the tank. The overflow should be placed in the front of the engine, so that the level of the oil would be higher when the machine was going up hill.

Very little attention has been paid to the question of keeping the oil in the crank case cool, and this is most important, as if the oil is kept comparatively cool, the engine gives considerably more power, uses less oil, which can also be thinner, and the engine is less liable to gum up. It should be quite a simple matter to cast radiating ribs on the crank case and, if these were not too close together, it should not be difficult to keep them clean, especially if the front mudguards were suitably designed.

Motor cycle engines are usually too noisy, the tappets causing quite an unnecessary clatter, for they could easily be silenced. The explosions can be silenced by taking a long pipe from the valve chamber along the back forks right to the end of the back mudguard, and this long pipe also helps to scavenge the exhaust gases. A great many riders object to a silent motor bicycle, as it necessitates blowing the horn very much more frequently than is necessary on a silent car, and there is a good deal to be said for this argument, although it is really not sound.

Engines are mostly too heavy for the power they give, and that falls off very rapidly when the speed slows down. By scavenging and super-induction the power can be doubled, but the compression must be reduced or the engine will overheat. There is not the necessity for a high compression with scavenging and super-induction, as the higher the compression the smaller is the increase of power due to scavenging. At the same time, if the compression is reduced to any extent, two or three sparking plugs must be placed in the cylinder as the charge will not fire so quickly. If the scavenging is done by means of compressed air from the crank chamber it will be necessary to have an outside flywheel in order to make the volume of the crank chamber small.

Control and Brakes.

Although the control has been greatly improved by bringing the levers to the handle bars, the details have in many cases been badly carried out. The levers are often stiff, due to the friction which is required to keep the take off spring from pulling back the lever, and a much better way is to have a spring at both ends of the Bowden wire. Again, a large movement of the lever should only move the throttle a small amount when it is nearly closed, and *vice versa* when it is almost fully opened. This is quite easy to attain, but is almost invariably neglected. The levers should be fitted to a cross tube which connects the two handle bars together just in front of the grips, so that they move backwards to retard the machine; this is rather important, as it acts as a safety device to prevent the machine from running away when starting, but thumb slides are much superior to levers if they are fitted with springs at both ends of the wire, and some means are taken to prevent water getting into the slide. The thumb slides with the guides inside the handle bar are very neat, and do not dirty the fingers, but they would be improved if they were made of gun-metal, or if they had a guard over the slot to keep out the water.

The throttle and air or variable jet levers should be worked from the right hand bar, as the right hand is the first to be pulled back if the machine runs away while starting : the spark levers should be on the left. The exhaust valve lifter should have a big range, and the lever should have a ratchet catch to hold the valves open, similar to Bowden's No. 12, and should be fitted on the opposite handle bar to the throttle, the left being usually the most convenient with most starting devices. The clutch, if fitted, should be worked both by the right hand and the left foot, as this is a convenience when mounting, while the hand lever should have a ratchet similar to the exhaust valve lifter. Inverted brake levers are theoretically the best, as the strongest fingers are at the end of the levers, they are neater, but they are inclined to catch one's coat when mounting or running alongside. Probably the exhaust valve lifter and the clutch lever might be made to protect them. The front brake, which should be worked from the right bar, is more satisfactory if it is similar to the back brake working on a dummy belt rim. The dummy belt rims, both back and front, should not be too big, 16 in. diameter and 1½ in. in width, being quite big enough, as then it is still easy to take off the tyres. Perhaps it is too much to ask for two back brakes, but if two are fitted, one should be worked by the right foot and the other by the left hand, while both should work on the belt rim or a dummy belt rim.

Two hand brakes are advisable, and are very useful for bringing the machine down an incline out of the motor house, or if the machine stops on a very steep hill, but they are absolutely invaluable when it is necessary to stop quickly to save an accident, for it is possible to dismount and still keep both brakes on.

All the brake blocks should be composed of rubber and canvas and not fibre, as the latter tends to glaze when used frequently. The former are more powerful, smoother in action and last a fair length of time if they are reasonably big. The foot lever should not be connected to the foot rest, but attached independently to the frame so that it is not damaged in a fall. All the Bowden wires should have double ended adjustable stops as well as the ordinary stops, or these latter should be twice as long as usual, while all these stops should have small wing nuts instead of hexagons.

Change speed and Transmission.

The change speed gear and the transmission govern the whole design of the machine, including the engine. For instance, if a belt drive is decided on there are difficulties to be overcome in fitting a four-cylinder engine. With engines as they are to-day an ideal transmission should contain a free engine, with an infinitely variable or a two or three speed gear. In any case the ratio of gearing should be adjustable and, in addition to the above, the transmission should be weather proof, silent and capable of absorbing a considerable proportion of the shocks which are unavoidable owing to the necessity for keeping the flywheels light. At the same time the whole com-

bination should be efficient, simple and cheap. However good a certain combination might be, it would not be commercially practical unless it was reasonably cheap to manufacture. The following is a list of different ways of attacking the problem :—

1. An ordinary belt drive.
   A. with clutch in back wheel.
   B. with clutch on engine shaft.
   C. with change speed gear on engine shaft.
   D. with change speed in back wheel.
   E. with expanding pulley on engine shaft.
2. Chain drive.
   A. with two chains to countershaft.
   B. with change speed gear on countershaft.
3. Belt and chain drive.
   A. with two chains to countershaft. and belt to back wheel.
   B. with chain to change speed gear on countershaft and belt drive to back wheel.
   C. belt drive to countershaft, chain drive from countershaft to back wheel, expanding pulleys on engine shaft and countershaft.
4. Shaft drive with change speed gear on engine shaft and worm drive.

1. Belt Drive.—The clutch or the change speed gear is better in the back wheel rather than on the engine shaft, as it permits an adjustable pulley to be fitted on the engine shaft and, in the latter case, allows the belt to run at a higher speed. The expanding pulley on the engine shaft allows only a small range of gears, the belt running slightly out of line on some of the gears as it does when the adjustable pulley is moved out of its normal position, but with the rubber and canvas belt this does not seem to make much difference. None of the above systems are weather proof, but of late years this has been improved by a better system of mudguarding which will be dealt with under that heading.

2. Chain Drive.—Although clutches and shock absorbing devices are fitted to the chain driven machines, the drive is still rather harsh and sometimes noisy. The ratio of the gearing can be adjusted, but neither so quickly or conveniently as in the case of the belt drive. The transmission can be made weather proof with properly designed gearcases, which should have oil baths.

3. Chain and belt drive.—With this a very considerable adjustment of the ratio of the gearing is available, as both the countershaft pulley and the chain wheels can be altered. There is a considerable stress on the belt when it runs between the countershaft and the back wheel, and this combination is not weather proof.

3c. The combination of the belt drive to the countershaft, with two adjustable pullies, and a chain drive to the back wheel complies with all conditions for an ideal transmission, for it gives a free engine, an infinitely variable gear between say $2\frac{1}{2}$ to 1 and 10 to 1, with the belt always running in line. It can be made weather proof by enclosing the chain in an oil bath gear case and the belt with some form of cover, and it is silent, elastic, efficient, simple and cheap. The variations on this system are many and interesting, and a study of them would well repay any manufacturer. The system was originally suggested by Mr. Mervin O'Gorman many years ago.

4. The shaft drive has a fascination for many people who have had trouble with belts. Although it is quite weatherproof, its cost and the inability to adjust the ratio of the gearing is likely to stop this form of transmission from becoming general.

Frame, Springing and Wheels.

It has often been said that the English way of improving anything is by taking something off, whereas the Continental method is to add something. It is to be hoped in making the modern motor bicycle more comfortable, the former method will at any rate be borne in mind. To have joints and springs in the frame between the engine and the back wheel is only asking for trouble, and at the same time adding to the cost and weight of the machine. The obvious alternative method is to have a longer wheel base, larger tyres and bigger wheels. Contrary to the general opinion, on all but the best roads $2\frac{1}{2}$ in. tyres are faster than $2\frac{1}{4}$ in., so over average roads no time would be lost, and there would be less trouble. Owing to the influence of racing and the practical standardization of the present design of frame, it is doubtful if longer wheel bases will ever come into general use, for it is impossible to make a practical machine with a diamond frame of a longer wheel base than about 55 inches; anything longer is too whippy, and so skids. To make the machine less whippy it is necessary to carry a pair of tubes well braced together from the head to the back axle.

The author had some connection with the design of a machine in which a pair of tubes were taken from the top of the head, a pair from the bottom of the head and another pair from the bottom of the head past the engine, while all were joined together at the back axle. The machine had a wheel base of 60 ins., was very comfortable, and very rigid sideways. The tubes were not braced vertically, so there was a certain amount of give in that direction which no doubt accounted to a certain extent for the comfort. The general look of the machine was horrible, but as an experiment it showed what could be done.

There is no doubt that the general discomfort of some of the standard machines tends very much against the universal use of motor bicycles, so, if the trade is to be increased, something will have to be done. Large wheels are quite impossible without still longer wheel bases, as it is essential to keep the saddle low, and riders of six feet require an enormous amount of leg room—far more than is usually provided. On standard machines the rake of the head varies from 23° to 30°, the average being about 27°; this is rather small, and might be increased with advantage to 33° or 35°. At the same time a line through the head should hit the ground three inches in front of the point of contact of the wheel with the ground. The advantages of having a big rake are :—The steering is steadier, there is greater mechanical power, which more than compensates for the shorter handle bars, the frame can be made lower, and the longer steering head compensates for the greater stress put on the ball races.

Perhaps the best compromise without introducing any radical alteration to the general design of the frame, which would doubtless frighten the public, would be to take two tubes from the bottom of the head to the engine. If the two tubes were made D-shaped at the head end and placed back to back they could be fitted into the same lug. Two $\frac{3}{4}$ in. tubes go into a lug for a 1 in. tube and two $\frac{7}{8}$ in. into a $1\frac{1}{4}$ ins. lug. This makes a really neat and sound job if well done. The back forks might then be made further apart and well braced together. If this was done the wheel base might be increased by from five to eight inches, which would make the machine more comfortable.

If footboards are fitted holes big enough to get the heel of a boot into should be made at intervals. Aluminium boards with rubber studs are perhaps the best, and there should be a row of studs between each of the holes.

The pedalling gear as such is not necessary on a powerful motor bicycle, its only useful purpose being for starting the engine. The best form of starting is that with which it is possible to start the engine whether sitting on the saddle or not. If hand operated, the right hand should work it, as the left is engaged with the exhaust valve lifter which must be on the left handle bar, but, if foot operated, it should be capable of being worked with either foot—not both. No loose handles or straps should be fitted, and foot levers should not project dangerously. Pedals are rather in the way and interfere with the design of the machine, and again, a hand lever, if it only moves through a quadrant, must be arranged so that the machine can be wheeled backwards; which can be done by taking out the clutch.

A strap is an excellent way of starting the engine, as it revolves faster and faster as the strap unwinds, but a spring would have to be fitted to wind up the strap and an automatic apparatus to put it into gear with the engine shaft or two to one shaft—preferably the latter, as it should be on the right hand side.

The handles should be long and the full width in the front, the tubing as small as possible in diameter, and the gauge as light as is consistent with a reasonable amount of strength. If this is done most of the vibration is insulated from the hands, while the bars can be strengthened sideways if a tube is fixed close to the grips. The last few inches should turn a little outwards and sharply downwards, and it is an advantage if the handle bars are adjustable horizontally as well as vertically. Control wires look much neater if carried inside the tubes, and there should be no difficulty in this.

Mudguards.

Although there has been a great improvement in mudguards of late years, there is still room for further improvements. In designing these fittings the following requirements should be borne in mind. They should not increase the windage to any appreciable extent, that is the vertical faces exposed to the direction in which the machine moves, should be as small as possible, but, if the mudguard shrouds the wheel, there is bound to be a small drag on the wheels due to the friction of the air. A little clip might be arranged to keep the leather mudflap

from stirring up the dust in dry weather. Drip guards help to keep the mud off the pulleys, dummy pulleys, belt brakes and bearings, and if any mud does happen to get on any of these fittings the drip guards keep the mud from being thrown on the rider. They can be made flat, double concave, or concave towards the mudguard, but never convex on the side nearest the mudguard.

The front mudguard should extend over the front wheel, making about 50° with a vertical line through the axle. If the mudguard extends more than this, mud is blown back off the edges and, if less, it flies back on the machine. The shrouding should hide the tyre completely, except at the end of the extension : at the bottom the shrouding should be carried well forward, in fact to within a few inches of a vertical line through the axle, and then turned horizontally outwards five inches from the ground until the two horizontal flaps measure twelve inches across. With the addition of a small leather flap twelve inches wide and three inches deep no mud can get on the machine or rider, and the engine is not in any way screened. The author has made numerous experiments on mudguards, and he has found the above the only successful method for the front wheel, while a small amount of shrouding with two drip guards is all that is required on the back wheel. Each mudguard should be all in one piece, firmly fixed, but quickly detachable for tyre repairs and cleaning.

Stand.

The stand should be wide enough to hold the machine up anywhere even on the side of a highly cambered road; at the same time, when folded up it should be quite narrow and not stick out past the back mudguard. It should be easy and quick to operate.

Conclusion.

In conclusion the author wishes to point out that a number of most important points in the design of a motor bicycle have not been considered in these notes for a variety of reasons. They are merely intended to show that there are many directions in which motor cycle makers can improve their machines, thereby increasing the size of their market and strengthening their position nationally.

CORRESPONDENCE.

No responsibility is accepted by the Editor for the opinions of correspondents, and the right is reserved to omit any portion of a letter if it is considered desirable to do so. Letters to be inserted with a nom de plume must be accompanied by the name and address of the writer. If publication in the next issue is desired letters should reach the Editor in no case later than the 20th of the preceding month.

THE ORGANISATION AND EQUIPMENT OF A SMALL AUTOMOBILE FACTORY.

Sir,—In your issue of August, 1911, I have read an article by Mr. A. T. H. Davey under the above heading, and concur with what you say in your editorial note at the end of the article in question, viz., many of the suggestions made by your contributor are of a highly controversial nature. As a matter of fact, if you will pardon me saying so, I hardly think that any suggestions were made, and the article seemed to be of such a superficial and elementary nature that it is really hard to know from what point of view it should be criticised. What struck me most was Mr. Davey's diagrammatic sketch of the general arrangement of a factory considered capable of employing between five and six hundred workmen. In the first place, I might venture to state that I do not think there are many small automobile factories equipped with a foundry equal to doing all their work, including steel castings. The production of the latter is more or less a specialised job, and requires a highly trained staff to produce satisfactorily, and as a consequence entails a heavy expense due to maintenance. Such being the case one would be naturally led to suppose that it would be far preferable for the owners of a small factory, such as the one under consideration, to purchase all their steel castings from firms who specialise in this work, and, further, it would not be likely that this foundry could always be worked to its fullest capacity, unless it could be employed in producing castings for outside customers. Of course, all the cylinders and aluminium castings might be accomplished in the shop. Against this argument it might be advanced that the delay often experienced in obtaining steel castings would warrant the inclusion of a foundry in a small works, but, again, on the other hand, delays in connection with the delivery of orders for steel castings perhaps only occur now and again, but the expense of maintaining the foundry would always be present, and also there is a large percentage of scrap attendant in the production of steel castings which would be eliminated if purchased from outside sources. On the diagram a fettling section is shown attached to the foundry, but the sand blasting equipment as mentioned in the article has no space allotted to it on the former.

The smithy, it will be noticed, and I presume the case-hardening shop, are situated next to the machine shop. Such an arrangement is certainly not good practice, as it is obvious the gases and smoke will find their way into the latter, with consequent bad effect on the shafting and bright work of the machines. Further, the operations of the power hammers, as mentioned by your contributor, would be certain to be felt in the machine shop, and would play havoc with the numerous delicate machining operations.

A very noticeable feature in the sketch under consideration is the position of the rough stores, where rough castings, etc., are kept. In the article it is pointed out, and rightly so, that great care must be taken in arranging the various departments with the objects of facilitating production; yet here we have the rough stores situated as far away from the foundry and machine shop as they possibly could be, so all the castings coming from the foundry have to be carted right through the various departments to these stores, and when issued, brought back again over the same road to the machine shop. It appears to the writer that the right position for both the tool and general stores would be, if possible, between the assembling and machine shops.

Another prominent omission in the diagram, which is given, I gather, ostensibly to show the whole general arrangement of the automobile factory, is the absence of any indication as to where the receiving department is situated. Mention of such is made in the article, and one might reasonably expect to find this very important branch denoted on the sketch, but this is not so. Also the position of the engine and chassis testing departments is not shown. Neither is any mention or indication given as to where one might find the department where the change speed gears and differentials are run in before being fitted in the chassis.

The arrangement of the power plant also seems to call for comment. Although provision is made for overhauling the producers by having a third plant as a standby, no such arrangement is made whereby either one of the engines might be similarly treated.

Moreover, by reason of the manner in which the power plants are utilised, namely, having each unit coupled to a D.C. machine, giving off 220 volts, and the two generators connected in series which becomes necessary, as the shop is wired on a three-wire system intended to carry a 440 volt circuit, it is evident that the two generating sets must always be running. In the event of the failure of either one or the other of the units, the whole factory must cease operations until the defect is remedied, and, further, in the event of any single department, such, for instance, as the jig and tool room, which is specially mentioned by your contributor, being required to work late at night or overtime, it appears that the whole generating plant must be kept running in order to supply the necessary power to run a few machines in this department. This, at any rate, is the conclusion I have come to after reading Mr. Davey's article, though he might not have meant it to be taken in this way.

There are several other points I should like to raise, but shall not presume to waste any more of your valuable space, and I hope that others of your readers will give their opinions upon "The Organisation And Equipment Of A Small Automobile Factory."

C.N.Z.

TUNING UP A CAR FOR THE TRACK.

Sir,—In his interesting article in your last issue, Mr. R. W. A. Brewer, while giving some useful hints that were quite new to me, omits to mention one thing of the very greatest importance in high speed work on Brooklands. This is the absolute necessity, not only of giving very careful attention to the springing, but of balancing the road wheels. If any of your readers care to take the trouble of making the experiment they will find a quite considerable weight will often need to be added to a wheel before it is even in rough static balance *when the tyre is on.* A wire wheel will probably be quite well balanced by itself; but the valve, the joint in the inner tube and variations in the homogenity of an ordinary cover may easily be great enough to set up a centrifugal unbalanced action of sufficient power to cause violent oscillations through the whole car. Cars which have been hard to hold on the track at high speeds, have been known to be cured completely by the adding of a pound or so of lead to the wheels, the usual method of attachment being to cut pieces of sheet lead about a couple of inches square, punch them through at the centre and fit a number under one or two security bolt wing nuts.

ANGLO-AMERICAN.

BRONZE STAMPINGS.

We have recently inspected some bronze stampings of French origin, which are being handled here by Messrs. Harris and Samuels. Such parts of control levers, pedals, radiator caps and numerous other small parts, can apparently be stamped with a finish almost as good as that obtainable with a die casting, the surface being clean and bright. It is claimed that the metal employed consolidates in the stamping process, and the following is a comparative table of strengths supplied by the makers :—

BRONZE.			
NON-STAMPED.		STAMPED.	
Ultimate Strength.	Elastic Limit.	Ultimate Strength	Elastic Limit.
59 Kilos.	22 Kilos.	62 Kilos.	23 Kilos.
60 ,,	23 ,,	67 ,,	25 ,,
62 ,,	25 ,,	72 ,,	27 ,,
65 ,,	26 ,,	75 ,,	30 ,,
70 ,,	29 ,,	80 ,,	35 ,,
BRASS.			
40 Kilos.	12 Kilos.	44 Kilos.	15 Kilos.
42 ,,	13 ,,	48 ,,	18 ,,
44 ,,	18 ,,	50 ,,	21 ,,
48 ,,	22 ,,	53 ,,	24 ,,
58 ,,	27 ,,	65 ,,	28 ,,

GEAR MANUFACTURE.

Some notes on the equipment of a specialising factory.

THE biggest problem which confronts the designer of the present day automobile is the necessity for obtaining the maximum silence from a naturally noisy spur gear. It is therefore extremely interesting to study some of the methods and machines which are used by a firm who have devoted practically the whole of their attention to the scientific and accurate manufacture of such gearing. In considering the production of such gears it must be remembered that a firm producing gear wheels on a competitive basis is not in exactly the same position as a firm whose machines are turning out gears solely for their own consumption, because, in the event of trouble due to noise or machining defects, the latter are able to cover up their troubles easily and to experiment in the quiet of their own laboratories. Now should such a firm obtain gear wheels from outside, it is at once expected that the supplier shall provide gears as near perfection in every way as can possibly be manufactured, consequently any defect is the cause of considerable trouble. Accordingly the supplier is forced to watch every operation with extreme care and to use every device capable of improving his output.

The firm selected for the purposes of this description is that of Messrs. E. J. Wrigley and Co., who are now turning out a great quantity of gears, ranging from large motor omnibus worm gears down to the small gearing used in connection with carrier tricycles. In order that the least may be left to the actual individual responsible for each machine, there is a card system on which each separate order is entered, and a card issued to the shop giving full particulars of the gear pitch, number of teeth, feed and speed, together with all setting details necessary for the machine on which the order will be executed. Thus as little as possible is likely to go amiss when the machine hand begins his work. For the most part the actual gear-cutting plant, as distinct from the lathes used in the first processes on a gear blank, consists of Reinecker full automatic machines supplemented by a considerable number of smaller semi-automatics made by the same firm.

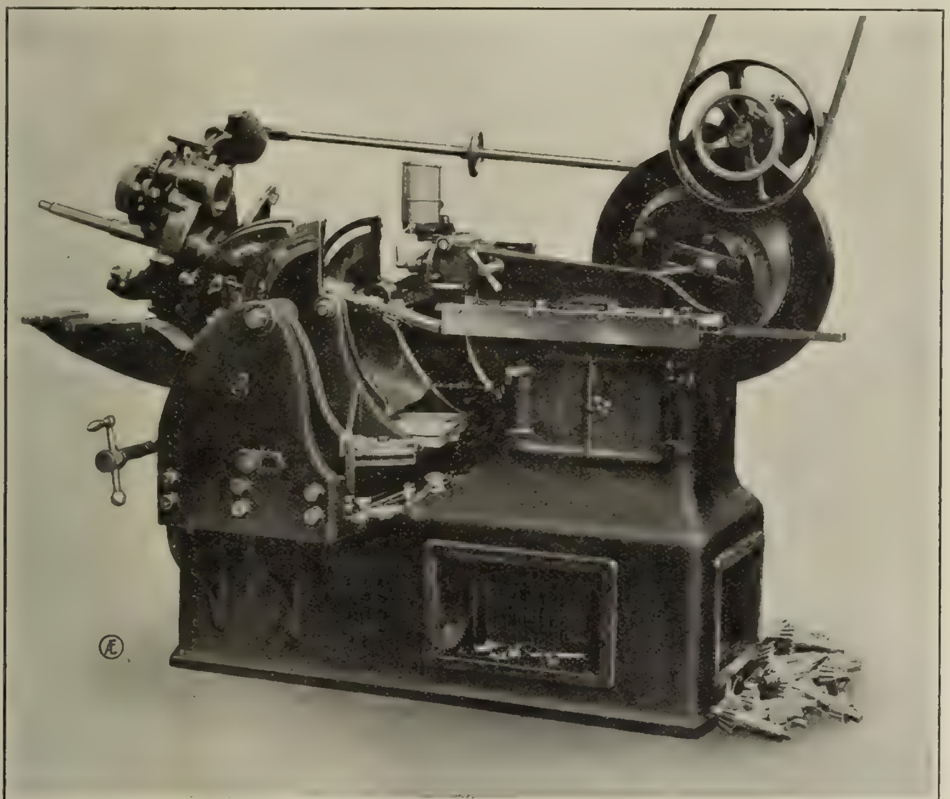


Fig. I. Reinecker Full Automatic Bevel Gear Cutter.

One of the bigger Reinecker full automatics is to be seen in Fig. I., which gives a view from the operating side. This type of machine, now so much used, is well known to almost everyone who is connected with the gear cutting trade, but a short description may be excused. The machine is of the tooth generating type, a crank motion for the oscillatory cutter being visible on the right, with the slide by which its forward motion can be limited. Around the crank can be seen the slot cam gear with its rotary traveller which operates the cutter disengaging gear. A pinion is seen on the spindle attached to the rolling carriage by which the tooth is formed, and above the spindle is the spur gearing and ratchet which governs the carriage motion. By means of the semi-rotational semi-transverse motion of the spindle any form of gear tooth can be generated, and there is little doubt that such a method produces accurate and well-finished teeth, though care must be taken to observe that the blank is true as to outside diameter, for setting purposes. Racks for the various gearing used for adjustment are seen in the base of the machine. The only part of the whole machine likely to cause trouble is the slot cam gear already mentioned, but this is more satisfactory in use than might at first be supposed.

All the smaller semi-automatics are on the same principle, differing only in minor details, such as the spindle carriage of some being rocked on a wire, which may be likely to give some trouble, while others are more efficiently provided with a broad steel ribbon for the same purpose.

Another type of machine used by the firm with complete success is the Gleason full automatic—(Fig. II.)—another of the

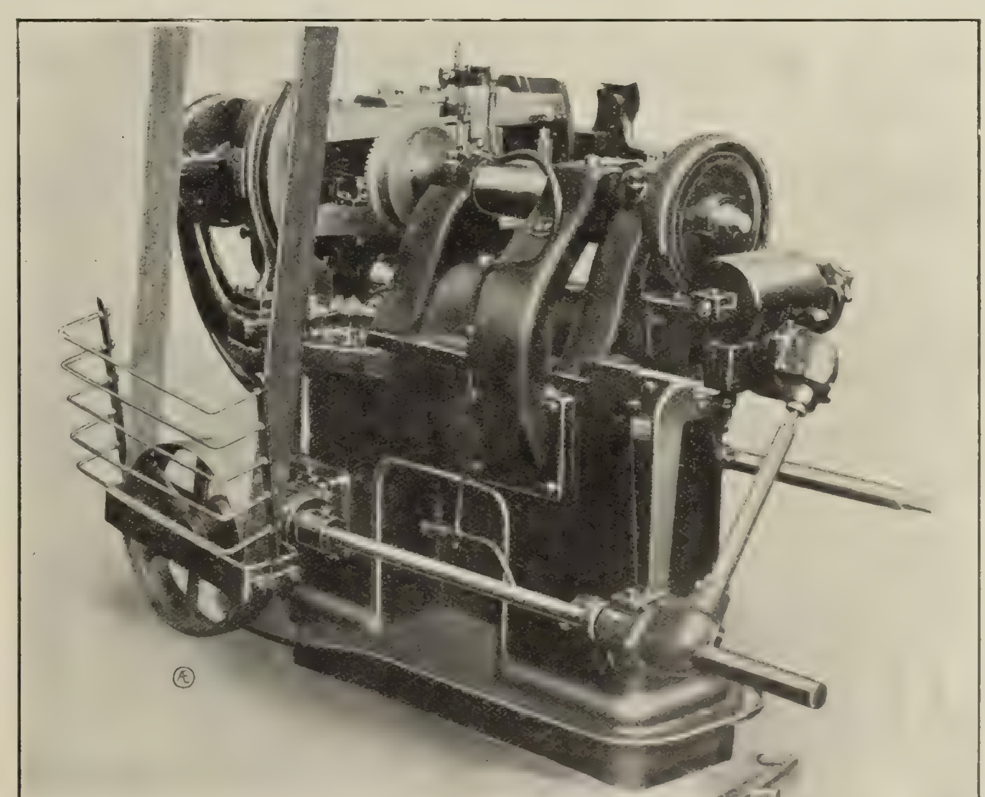


Fig. II. Gleason Automatic Bevel Gear Cutter.

tooth generating machines, differing considerably however, from the Reinecker already described. In this machine the whole principle of operation is the rocking of the straight cutting tools and the gear wheel, exactly as if they were actually in mesh. Each cutting tool has a reciprocating motion in opposite directions, working on either side of the same tooth, the form of which is governed by a cam actuated arm operating the entire cutter carriage. In the illustration a bevel can be observed on the spindle, the tool

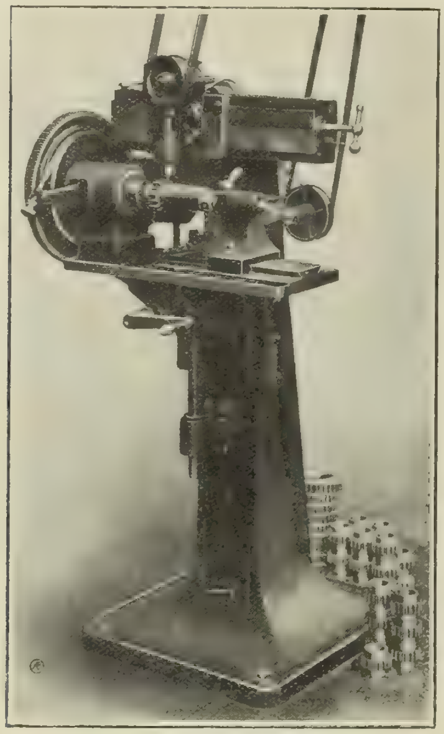


Fig. III. Holroyd Machine.

holders being immediately behind the vertical support used for the soapsud spray pipe*. Fig. III. shows a neat little Holroyd machine, used for machining the end of each tooth to allow easy engagement when one is slid into mesh with its fellow. This profiling machine has an actuated tool slide—the front view of the slide and cutter being clearly shown in the photograph. By noting the end of the cam actuated arm attached to the slide, the whole motion can be followed and the manner in which the small tool follows the contour of the tooth explained. Any size of gear wheel can be dealt with, as the work table is adjustable by the screw gear handle seen underneath the machine. On the floor are a number of pinions varying in size, all of which are dealt with on this machine.

Standing out among the modern gear cutters is an old Reinecker worm hobbing machine which is still able to give a satisfactory account of itself after some years of service. At the finish of the last operation connected with the generating of the gears they are delivered to the testing staff, whose duty it is to obtain the quietest possible running and to see that a proper contact is made between wheel and pinion. For enabling this work to be accomplished in a more accurate manner than hitherto, and also with a view to expediting the whole operation, it was found necessary to construct the two special testing machines shown in Fig. IV. Here a bevel and a worm wheel, with its accompanying worm, will be observed, set up for inspection and test. The bevel testing machine consists of a bed appertaining to an ordinary screw cutting lathe, but having both head and tail stocks removed, and a special carriage and head substituted. A cast iron pillar set up at the end of the slide replaces the head stock. On this pillar is fixed a sliding spindle holder, adjusted on grooves by means of the hand wheel and screw thread seen at the top of the upright. A heavy, ribbed iron casting supplies a bearing for the spindle, which is driven by a pulley and belt from the line shafting, which has a reversing pulley provided for additional tests. At the end of the spindle there is a Morse taper suitable for pinions, which may be solid with their shafts, while a free shaft has to be fitted therein to hold loose pinions. On the lathe bed, traversed by the left-hand wheel, is a carriage so constructed that a bevel can be clamped therein and raised to any given centre by means of packing pieces. Thrust on the carriage during the test is taken by the flat iron collar seen projecting towards the onlooker. In testing, a pinion is fitted to the upright and a bevel to the carriage; the pinion is then lowered into mesh and allowed to enter a predetermined distance as shown by the scale, another scale indicating the amount the carriage has traversed. When meshed, the gears are run for a considerable time, any alterations necessary to enable a better contact to be made being carried out with oilstones. As can be imagined, some extremely interesting experiments can be conducted on the subject of meshing depth in relation to noise. On the right in Fig. IV. is the worm testing machine, which is operated in much the same manner, save that adjustment for height is effected by means of suitable size packings inserted below the worm wheel and not by a screw, while the worm wheel is meshed by the traverse wheel shown. The worm seen in the illustration gives some idea of the size of work which frequently comes into the firm's hands and which requires an inordinate amount of careful handling.

In regard to the attainment of silence, it has been found that nearly everything depends on the amount of time which can be allowed for the production of the gears, since extreme care has to be used where silence is an essential point in the order.

Apart from the gear cutting plant there were two small details in the equipment which were of a novel and interesting nature. One was a brake fitted to the heavy chuck of the lathes, which is an undoubted time saver and considerably in advance of the ordinary hand stopping, both from the personal and the machine point of view. The other was a small machine used for milling cams. With this a large size master cam, previously set at the required position, automatically controlled the travel of a carriage on which was the rotating cutter, the cam blank being slowly rotated at the same time. The machine was extremely neat albeit not easy to set up, yet when once adjusted would perform with great reliability.

The most striking feature of a gear cutting plant such as the one described is the complete absence of evidence betraying the complicated nature of the gear cutting problem and the apparent simplicity both of the machine shop arrangement and of the operating machines themselves, while no difficulty has been encountered during the search for suitable operators.

The whole problem of quiet spur gearing seems to turn to a great extent on the care taken while cutting gear teeth, and the nature and accuracy of the cutter, rather than on some form of automatic process or specially devised gear forming machine.

Tooth generating machines as distinct from the milling type are used almost completely, the blank being gashed roughly by a revolving cutter previous to the formnig process which is responsible for the accuracy of the finished wheel, but beyond this no especial form of machine seems in itself to be a prominent advantage.

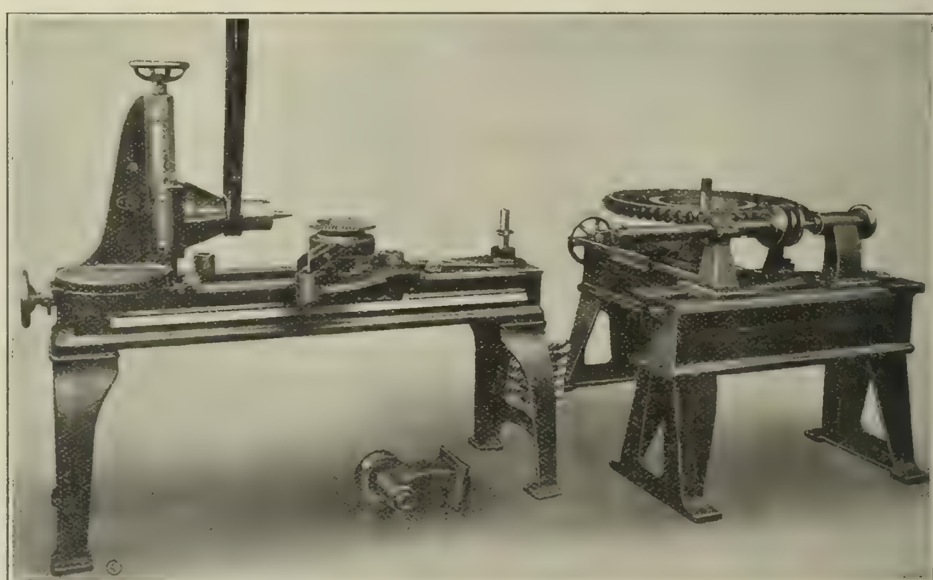


Fig. IV. Bevel and Worm Testing Machines.

In the firm's experience more has been done towards obtaining quiet gears by extreme care, and by the close supervision of competent men, than by special jigs or other aids to manufacture. It must therefore be for the purchaser to decide whether the quietness of gears is worth while, or whether to try and strike a medium level between quietness and cost.

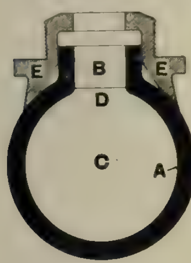
In the Automobile Engineer for December, 1910, reference can be made to a full and particular description of the Gleason machine.

RECENT AUTOMOBILE PATENTS.

By Eric W. Walford, F.C.I.P.A.

To prevent burst Water Jackets.

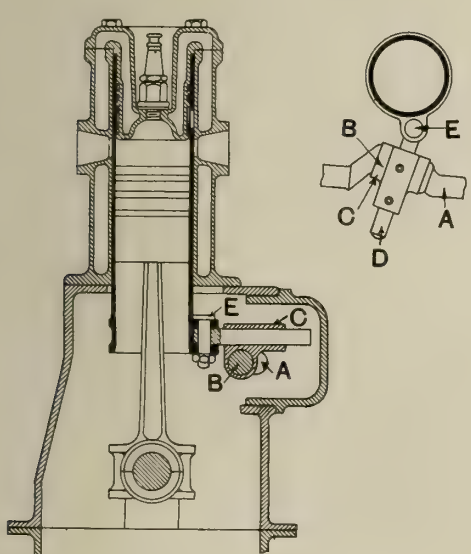
This little device is intended to be fitted to water cooling systems in order to effect automatically the drawing off of the water when the temperature approximates to freezing point. It operates by the employment of a small bottle A, containing a liquid adapted to freeze at a temperature slightly higher than that of the freezing point of water. When the liquid contained in this bottle freezes it expands, setting up a leak and so causing the water to run out in the case illustrated. The bottle neck is closed by a plug B, and between this and the surface of the distilled water C is arranged a layer of grease D. The bottle neck is lightly soldered at E into a plug which screws into the bottom of a radiator. As the liquid C expands it exerts a pressure on the immovable plug B, with the consequence that the bottle breaks away from its soldered joint E, and falls, together with the plug B. The water then runs out of the radiator, and any damage from frost is prevented.



No. 19560/10. C. L. Sumpster and H. K. Foster.

An Oscillating Sleeve Valve.

In the sleeve valve engine, which is being tried by the Argyll Co., the valve receives an oscillatory as well as a reciprocatory movement, these motions being combined so that a point on the sleeve traces practically an elliptical or circular path, being compounded of a vertical movement, and a horizontal one.

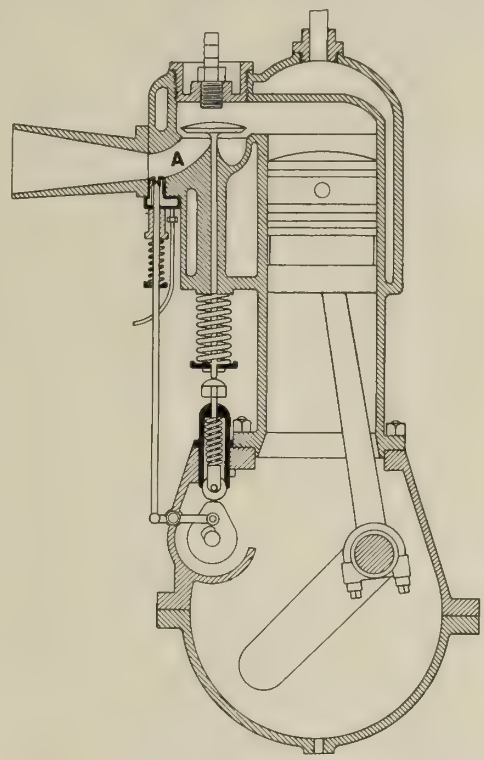


To provide this movement the valve actuating shaft A is provided with an angular crank pin B, on which is mounted a journal block C, whilst through this slides a spindle D, pivotally connected to the sleeve E. This angular crank actuates the sleeve, and it combines with the vertical movement the necessary oscillating one, which gives the sleeve the desired travel.

No. 24538/10. P. Burt and Argylls, Limited.

A Single-Valve Engine.

The engine is provided with one valve which does duty for both inlet and exhaust, and a single gas passage A, along

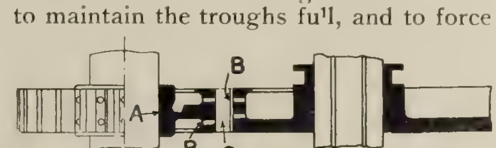


which the exhaust gases are ejected and through which air is drawn in. It will be understood that the valve remains open continuously for the exhaust and inlet strokes, and that after the exhaust gases are ejected, suction is set up along the passage A, drawing in air, and possibly some exhaust gas, so filling the cylinder. During the passage of the air along the passage A it is carburetted by means of a fuel injector actuated by a needle valve controlled from the cam shaft. By the employment of a single valve to do duty on the inlet and exhaust strokes this engine resembles one or two which have been lately produced for aeronautical work. It also resembles the valve system used on the early horizontal-engined Lanchester cars, with the exception that what corresponds to the passage A was provided with a two-way disc valve, so that the inlet and exhaust gases were separated. There was, however, only one engine valve.

No. 15662/10. P. L. F. Fouchet.

Gear Wheel Lubrication.

The gear wheels are formed with central webs A, on each side of which troughs are left, and from these troughs are drilled passages B, which lead to the spaces between the gear wheel teeth shown at C. The gear wheels are run in an oil bath and centrifugal action tends to maintain the troughs full, and to force

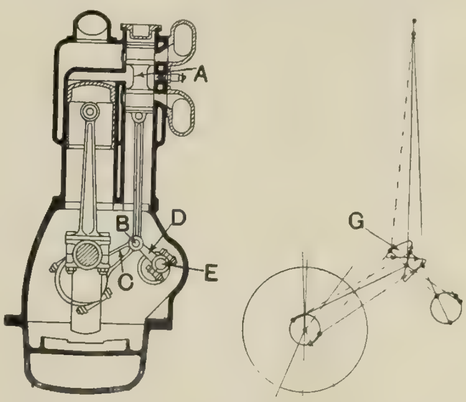


the lubricant through the passage B, and lubricate the pressure surfaces of the gear wheel teeth.

No. 19780/10. F. E. Beaumont and C. F. B. Marshall.

A Piston-Valve Engine.

The engine is provided with a single dumb-bell piston A, adapted to put the cylinder into communication either with the inlet or exhaust passage according to its vertical position. The invention consists in the method of operation of the piston, and it will be seen that the piston rod is connected at B with an eccentric rod C, actuated by an eccentric on the engine crank shaft, whilst the valve piston rod is also connected by a link D to the crank pin E, on the half-time shaft. The links C and D therefore act at times as a toggle, with the consequence that the valve A receives a very rapid movement when the ends of the toggle links, D and C, move towards one another, whilst periods of dwell can be obtained when the ends of the toggle links move in the same direction. The diagrammatic drawing shows at G the path of the centre of the pin B, and it will be seen to be roughly 8-shaped. When the pin B is moving along the vertical portion of the 8, the valve is moving rapidly from one open position to the other. When it is moving in an approximately horizontal

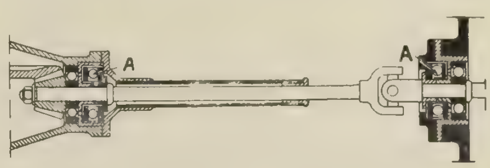


direction it will be understood that the piston A hardly receives any movement at all, this being the case during most of the compression and exhaust strokes.

No. 16783/10. Wolseley Tool and Motor Car Co., Ltd., A. A. Remington and A. J. Rowledge.

An Axle System.

To simplify the connection between the axle and the frame, and to abolish anchored torque tubes and radius rods and leave the springs free at both ends



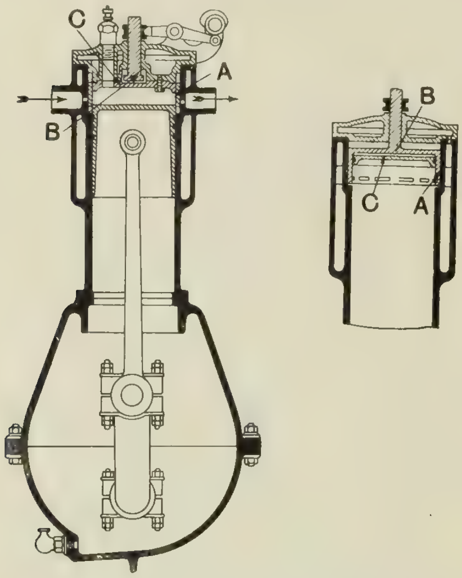
this construction is suggested. The torque is transmitted through the propeller shaft tube and the axle thrust due to the drive and braking effect are passed through the propeller shaft. Each end of the propeller shaft is provided with a thrust bearing A, so housed that the shaft can transmit thrust in either direction to the supporting parts of the ball bearings, that is to say, the axle casing on the one hand and the gear box on the other. It will be seen that the universal joint is constructed so as not to provide any sliding motion, which, of course, would interfere with the transmission of

driving thrust and retarding effect through the propeller shaft.

No. 2905/1911. A. Joubert and L. Ravel.

A Slide Valve Engine.

This engine is provided with a short sleeve valve A, which is provided with ports the whole way round, adapted to communicate with the inlet and exhaust passages. The inlet and exhaust passages are made at different levels, although this is not clear in the drawings. The valve has, therefore, to be

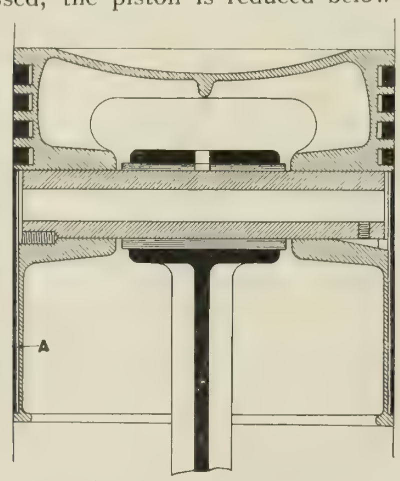


raised to open up the one passage and lowered to put the cylinder into communication with the other. The cylinder is bored out larger at the top for the valve, and the piston is adapted to project right up into the sleeve valve and cover the port openings in it when at the top of the stroke. The valve is provided with a cross member or bridge, B, which is attached to a rod which in turn receives its movement from a rocking lever and overhead cam shaft. The bridge piece B lies in a recess in the jacketed cylinder head, and is cut off from the cylinder by a cover plate C, which lies inside the sleeve valve and is bolted to the cylinder cover.

No. 21831/10. F. Forest.

A New Piston.

To prevent the noise due to the changing of bearing side between the piston and the cylinder as the dead centres are passed, the piston is reduced below the

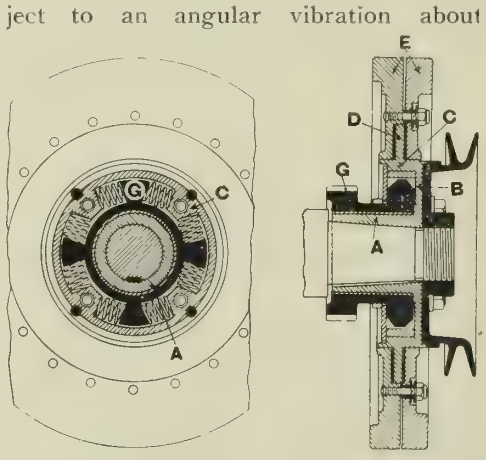


gudgeon pin, and in the space between this reduced part and the cylinder wall lies a sleeve A, which is split vertically and possesses some natural spring which keeps it in contact with the cylinder walls. A certain amount of oil lies between the ring and the piston, so as to cushion the movement of the piston from one side to the other. The bottom edge of the ring A is tapered off, so that the tendency of the ring is to scrape the oil into the space between the piston and the ring on each down stroke. In the event of seizure of the piston, it will be understood that the ring A would seize and probably protect the actual piston from receiving damage.

No. 17973/10. C. Y. Kngiht.

To Reduce Engine Vibration.

This invention relates to the mounting of combined gear wheels and flywheels, both resiliently and frictionally, on the shaft, so that vibrations are damped out and not transmitted to or from the half-speed gearing. The crank shaft has keyed to it a sleeve A, which is flanged radially and concentrically as shown at DCD. The part D is provided with anti-friction surfaces and is gripped by the twin flywheels E, which are bolted together by the bolts illustrated. These bolts act through springs, so that the gripping effect may be easily varied by adusting the bolts. The fly wheels are therefore free to slide relatively to the flange D fixed to the crank shaft, as soon as the angular axial acceleration of the crank shaft about its axis rises above a predetermined limit. Thus the friction between the flywheel faces and the flange D absorbs some of the energy of the resulting vibrations, and their amplitude and violence are reduced. The specification states that the limit referred to is not reached by the angular acceleration, which is due to normal load, but is only arrived at when the crank shaft is subject to an angular vibration about

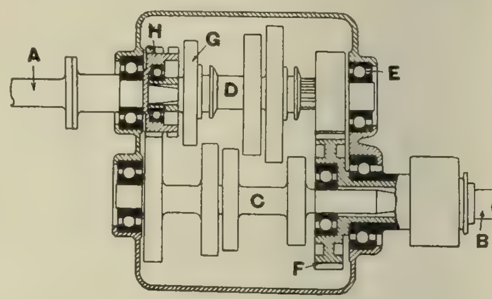


its axis. The sleeve part C has keyed to it a ring provided with internal projections between which and the external projections on the gear wheel G lie springs. The springs are so stiff as not to yield under normal load, but will allow the crank shaft to vibrate relatively about its axis within certain limits without transmitting these vibrations to the gear wheel.

No. 9262/11. F. H. Royce and Rolls Royce, Limited.

A Gear Box.

This gear box gives a constant reduction of speed, and also a second reduction which is dependant on the particular gear wheels in mesh. The driving shaft is shown at A and the driven at B. The driving shaft gears by constantly meshed gear wheels with the gear wheels fixed to the lay shaft C. There is a second lay shaft D, and on it are mounted, to slide, various gear wheels, and running free on this shaft is a wheel E, which constantly meshes with the wheel F on the driven shaft. The highest gear is obtained when the gear wheel G is moved to the left to be clutched to part of the gear wheel H. The drive thus passes from the shaft A direct to the shaft D, and therefrom to the driven shaft B, through the constantly meshed wheels E and F. It will be seen, even on the "direct" drive, that there is a constant reduction due to the relative dimensions of the



wheels E and F. On the other gears the wheels on the shaft D are moved to mesh with those on the shaft C, and the power passes through the wheel H, and that on the shaft C, with which it is constantly in mesh, and from the shaft C to the lay shaft D, through any one pair of gear wheels selected, and from the shaft D to the driven shaft B through the constantly meshed wheels E and F.

No. 15989/11. Daimler Motoren Gesellschaft.

THE INSTITUTION OF AUTOMOBILE ENGINEERS.

The first meeting of the session of the Institution of Automobile Engineers will be held at the Institution of Mechanical Engineers, Storey's Gate, St. James's Park, S.W., on October 11th, at 8 p.m., when the newly-elected President, Mr. L. A. Legros, will give his presidential address. An excellent programme of papers has been arranged for the session, and these will be read on November 8th and December 13th, 1911; January 10th, February 14th, March 13th, April 10th, and May 8th, 1912.

'TILLING-STEVENS' PETROL ELECTRIC OMNIBUSES.

As a result of the successful running of the new "Tilling-Stevens" petrol electric omnibuses in London service, Messrs. Tilling have decided to adopt that type for all their future omnibuses, and are constructing a fleet to replace their present horse-drawn vehicles. Messrs Tilling have designed the chassis and body, which are being built at their works, but the electrical transmission is supplied by Messrs. W. A. Stevens, Ltd. An agreement has been entered into between the two companies, in which Messrs. Tilling have secured the sole rights of use and sale of the "Stevens" patent electrical transmission for use in omnibuses and mail vans in the Metropolitan police area, Messrs. Stevens to have the sole rights of manufacture and sale of the complete "Tilling-Stevens" petrol electric chassis outside the Metropolitan police area, and also inside the Metropolitan area for vehicles other than omnibuses or mail vans.

FORTHCOMING AMERICAN SHOWS.

Jan. 1-5.—New York City, Grand Central Palace, Annual Show, Automobile Manufacturers' Association of America.

Jan. 6-13.—New York City, Madison Square Garden, Twelfth Annual Show, Pleasure Car Division, Automobile Board of Trade.

Jan. 10-17.—New York City, Madison Square Garden, Annual Show, Motor and Accessories Manufacturers.

Jan. 10-17.—New York City, Grand Central Palace, Twelfth Annual Show, National Association of Automobile Manufacturers.

Jan. 15-20.—New York City, Madison Square Garden, Twelfth Annual Show, Commercial Division, Automobile Board of Trade.

Jan. 27.-Feb. 10.—Chicago Coliseum, Eleventh Annual Automobile Show, under the auspices of the National Association of Automobile Manufacturers.

PISTON DISPLACEMENTS in Cubic In

STROK

Bore in Inches.	3	3 1/8	3 1/4	3 3/8	3 1/2	3 5/8	3 3/4	3 7/8	4	4 1/8	4 1/4	4 3/8	4 1/2	4 5/8	4 3/4	4 7/8	5	5 1/8	5 1/4	
3	84.8	88.4	91.9	95.4	99.0	102.5	106.0	109.6	113.1	116.6	120.2	123.7	127.2	130.8	134.3	137.8	141.4	144.9	148.4	15[illegible]
3 1/16	88.4	92.1	95.8	99.4	103.1	106.8	110.5	114.2	117.9	121.5	125.2	128.9	132.6	136.3	140.0	143.6	147.3	151.0	154.7	15[illegible]
3 1/8	92.0	95.9	99.7	103.5	107.4	111.2	115.0	118.9	122.7	126.6	130.4	134.2	138.1	141.9	145.7	149.6	153.4	157.2	161.1	16[illegible]
3 3/16	95.8	99.7	103.7	107.7	111.7	115.7	119.7	123.7	127.7	131.7	135.7	139.6	143.6	147.6	151.6	155.6	159.6	163.6	167.6	17[illegible]
3 1/4	99.5	103.7	107.8	112.0	116.1	120.3	124.4	128.6	132.7	136.9	141.0	145.2	149.3	153.5	157.6	161.8	165.9	170.1	174.2	17[illegible]
3 5/16	103.4	107.7	112.0	116.3	120.7	125.0	129.3	133.6	137.9	142.2	146.5	150.8	155.1	159.4	163.7	168.0	172.4	176.7	181.0	18[illegible]
3 3/8	107.4	111.8	116.3	120.8	125.2	129.7	134.2	138.7	143.1	147.6	152.1	156.6	161.0	165.5	170.0	174.5	178.9	183.4	187.9	19[illegible]
3 7/16	111.4	116.0	120.6	125.3	129.9	134.6	139.2	143.8	148.5	153.1	157.8	162.4	167.1	171.7	176.3	181.0	185.6	190.3	194.9	19[illegible]
3 1/2	115.5	120.3	125.1	129.9	134.7	139.5	144.3	149.1	153.9	158.7	163.6	168.4	173.2	178.0	182.8	187.6	192.4	197.2	202.0	20[illegible]
3 9/16	119.6	124.6	129.6	134.6	139.5	144.5	149.5	154.5	159.5	164.5	169.5	174.4	179.4	184.4	189.4	194.4	199.4	204.3	209.3	21[illegible]
3 5/8	123.8	129.0	134.2	139.3	144.5	149.6	154.8	160.0	165.1	170.3	175.5	180.6	185.8	190.9	196.1	201.3	206.4	211.6	216.7	22[illegible]
3 11/16	128.2	133.5	138.8	144.2	149.5	154.9	160.2	165.5	170.9	176.2	181.6	186.9	192.2	197.6	202.9	208.3	213.6	218.9	224.3	22[illegible]
3 3/4	132.5	138.1	143.6	149.1	154.6	160.1	165.7	171.2	176.7	182.2	187.8	193.3	198.8	204.3	209.8	215.4	220.9	226.4	231.9	23[illegible]
3 13/16	137.0	142.7	148.4	154.1	159.8	165.5	171.2	176.9	182.7	188.4	194.1	199.8	205.5	211.2	216.9	222.6	228.3	234.0	239.7	24[illegible]
3 7/8	141.5	147.4	153.3	159.2	165.1	171.0	176.9	182.8	188.7	194.6	200.5	206.4	212.3	218.2	224.0	230.0	235.9	241.8	247.7	25[illegible]
3 15/16	146.1	152.2	158.3	164.4	170.5	176.6	182.7	188.7	194.8	200.9	207.0	213.1	219.2	225.3	231.4	237.4	243.5	249.6	255.7	26[illegible]
4	150.8	157.1	163.4	169.6	175.9	182.2	188.5	194.8	201.1	207.3	213.6	219.9	226.2	232.5	238.8	245.0	251.3	257.6	263.9	27[illegible]
4 1/16	155.5	162.0	168.5	175.0	181.5	188.0	194.4	200.9	207.4	213.9	220.4	226.8	233.3	239.8	246.3	252.8	259.2	265.7	272.2	278[illegible]
4 1/8	160.4	167.1	173.7	180.4	187.1	193.8	200.5	207.1	213.8	220.5	227.2	233.9	240.6	247.2	253.9	260.6	267.3	274.0	280.6	287[illegible]
4 3/16	165.3	172.2	179.0	185.9	192.8	199.7	206.6	213.5	220.4	227.2	234.1	241.0	247.9	254.8	261.7	268.6	275.4	282.3	289.2	29[illegible]
4 1/4	170.2	177.3	184.4	191.5	198.6	205.7	212.8	219.9	227.0	234.1	241.2	248.3	255.4	262.4	269.5	276.6	283.7	290.8	297.9	305[illegible]
4 5/16	175.3	182.6	189.9	197.2	204.5	211.8	219.1	226.4	233.7	241.0	248.3	255.6	262.9	270.2	277.5	284.8	292.1	299.4	306.7	314[illegible]
4 3/8	180.4	187.9	195.4	202.9	210.5	218.0	225.5	233.0	240.5	248.0	255.6	263.1	270.6	278.1	285.6	293.1	300.7	308.2	315.7	323[illegible]
4 7/16	185.6	193.3	201.1	208.8	216.5	224.3	232.0	239.7	247.4	255.2	262.9	270.6	278.4	286.1	293.8	301.6	309.3	317.0	324.8	332[illegible]
4 1/2	190.9	198.8	206.8	214.7	222.7	230.6	238.6	246.5	254.5	262.4	270.4	278.3	286.3	294.2	302.2	310.1	318.1	326.0	334.0	341[illegible]
4 9/16	196.2	204.4	212.5	220.7	228.9	237.1	245.2	253.4	261.6	269.8	277.9	286.1	294.3	302.5	310.6	318.8	327.0	335.2	343.3	351[illegible]
4 5/8	201.6	210.0	218.4	226.8	235.2	243.6	252.0	260.4	268.8	277.2	285.6	294.0	302.4	310.8	319.2	327.6	336.0	344.4	352.8	361[illegible]
4 11/16	207.1	215.7	224.3	233.0	241.6	250.2	258.9	267.5	276.1	284.7	293.4	302.0	310.6	319.3	327.9	336.5	345.1	353.8	362.4	371[illegible]
4 3/4	212.6	221.5	230.4	239.2	248.1	256.9	265.8	274.7	283.5	292.4	301.2	310.1	319.0	327.8	336.7	345.6	354.4	363.3	372.1	381[illegible]
4 13/16	218.3	227.4	236.5	245.6	254.7	263.8	272.8	281.9	291.0	300.1	309.2	318.3	327.4	336.5	345.6	354.7	363.8	372.9	382.0	391[illegible]
4 7/8	224.0	233.3	242.7	252.0	261.3	270.6	280.0	289.3	298.6	308.0	317.3	326.6	336.0	345.3	354.6	364.0	373.3	382.6	392.0	401[illegible]
4 15/16	229.8	239.3	248.9	258.5	268.1	277.6	287.2	296.8	306.4	315.9	325.5	335.1	344.6	354.2	363.8	373.4	382.9	392.5	402.1	411[illegible]
5	235.6	245.4	255.3	265.1	274.9	284.7	294.5	304.3	314.2	324.0	333.8	343.6	353.4	363.2	373.1	382.9	392.7	402.5	412.3	422.[illegible]
5 1/16	241.5	251.6	261.7	271.7	281.8	291.9	301.9	312.0	322.1	332.1	342.2	352.3	362.3	372.4	382.4	392.5	402.6	412.6	422.7	432.[illegible]
5 1/8	247.5	257.9	268.2	278.5	288.8	299.1	309.4	319.7	330.1	340.4	350.7	361.0	371.3	381.6	392.0	402.3	412.6	422.9	433.2	443.[illegible]
5 3/16	253.6	264.2	274.8	285.3	295.9	306.5	317.0	327.6	338.2	348.7	359.3	369.9	380.4	391.0	401.6	412.1	422.7	433.3	443.8	454.[illegible]
5 1/4	259.8	270.6	281.4	292.2	303.1	313.9	324.7	335.5	346.4	357.2	368.0	378.8	389.7	400.5	411.3	422.1	433.0	443.8	454.6	465.[illegible]
5 5/16	266.0	277.1	288.2	299.2	310.3	321.4	332.5	343.6	354.7	365.7	376.8	387.9	399.0	410.1	421.2	432.2	443.3	454.4	465.5	476.[illegible]
5 3/8	272.3	283.6	295.0	306.3	317.7	329.0	340.4	351.7	363.1	374.4	385.7	397.1	408.4	419.8	431.1	442.5	453.8	465.2	476.5	487.[illegible]
5 7/16	278.7	290.3	301.9	313.5	325.1	336.7	348.3	359.9	371.5	383.2	394.8	406.4	418.0	429.6	441.2	452.8	464.4	476.0	487.6	499.[illegible]
5 1/2	285.1	297.0	308.9	320.7	332.6	344.5	356.4	368.3	380.1	392.0	403.9	415.8	427.6	439.5	451.4	463.3	475.2	487.0	498.9	510.[illegible]
5 9/16	291.6	303.8	315.9	328.1	340.2	352.4	364.5	376.7	388.8	401.0	413.1	425.3	437.4	449.6	461.7	473.9	486.0	498.2	510.3	522.[illegible]
5 5/8	298.2	310.6	323.1	335.5	347.9	360.3	372.8	385.2	397.6	410.0	422.5	434.9	447.3	459.7	472.2	484.6	497.0	509.4	521.9	534.[illegible]
5 11/16	304.9	317.6	330.3	343.0	355.7	368.4	381.1	393.8	406.5	419.2	431.9	444.6	457.3	470.0	482.7	495.4	508.1	520.8	533.5	546.[illegible]
5 3/4	311.6	324.6	337.6	350.6	363.5	376.5	389.5	402.5	445.5	428.5	441.4	454.4	467.4	480.4	493.4	506.4	519.3	532.3	545.3	558.[illegible]

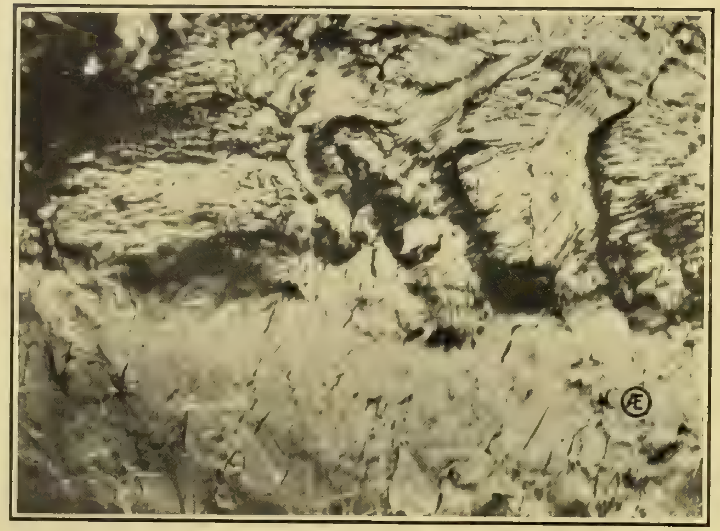
hes Calculated for Four-cylinder Engines.

INCHES. [LIMIT OF ERROR = 0.05 CU. IN.]

$5\frac{1}{2}$	$5\frac{5}{8}$	$5\frac{3}{4}$	$5\frac{7}{8}$	6	$6\frac{1}{8}$	$6\frac{1}{4}$	$6\frac{3}{8}$	$6\frac{1}{2}$	$6\frac{5}{8}$	$6\frac{3}{4}$	$6\frac{7}{8}$	7	$7\frac{1}{8}$	$7\frac{1}{4}$	$7\frac{3}{8}$	$7\frac{1}{2}$	$7\frac{5}{8}$	$7\frac{3}{4}$	$7\frac{7}{8}$	8
55.5	159.0	162.6	166.1	169.6	173.2	176.7	180.2	183.8	187.3	190.9	194.4	197.9	201.5	205.0	208.5	212.1	215.6	219.1	222.7	226.2
62.1	165.7	169.4	173.1	176.8	180.5	184.2	187.8	191.5	195.2	198.9	202.6	206.3	209.9	213.6	217.3	221.0	224.7	228.4	232.0	235.7
68.7	172.6	176.4	180.2	184.1	187.9	191.7	195.6	199.4	203.3	207.1	210.9	214.8	218.6	222.4	226.3	230.1	233.9	237.8	241.6	245.4
75.6	179.5	183.5	187.5	191.5	195.5	199.5	203.5	207.5	211.5	215.5	219.4	223.4	227.4	231.4	235.4	239.4	243.4	247.4	251.4	255.4
82.5	186.6	190.8	195.0	199.1	203.2	207.4	211.5	215.7	219.8	224.0	228.1	232.3	236.4	240.6	244.7	248.9	253.0	257.2	261.3	265.5
89.6	193.9	198.2	202.5	206.8	211.1	215.4	219.8	224.1	228.4	232.7	237.0	241.3	245.6	249.9	254.2	258.5	262.8	267.2	271.5	275.8
96.8	201.3	205.8	210.2	214.7	219.2	223.7	228.1	232.6	237.1	241.5	246.0	250.5	255.0	259.4	263.9	268.4	272.9	277.3	281.8	286.3
04.2	208.8	213.5	218.1	222.7	227.4	232.0	236.7	241.3	245.9	250.6	255.2	259.9	264.5	269.1	273.8	278.4	283.0	287.7	292.3	297.0
11.7	216.5	221.3	226.1	230.9	235.7	240.5	245.3	250.1	255.0	259.8	264.6	269.4	274.2	279.0	283.8	288.6	293.4	298.3	303.1	307.9
19.3	224.3	229.3	234.2	239.2	244.2	249.2	254.2	259.2	264.1	269.1	274.1	279.1	284.1	289.1	294.1	299.0	304.0	309.0	314.0	319.0
27.1	232.2	237.4	242.5	247.7	252.9	258.0	263.2	268.3	273.5	278.7	283.8	289.0	294.1	299.3	304.5	309.6	314.8	319.9	325.1	330.3
35.0	240.3	245.6	251.0	256.3	261.6	267.0	272.3	277.7	283.0	288.3	293.7	299.0	304.4	309.7	315.0	320.4	325.7	331.1	336.4	341.7
43.0	248.5	254.0	259.5	265.1	270.6	276.1	281.6	287.2	292.7	298.2	303.7	309.3	314.8	320.3	325.8	331.3	336.9	342.4	347.9	353.4
51.1	256.9	262.6	268.3	274.0	279.7	285.4	291.1	296.8	302.5	308.2	313.9	319.6	325.4	331.1	336.8	342.5	348.2	353.9	359.6	365.3
59.5	265.3	271.2	277.1	283.0	288.9	294.8	300.7	306.6	312.5	318.4	324.3	330.2	336.1	342.0	347.9	353.8	359.7	365.6	371.5	377.4
67.9	274.0	280.1	286.2	292.2	298.3	304.4	310.5	316.6	322.7	328.8	334.9	340.9	347.0	353.1	359.2	365.3	371.4	377.5	383.6	389.7
76.5	282.7	289.0	295.3	301.6	307.9	314.2	320.4	326.7	333.0	339.3	345.6	351.9	358.1	364.4	370.7	377.0	383.3	389.6	395.8	402.1
85.2	291.6	298.1	304.6	311.1	317.6	324.1	330.5	337.0	343.5	350.0	356.5	362.9	369.4	375.9	382.4	388.9	395.3	401.8	408.3	414.8
94.0	300.7	307.4	314.1	320.7	327.4	334.1	340.8	347.5	354.1	360.8	367.5	374.2	380.9	387.6	394.2	400.9	407.6	414.3	421.0	427.6
03.0	309.9	316.8	323.6	330.5	337.4	344.3	351.2	358.1	365.0	371.8	378.7	385.6	392.5	399.4	406.3	413.2	420.0	426.9	433.8	440.7
12.1	319.2	326.3	333.4	340.5	347.6	354.7	361.7	368.8	375.9	383.0	390.1	397.2	404.3	411.4	418.5	425.6	432.7	439.8	446.9	454.0
21.3	328.6	336.0	343.3	350.6	357.9	365.2	372.5	379.8	387.1	394.4	401.7	409.0	416.3	423.6	430.9	438.2	445.5	452.8	460.1	467.4
30.7	338.2	345.8	353.3	360.8	368.3	375.8	383.3	390.9	398.4	405.9	413.4	420.9	428.4	436.0	443.5	451.0	458.5	466.0	473.5	481.1
40.2	348.0	355.7	363.4	371.2	378.9	386.6	394.4	402.1	409.8	417.6	425.3	433.0	440.8	448.5	456.2	464.0	471.7	479.4	487.2	494.9
49.9	357.8	365.8	373.8	381.7	389.7	397.6	405.6	413.5	421.5	429.4	437.4	445.3	453.3	461.2	469.2	477.1	485.1	493.0	501.0	508.9
59.7	367.9	376.0	384.2	392.4	400.6	408.7	416.9	425.1	433.3	441.4	449.6	457.8	466.0	474.1	482.3	490.5	498.6	506.8	515.0	523.2
69.6	378.0	386.4	394.8	403.2	411.6	420.0	428.4	436.8	445.2	453.6	462.0	470.4	478.8	487.2	495.6	504.0	512.4	520.8	529.2	537.6
79.7	388.3	396.9	405.5	414.2	422.8	431.4	440.1	448.7	457.3	465.9	474.6	483.2	491.8	500.5	509.1	517.7	526.3	535.0	543.6	552.2
89.9	398.7	407.6	416.4	425.3	434.2	443.0	451.9	460.7	469.6	478.5	487.3	496.2	505.0	513.9	522.8	531.6	540.5	549.3	558.2	567.1
00.2	409.3	418.4	427.5	436.6	445.7	454.7	463.8	472.9	482.0	491.1	500.2	509.3	518.4	527.5	536.6	545.7	554.8	563.9	573.0	582.1
10.6	420.0	429.3	438.6	448.0	457.3	466.6	476.0	485.3	494.6	504.0	513.3	522.6	532.0	541.3	550.6	560.0	569.3	578.6	588.0	597.3
21.2	430.8	440.4	450.0	459.5	469.1	478.7	488.3	497.8	507.4	517.0	526.5	536.1	545.7	555.3	564.8	574.4	584.0	593.6	603.1	612.7
32.0	441.8	451.6	461.4	471.2	481.1	490.9	500.7	510.5	520.3	530.1	540.0	549.8	559.6	569.4	579.2	589.0	598.9	608.7	618.5	628.3
42.8	452.9	463.0	473.0	483.1	493.2	503.2	513.3	523.4	533.4	543.5	553.5	563.6	573.7	583.7	593.8	603.9	613.9	624.0	634.1	644.1
53.8	464.2	474.5	484.8	495.1	505.4	515.7	526.0	536.4	546.7	557.0	567.3	577.6	587.9	598.2	608.6	618.9	629.2	639.5	649.8	660.1
65.0	475.5	486.1	496.7	507.2	517.8	528.4	538.9	549.5	560.1	570.6	581.2	591.8	602.4	612.9	623.5	634.1	644.6	655.2	665.8	676.3
76.2	487.1	497.9	508.7	519.5	530.4	541.2	552.0	562.8	573.7	584.5	595.3	606.1	617.0	627.8	638.6	649.4	660.2	671.1	681.9	692.7
87.7	498.7	509.8	520.9	532.0	543.1	554.2	565.2	576.3	587.4	598.5	609.6	620.6	631.7	642.8	653.9	665.0	676.1	687.1	698.2	709.3
99.2	510.5	521.9	533.2	544.6	555.9	567.3	578.6	590.0	601.3	612.6	624.0	635.3	646.7	658.0	669.4	680.7	692.1	703.4	714.8	726.1
10.9	522.5	534.1	545.7	557.3	568.9	580.5	592.1	603.8	615.4	627.0	638.6	650.2	661.8	673.4	685.0	696.6	708.3	719.9	731.5	743.1
22.7	534.6	546.4	558.3	570.2	582.1	594.0	605.8	617.7	629.6	641.5	653.4	665.2	677.1	689.0	700.9	712.7	724.6	736.5	748.4	760.3
34.6	546.8	558.9	571.1	583.2	595.4	607.5	619.7	631.8	644.0	656.1	668.3	680.4	692.6	704.7	716.9	729.0	741.2	753.3	765.5	777.6
46.7	559.1	571.6	584.0	596.4	608.8	621.3	633.7	646.1	658.5	671.0	683.4	695.8	708.2	720.7	733.1	745.5	757.9	770.4	782.8	795.2
58.9	571.6	584.3	597.0	609.7	622.4	635.1	647.8	660.6	673.3	686.0	698.7	711.4	724.1	736.8	749.5	762.2	774.9	787.6	800.3	813.0
71.3	584.3	597.2	610.2	623.2	636.2	649.2	662.2	675.1	688.1	701.1	714.1	727.1	740.1	753.0	766.0	779.0	792.0	805.0	818.0	831.0

PLATE III., accompanying article on "Steel," by Dr. Walter Rosenhain.

(See page 511.)



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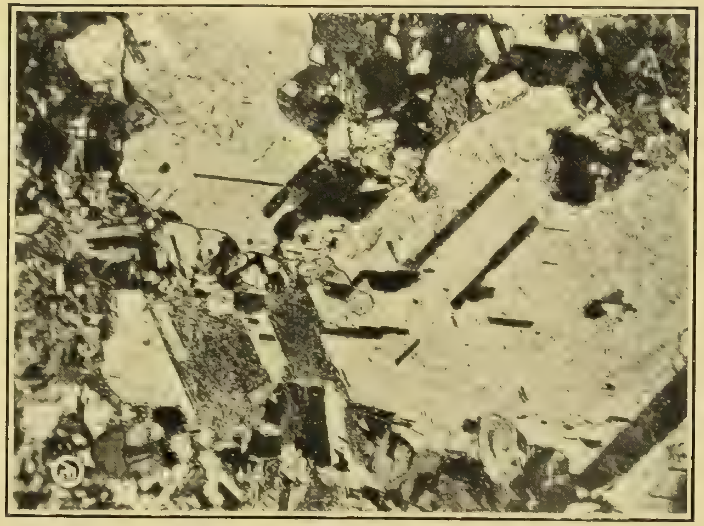

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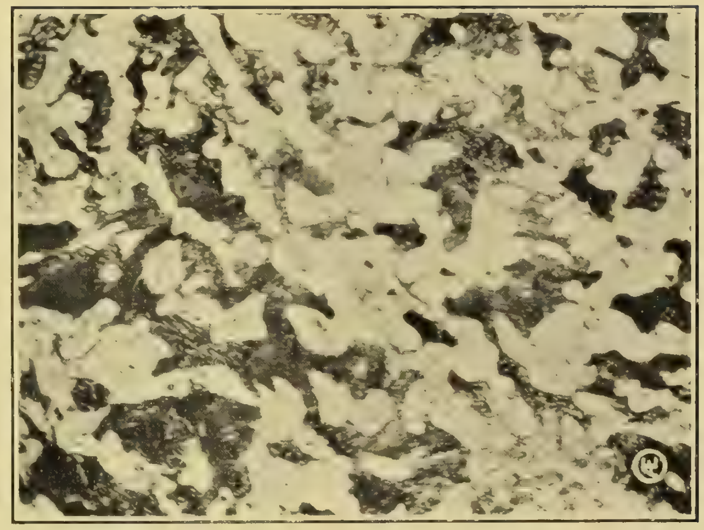

Fig. V.

Fig. VI.

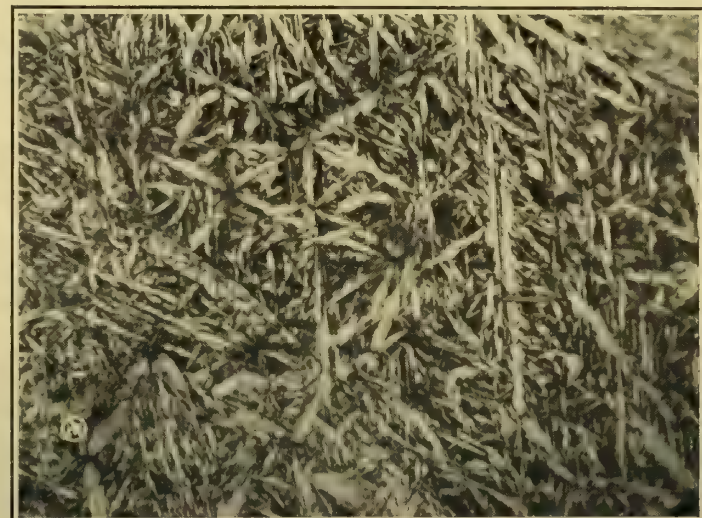

Fig. VII.

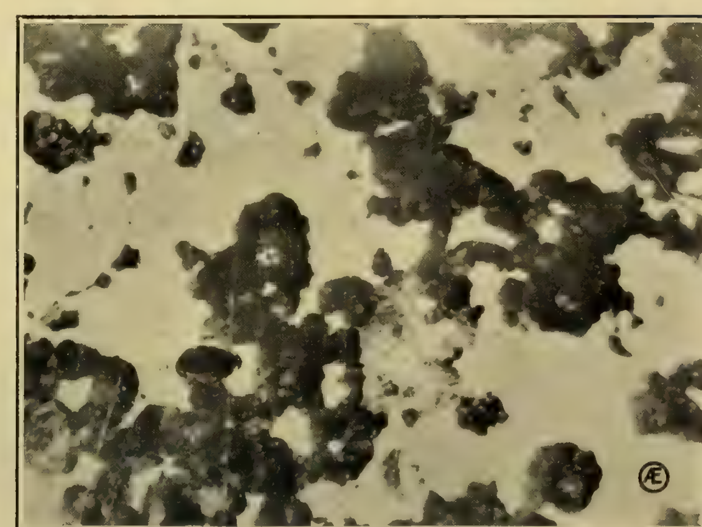

Fig. VIII.

THE

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CONTENTS.

Also a special loose Supplement, giving the displacement volumes in cubic inches for engines of varying bore and stroke, accompanies this issue.

* A supplementary plate of photographic illustrations accompanies this article

THE VALUE OF RACING CAR BUILDING.

THIS year, for the first time in the history of Brooklands motor racing, there have been races of the type which might be termed standard car races. As there appears every likelihood of a repetition of this particular type of competition, it is well to see exactly what effect such races are likely to have on the ordinary touring car, and how much they are likely to benefit the trade. In the first place, cars entered for a standard car race must be standard in every particular, and it is therefore out of the question to attempt any new experiment or alter any existing design with a view to discovering its efficiency or otherwise. It is perfectly safe to be guided by such a race in choosing a car—as far as the reliability of that car is concerned—though it would probably be better to arrange that the bodies of the cars were in place, instead of racing the bare, stripped chassis, as has been done in those races already held. Accordingly it may be inferred that a standard car race is far more a guide as to what exactly has been accomplished by manufacturers, than it is a race likely to provide lessons for the designers of the car which may lead to increased speed and efficiency.

Racing car competitions pure and simple are very different affairs to the ordinary standard car race; in every case the cars competing therein differ considerably from those which are exhibited in the showrooms of the firm in question. Accordingly a number of useful and instructive experiments can be carried out which are bound to teach the designer what he can incorporate in his touring vehicle and what can be done to increase the power, speed and flexibility of the machine he sells. Wind resistance must be studied, and therefrom must arise numerous devices which all tend to simplify body line and an absence of obstructions which are likely to create undue disturbance, when the car is travelling at any speed. It is an undoubted fact that the running of a racing car has a very great effect on the actual touring car, and a still greater effect on the staff of the factory that produces it. Before attempting the construction of one of these speed machines, it is quite safe to say that the factory staff and mechanics do not know what the real capabilities of a petrol internal combustion engine are. They are content to produce a vehicle which may or may not compare favourably with the creations of other motor car manufacturers but, as a general rule, the standard car produced by makers with long racing experience is a good one. This is especially the case in modern times, because the engine bore is gradually becoming limited and consequently the engine has to show a considerable liveliness and be capable of great speeds of revolution, before it can become satisfactory or tackle the loads which are apportioned to it. Unbalanced weight, weight of reciprocating parts, systems of lubrication, water cooling systems, and, above all, the carburettor, are things which are altered to an immense degree in the course of racing practice. All of these subsequently show up in the touring car made by the same firm, whilst innumerable details rendering the car excessively easy to handle and making it accessible, are all produced from the same cause. Accordingly, if the standard car race were to have taken the place of the racing car competition, much less would have been learnt by designers than has actually been the case with racing as it has been.

It is exceedingly unlikely that the speed of standard cars could ever have been raised to the extent found possible in racing cars pure and simple, and so the thousand and one details, the presence or absence of which bring about the speed, would not have been discovered with the same ease and rapidity as has been the case. It would not have become so necessary to eliminate wind resistance, and beyond the thorough proving of the touring car could not possibly have resulted in the enormous advance in design observable in the standard cars of the last few years. It is only by trying new things and new designs that improvement in any car can be effected, and the rules of any standard car race will prohibit the adoption of any novelty in the race in question. It is possible that in a length of time a considerable number of standard car races would result in the appearance of a special design of car, catalogued as one of the standard cars, and in reality a racing car, but if this is done the trouble originally mentioned still remains, namely, that having settled on a design, though it may be suitable for the competition it is intended to enter the car for, any small point which proves to be unsatisfactory cannot be altered until the new model appears in the maker's catalogue. It is probable that the wisest plan of action would be to settle on one or two standard car races during the season exactly in the manner in which they were held during the present year. This would leave the manufacturers perfectly free to experiment with racing vehicles of new designs and to alter these vehicles as much as they think fit or the conditions demand, laying up for themselves a considerable stock of very valuable information.

AMERICAN MANUFACTURING.

Being the fourth of a series of articles on the American industry by a member of "The Automobile Engineer" staff who has recently made an extended tour of the North Eastern United States.

IN the article which appeared in last month's issue some general phases of American manufacturing were considered, and a little special attention was given to the constitution of American companies and their staffs. There is, however, one temperamental thing which has a great effect on all industries in America, and this is the fact that men do not choose a profession or a business for life, in the way which is usual in this country. A man who is occupying a high position on the engineering side of the automobile industry to-day may be the moving spirit of a typewriter concern to-morrow, and the next day his only interest in life might be something equally different. Similarly, the capitalists who are now taking up motor car manufacture are not by any means always doing so with the intention of studying the business and building up great firms with great reputations—they are often content if they can work at it for three, four, or five years, clear some profit, and then transfer their attention to the next new thing. Really the whole business amounts to the fact that in a large, sparsely-populated country, of which the natural resources are not yet fully attacked, it is possible to look ahead no further than three or four years. There are, of course, exceptions, and many of them, but this opinion represents the average with very fair accuracy. This is why America is the producer of such an enormous number of cheap and far from nice cars. The big concerns who plan to flood the market with very cheap vehicles do not expect to keep on doing so; they calculate their prices and their outputs so that they may be able to retire with a profit very quickly, and after that it is of no interest to them whether the industry continues or dies absolutely.

However, none of the foregoing in any way explains how it is that the cheap American cars can be sold profitably at such low prices, especially when it is remembered that labour is nearly three times as costly. Not only does the labour in the shops cost more, but the staff expenses are much greater; the railway costs for distributing the produce amount to a great deal; agents' discounts are extremely generous, advertising is necessarily prolific, and advertising rates are mostly in keeping with the national scale of expenditure. Every sort of establishment charge is thus twice or thrice as much as it would be anywhere in Europe, and practically nothing is cheaper, because steel producers, for example, have the same sort of wages to pay as any other kind of manufacturer. Of course, without actually going through the books of a number of concerns it would be impossible to say with absolute certainty how it is that the American car prices can be kept down, but I believe that large outputs enable the established charges to be split up so that the amount per car becomes even smaller than it is here (for example, it takes the sales department probably no longer to sell one hundred cars to one of the big-district American agents than it does to sell two to a small provincial business here), and in the works far more automatic and semi-automatic machining is used, while there is no trouble in keeping the whole plant occupied. Where two gear cutters will supply all the transmissions necessary for an output, probably one multi-spindle boring machine would be equal to machining all the cylinders in a week or two. In America the position is different, for there several boring machines can be kept busy, and a dozen or more gear cutters. Taking operation by operation there is no great difference between one country and another, beyond the fact, and which has already been mentioned, that machines are all new, but tools are set up to produce automatic parts which would not be so treated here. In several works, in which I was a visitor, for example, pistons were being turned out in automatic capstan lathes, the only hand operation being the removal of the finished article and the chucking of the fresh casting. Again, the larger the quantity of any sort of product, the smaller is the wastage, with an equal administrative care, but where the greatest difference of all comes in is in the *elimination* of machining. Crankshafts are customarily stamped, and the journals are turned or ground, but the webs are often never touched at all by any tool. The starting handle instead of being fitted to a shaft carrying the jaw (four pieces usually) will be made from one piece of rod bent twice at right angles. For everywhere where we use malleable cast iron, stamped or drop forged steel will be found, and usually with so high an accuracy of finish that on such parts as the brake arms and links the amount of machining needed is very small indeed. Nor is the same care taken in testing; the majority of the cheap cars certainly do get a road test, but it is not very severe, and it is only for the better-class cars that a brake test is made of the engine separately. Engines are invariably run in, and in some districts where natural gas is obtainable this fuel is used in preference to petrol. It is well known here that the amount of testing needed by cars has decreased a good deal in late years, and because the testing is none too searching it does not follow necessarily that a high percentage of American cars leave their factories less efficient than they ought to be. When cars are being turned out at six or seven times the rate of our largest places here, any sort of weaknesses show up fairly promptly and are easy to eliminate. Also the higher the quantity of repetition for each part, the smaller the variation in accuracy, and perhaps even more important, the greater the extent to which erecting is split up in its details and the greater the skill acquired by the erector at his particular job. Again taking, for example, the Cadillac Company, here several men do nothing but fit big ends, while others fit the small ends, and so on throughout the engine. By such means the chances of human error are reduced very considerably.

Another way in which I fancy expenses are kept down is in working more accurately to system. Of course, most of the good works here are now running their business systematically, but it is only in a few really big works that one finds a system working with perfect smoothness and the accuracy which its inventor imagined. In America time saving has been reduced to the finest of fine arts, though one is sometimes tempted to fancy that the systems in use for costing and checking one thing and another are in themselves needlessly extravagant. One big works, which it is not necessary to name, had 8 per cent. of its employees in the costing department, though as the company is particularly successful, possibly this is not an extravagant quantity. Wherever machines can be used instead of manual work one finds them in employment; in fact, often nearly everything, from the workmen's job card and time clock to the paying out of the money earned at the end of the week, is entirely automatic. To instal a complete automatic wages costing system in a factory employing 500 hands would cost nearly as much as a similar installation for a works of ten times the size, and to say how big a percentage of one's capital it is advisable to spend on what is really checking apparatus is a matter which each man must decide for himself; no hard and fast rule has ever been laid down, nor is it likely to be for many years to come. An example of the way in which valuable staff time is saved, which may perhaps be cited, is known as the call bell system, and this one finds in nearly every big works. The president, vice-president, general manager, chief engineer, and a few others are liable to be called to any part of the factory on one thing and another, while they are also needed constantly in the office. Even in a small works everyone knows that finding a particular individual may take quite a long time, and in a really big factory an important man might easily be lost for half a day. Therefore each individual has assigned to him a Morse call, and every shop has a bell which will ring a Morse signal when switched on from the works telephone exchange. Thus, supposing the chief engineer is required and is not in his office, the telephone operator merely switches on his call, which immediately starts all the bells throughout the works ringing his particular Morse designation. On hearing it he takes up the nearest telephone and deals with the matter for which he is required. A variation of this system is to use colours instead of codes and to substitute a row of different tinted electric lamps for the bells. Probably neither has a special advantage, but the bell installation is cheaper, as it requires very much less wiring.

With the exception of a few of the leading concerns there is no doubt that American-made cars are not only rather rough externally, but have not quite the same accuracy of internal finish as most European vehicles. Although there is little truth in the belief of the so-called "man in the street" here that everything in America is done in a hurry—no more truth, in fact, than the corresponding American notion that everybody here sleeps about twelve hours out of twenty-four—still undoubtedly it is more difficult in America to obtain very careful,

painstaking workmanship. Automobile builders have had very great trouble to get satisfactory steels, because the big steel concerns do not appear to care about comparatively small orders for high-class material. Really good alloy steels are very troublesome to produce satisfactorily, and the American steel maker prefers to keep his establishment busy on the production of cheap, simple steels, and not to trouble about cultivating a new market. It is largely this undoubted fact which has led to the extremely elaborate report of the material standards committee of the Society of Automobile Engineers, which was published in these columns in the July issue. Here there is no difficulty in persuading steel makers to guarantee composition within very fine limits indeed. In America it is not too easy, apparently, to get any sort of composition guaranteed. The same applies to pig iron and to fine qualities of malleable iron, these being practically unobtainable. Stamping, or drop forging as it is called in America, has been developed to a high degree of perfection and a large number of the really skilled mechanics of America are to be found in the die rooms of stamping firms. An indication of the acknowledged superiority of British work is given by the facility with which an English fitter can obtain employment at high wages—high even for America. Several of the largest motor manufacturing companies have made strenuous efforts to staff their tool rooms with English labour because of the difficulty in getting sufficiently careful work from the natives.

So far consideration has been given in these articles to the design of touring cars and their manufacture, and all that has been said on the latter section applies with equal force to traction vehicles. At present there is a mild boom in lorries, or "trucks" as they are called in the United States, "truck" being a generic word applicable to almost every sort of road vehicle excepting a carriage. Some such word is needed rather badly here to replace the clumsy "commercial vehicle." This little boom has led to the establishment of quite a number of small concerns making a very limited output, and these are, of course, assembling factories always. The word "small" is used here in the English sense, some of the little companies doing no more than a chassis or two per week. The future before such concerns is a debatable matter, and it is interesting to notice that a good number of the large touring car companies do not place much faith in the possibilities of the heavy car trade, because American road conditions confine anything weighing more than an ordinary touring car to the paved roads in urban areas. Even suburban delivering round the larger provincial towns can scarcely be undertaken in winter, except by quite light cars. Thus one finds in New York and Chicago, where there are many miles of comparatively well-paved highway, that the larger business houses transport their goods principally by motor vehicles, but the country districts are not a possible market. This means that while the heavy car trade will assuredly soon be a bigger thing here than the touring car business, in America the latter is likely to hold the upper position for a very long time, although, of course, this does not prevent the American heavy car trade assuming very large proportions as compared with our own as it is to-day.

In the first issue of *The Automobile Engineer* the editor of *The Autocar* commented upon the prevalence amongst manufacturers here of an extraordinary lack of knowledge concerning the cars produced by their competitors. We have since several times referred to this undoubted fact, and it is often amazing how easily satisfied some manufacturers are, simply through failure to appreciate what other cars than their own are capable of. In America this is by no means the case so far as one and another American manufacturer are concerned. In fact, the American trade on the professional side go out of their way to assist each other. The chief engineer, a high official, or anyone from the designing department in an American automobile factory, is always welcomed at a rival's works, and at that works he will be shown everything there is going on, with the possible exception of experimental or research work. It seems that it is the regular thing for official visits of inspection to be interchanged betweeen one works and another, a party consisting of perhaps the president, general manager, and chief engineer being invited to exchange courtesies with a similar trio from another works. Thus one finds that American designers are not only entirely in touch with all that is going on around them, but in the actual machine shop processes there are no secrets. It has been realised long ago in America that in commerce the only sort of secret which can be kept is the kind that is not worth keeping. A manufacturer who exchanges visits with half-a-dozen other firms finds that the hints which he picks up by observing the practice of the others are of far more value to his business than the few things which the others learn from him would have been had they been kept secret. Here it is to be noticed that firms often persist in the employment of wasteful old-fashioned methods solely through lack of knowledge concerning modern ones. In America everyone uses the most modern and up-to-date methods, because they keep in close touch with each other, and the result is that the whole trade is in a very flourishing condition.

Curious as it may seem, commercial history would lead one to believe that the most prosperous trades are those which are conducted best when taken altogether. For one or two firms to be conspicuously the best managed out of fifty or sixty, say, would seem to augur well for the prosperity of those firms, but if the others were using wasteful processes and running their businesses in a slipshod way, they could not be giving their customers reasonable value for money in the shape of the cars they were turning out, and in service by way of accurately fitting replacement parts delivered promptly, and so on. Now the one or two well-managed firms might be giving everything as it should be, but motor cars in the lump would appear to the general public to be handled by an unbusinesslike set of men, and this would not tend towards the magnification of the industry. The more really good, pushing firms there are in the trade the more prosperous is any one of them likely to be, because the better the quality of cars becomes, the better is the education of the public to motoring. The greater the commercial activity of the industry, the more are people encouraged to become car users and the greater the volume of business is there to be done.

In concluding these articles, in which I have attempted to outline the policy of the American manufacturers, mention must necessarily be made of the competition which now exists between a small number of the American concerns and our home manufacturers. *The Automobile Engineer* is not a commercial paper, and has no ambition to become one. It is content to deal purely with the professional side of the industry, to devote itself to engineering *per se*. However, in the next issue the present writer hopes to express some views concerning the best way in which British manufacturers can face competition from without in this country and, what is probably still more important, proceed actually to carry competition into the colonies which have been supplied very largely with cars of American origin.

STEEL.

By Walter Rosenhain, B.A., D.Sc.

(Continued from page 487.)

IN the previous articles we have considered the behaviour of steel when cooled from high temperatures at moderate rates, and both the thermal phenomena and the resulting micro-structures have been briefly dealt with. We have now to consider the effects both of more rapid cooling and of prolonged heating. The first of these questions brings us to the subject of the hardening and tempering of steel, while the second relates to "heat treatment."

The phenomena connected with the hardening of steel by quenching have been known for many centuries, but it is only the advent of metallographic methods of investigation which has given us an insight into the nature and causes of these phenomena. The process of rapid cooling or "quenching" is one of the methods employed in the study of many groups of alloys, but its effects are nowhere more marked than in the case of steel. But at the outset it is necessary to notice that although quenching implies rapid cooling, such cooling is never really instantaneous —even small specimens of metal cannot be cooled down from a red heat to the ordinary temperature in much less than five seconds, even when immersed in iced brine. Now, if it were possible to chill a specimen of metal really instantaneously —so that the time of cooling was, perhaps, less than one-tenth of a second—we might expect the effect to be a complete

retention of the metal in just that condition of structure and constitution in which it happened to find itself at the moment of quenching; in such a case the molecules would be locked in the comparative rigidity of the low temperatures before they had time to undergo any kind of re-arrangement. This is what might be termed "ideal" quenching, although its results would, in the case of steel, be far from ideal from the practical

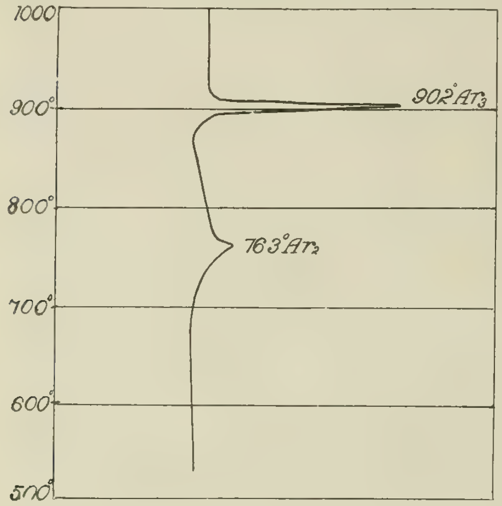


Fig. 1.

point of view. Practical quenching, however, always allows time for a considerable amount of molecular re-arrangement, provided that the tendency for such changes to take place is very strong. Thus some alloys undergo changes on cooling which can be entirely suppressed by a rate of cooling as slow as 20° C. per second; in the case of steel these changes are not entirely prevented by a rate of cooling as rapid as 200° C. per second.

What is the nature of the changes which cooling steel undergoes with such vigour and rapidity, and how are they modified by quenching? We must begin by briefly considering the behaviour of pure iron, which is the end-member and —after all—the dominating partner in the series of iron-carbon alloys. A cooling-curve of pure iron, such as that shown in Fig. I., shows two marked evolutions of heat, one near 900° C., and the second near 750° C. These evolutions of heat show that the iron undergoes profound internal re-arrangement, and this evidence is corroborated by other facts. One of the most striking is that above the lower of these heat-evolutions —generally called "Ar_2"—the metal is non-magnetic, while it becomes strongly magnetic as soon as it has passed through the critical point in question. Other differences, more directly bearing on our subject, have recently been discovered. If a thin strip of iron is heated—preferably in a highly exhausted vessel so as to avoid all oxidation—in such a manner that the middle of the strip is hotter than the two ends, it is found that one of the critical points makes itself felt by a striking change in the strength of the iron which takes place at that temperature. Such a thin strip when strained in tension, is found not to be weakest in the middle, where it is hottest, but considerably nearer the cooler ends, and if the surface of the strip has been polished before it is put into the apparatus, then the effects of the applied strain can afterwards be observed on the surface of the specimen. These observations show the existence of one very marked change in the properties of the metal. This change is illustrated in Fig. II. (Plate III.). Towards one side of this photograph the surface of the iron is seen to be closely cross-hatched with more or less curved lines and ridges, but this disturbed area ends at a definite line. To the right of this line the surface of the metal has remained almost perfectly smooth. Now the depth and intensity of the cross-hatching seen on the surface of strained metal is an indication of the extent to which the specimen has undergone plastic deformation; thus the presence of this hatching to one side of a definite line, while it is absent on the other side, clearly shows that on the first side the metal was much softer and weaker than on the second. Now the remarkable fact in this case is that the iron was hotter on the second and colder on the first side of the area represented by the photograph. This means that, contrary to the almost universal rule, the metal had become harder and stronger in spite of being hotter—the same pull which could deform the iron to the point of breaking it to the left of the critical point, was insufficient to strain it markedly on the right, and this change had occurred in a distance of less than one hundredth of an inch. Such a sudden transition in properties is typical of what are known as "allotropic" changes, and its importance lies in the present instance in the fact that we have a change from a softer to a harder modification of iron on passing through a critical point. On cooling, this change is reversed and the iron returns to its softer state unless the change can be prevented by sufficiently rapid cooling. In the case of pure iron, this prevention is found not to be possible, and for that reason pure iron cannot be hardened by quenching. But reference to our equilibrium diagram, as explained in Fig. XI. of the second article and reproduced here (Fig. III.) for convenience, will at once explain why the iron-carbon alloys differ materially from pure iron in this respect. The allotropic or γ-iron, existing above the critical points (Ar_3) and (Ar_2) of the diagram, in changing into the ordinary soft α-iron, has merely to undergo a re-arrangement of its own molecules, but there is no kind of separation of different constituents. The iron-carbon alloys are in a different position; at temperatures which lie within the region A.D.I.F. they also consist essentially of allotropic or γ-iron, but this γ-iron holds a certain quantity of iron carbide in solid solution. Now when the iron undergoes its allotropic transformation it loses its power of holding iron carbide in solution, and during the transformation of the iron itself the carbide is separated into a distinct constituent. This means that the molecules of iron and of iron-carbide are obliged to sort themselves out and to move through comparatively long distances. Such an operation requires a certain short time for its completion and the result is that the separating and sorting-out process is partially prevented when the metal is quenched. It follows that, although quenching is unable to retain the steel in the condition of the homogeneous solid solution of iron-carbide in γ-iron, it prevents the formation of ferrite and pearlite which would occur with slow cooling, and instead we meet with a series of transition products which correspond to the various stages of the sorting-out process referred to above. It is found, however, that the addition of certain elements to the iron tends to retard the separations in question, with the result that a steel containing, for instance, about 2 per cent. of manganese, can be obtained in the condition of a homogeneous solid solution by vigorous quenching This homogeneous solid solution, which is the product of an "ideal" quenching of carbon steels, has received the name "Austenite." It is met with in practice only in small patches in steels containing much carbon and violently quenched from excessively high temperatures. In alloy steels, however, it is frequently found and consequently certain of these steels are sometimes called "Austenitic." That carbon steel really consists entirely of such Austenite when at a high temperature (within the region of the diagram) has been proved by etching polished specimens of steel at these high temperatures by means of gaseous hydrochloric acid—a process first employed by Baykoff. The appearance of such a hot-etched steel is shown in Fig. IV., which shows the typical structure of a homogeneous solid solution and the "twinning" typical of γ-iron. The same steel when slowly cooled has the structure shown in Fig. V. These two photographs represent the two extreme limits of condition in which this steel can exist —the one stable only at high temperatures, the other stable at low temperatures; the products of quenching and tempering are intermediate stages between these two.

One of the most striking of these intermediate stages is that known as "Martensite," although this name rather indicates a transitory condition than a true constituent. Martensite is always found in association with Austenite in severely-quenched high-carbon steels; its appearance in these circumstances is shown in Fig. VI., while its typical appearance when seen alone is shown in Fig. VII. The appearance of interlacing needles is very characteristic, and it is interesting to note that the higher the temperature to which the steel was raised before quenching, the coarser is the grain

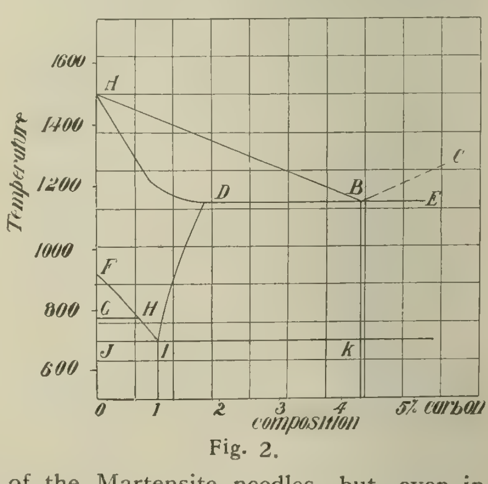


Fig. 2.

of the Martensite needles, but even in carefully-hardened steel quenched from a temperature just above the critical point, suitable treatment will reveal the presence of Martensitic structure. As to the nature of this transition-form, it can only be said that it is the hardest condition of carbon steel, being distinctly harder than

Austenite. The interlacing structure appears to be connected with the cleavage and twinning-planes of the previously-existing Austenite crystals. In fact, we know that, just as a supersaturated solution will deposit its over-burden of crystals wherever the solution has been disturbed, so will over-burdened Austenite deposit its excess of carbide on the cleavage and twinning planes where disturbance is most likely to have taken place. We may say, then, that Martensite represents the first stage of the "sorting-out process," which has as yet been confined to the cleavage and twinning planes of the Austenite. The fact that Martensitic steels are magnetic indicates that some of the iron, at all events, has already passed into the α-condition, but it is probable that this change has only taken place to a very limited extent at those places where the carbide has begun to separate out. The extreme hardness of Martensitic steel may thus be due to the joint action of several causes :—First, the presence of a large proportion of iron in a hard allotropic form, although there is no reason for supposing the allotropic variety of iron to possess fabled "adamantine" hardness, and second to the presence of what one might term incipient separated carbide in all the cleavage or gliding planes of the crystals, cementing them together and preventing them from displaying the softness and ductility they might otherwise possess.

The Martensite structure just described is only found in fully-hardened or "glass hard" steels, at all events so far as high-carbon steels are concerned. In mild, low-carbon steels Martensitic structures can also be obtained by vigorous quenching, but in the absence of a sufficient amount of carbon the hardness attained is not very great—too much of the iron has had time to run down into the soft α-condition. On the other hand, when high-carbon steels are either tempered after hardening or are only partially hardened in the first instance, by such means as quenching in oil or by quenching during the progress of the recalescence, then the Martensite is more or less completely replaced by other transition forms.

Perhaps the most interesting of these is that known as "Troostite," which is the next stage after Martensite in the "sorting-out" process. The characteristic appearance of Troostite is shown in Fig. VIII. (Plate III). It is found in roughly rounded masses, frequently occurring in rows or chains, and readily recognised by their tendency to darken very rapidly under the influence of the usual etching reagents. This readiness to be attacked by chemical agents differentiates Troostite from Martensite, although the line of demarcation between the two forms is not sharp—the darker bands of Martensite may, indeed, be regarded as minute regions of Troostite. The appearance and character of Troostite are perhaps best explained by supposing that at this stage the carbide has begun to separate from the iron throughout the mass of crystals, and that it now exists in the form of ultra-microscopic particles, somewhat akin to a colloidal suspension. The hardness of Troostite is decidedly less than that of Martensite, probably because the transformation of the iron itself has gone on to a greater extent.

Troostite and Martensite are frequently found side by side in the same specimen of steel, but one definite stage is reached, particularly by tempering hardened steel at 400° C., where the steel contains a maximum proportion of Troosite; steel in this stage has been given the name of "Osmondite." This also is a transition stage, and not a true constituent like Ferrite or Austenite. Finally, in the passage from the Troostite stage to the condition of Pearlite, some observers identify another stage under the name of "Sorbite," which we may, however, regard as merely a variety of Pearlite, having an extremely fine duplex structure. Where Sorbite ends and Pearlite begins is merely a question of the resolving power of the microscope.

We have now described and illustrated some of the principal structures met with in hardened and tempered carbon steels, and some account of the causes at work in the production of hardened steel have been indicated. It is perhaps true to say that a perfectly satisfactory explanation of the hardness of quenched steels has yet to be given, but the metallography of steel has already thrown much light on this difficult question and has furnished us with ideas which are adequate to serve as a sound working hypothesis until fresh discoveries afford a still deeper insight into these complex phenomena.

(To be continued.)

TUNING UP A CAR FOR THE TRACK.

Some additional Notes by Robert W. A. Brewer, A.M.I.C.E., M.I.A.E., M.I.M.E., F.S.E.

In the September issue of the Automobile Engineer, the article on the above subject which I contributed naturally suffered from several omissions, and it is a pleasure to notice that "Anglo-American" has picked up one of these points, namely, the necessity for balancing the road wheels.

It is really surprising how some wheels refuse to run upon the track in a manner which looks at all safe. During the past few racing seasons, one could not fail to have noticed how very bad some cars were in this respect, their wheels and axles bumping about in the most extraordinary manner. The question of wheel balance was particularly brought home to the writer in the case of the six-cylinder Sunbeam car, which recently made a twelve hours' record. When Mr. Coatalen first had this car on the track it was impossible to drive it at anything like its maximum speed; I believe he was only able to do about 65 m.p.h., on account of the front wheels bumping so badly that the car was difficult to steer. It was first thought that the shock absorbers were at fault, but this was not the case. The difficulty was entirely due to the fact that the air valves threw the wheels out of balance, and when lead washers were correctly inserted behind the fly nuts of the security bolts, the wheels ran perfectly smoothly, and it was possible to drive the car all out.

The question of springing has also a large bearing upon that of smooth running, and the writer finds that three-quarter eliptic springs, or a modification such as is used in the Austin car, are far preferable to the ordinary semi-eliptical type. No doubt this is due to the longer period of such springs, and in combination with absorbers of the J.M. type, it should be quite possible for the different vibrations to damp out in a satisfactory manner. Attention should particularly be called to the advantage that flat springs have over the cambered type for track use, and an experience of the writer's at the last Brooklands meeting may be of interest. The car in question came straight from the works with well-cambered three-quarter eliptic springs, and on the way to Brooklands a pair of J.M. absorbers was fitted. Some doubt was expressed by the writer at the time as to the sufficient lightness of the J.M. springs. The first two or three laps proved that this doubt was well founded, as the riding was very hard, and the bumping so bad that the J.M. absorbers turned round through 180 degrees, and were only with difficulty restored to their natural position. The camber of the springs was responsible for this occurrence, the springs themselves lengthening out considerably between the shackle bolt holes during the bumping. Lighter springs were then fitted to the J.M. absorbers, and a very great improvement was noticed.

There is undoubtedly a distinct relation between the length of the wheel base and the track performance. In road racing other considerations sometimes make it expedient for the wheelbase to be shortened, as instance several notable French racing cars. On the track, however, where there is no cornering, the relation between wheelbase, weight and speed, does not appear to have been definitely investigated, at any rate no publicity has been given to any investigations of this nature. It would appear that a heavy car with a long wheelbase would have the advantage, that for a given amount of vertical oscillation of the ends of the chassis there would be less radial oscillation, if it may be so termed, of various parts consisting of heavy masses. This means that less power would be absorbed in overcoming the inertia of these masses in moving them from their normal position.

Supposing a car is travelling at a high speed and the centre of gravity of the machine is well forward, as is usually the case in a racing car. When a bump is experienced, the centre of gravity of the whole mass is lifted, usually through a curved path, with reference to the car itself. The shape and magnitude of this path are directly proportional to the wheelbase and to the position of the centre of gravity with regard to it. It will generally be conceded that in a short wheelbase car this movement will be more pronounced than in a long wheelbase car, and that therefore, for equal weights, the loss of power is greater in the car with the short wheelbase. Furthermore, it is

quite reasonable to suppose that oscillations of this nature affect the carburation, and they undoubtedly inconvenience the driver.

The use of disc wheels lessens the wind resistance considerably, as compared with plain wooden wheels, but it is difficult to approximate as to the magnitude of the saving of power, as this depends upon car speed and wind velocity. It is quite safe, however, to predict an increase of car speed of four or five miles per hour on the top of, say, 80 m.p.h., as a result of substituting disc for wooden wheels.

In the previous article on the above subject, the shape of valves was referred to, but no mention was made of their weight. Of course, the valves must naturally be reckoned as reciprocating weight, and as such should receive very careful attention. The weight of the valves in some types of car is enormously high and, should it be desired to obtain a high speed of rotation from an engine so fitted, the weight of the valves should be reduced by so forming the top of the valve heads that their shape conforms to that of the lower side of the head, the metal being cut away into the form of a depression. Cutting away metal in this manner slightly increases the clearance volume, of course, but a distinct gain is obtained by this proceeding. The spindles also can often be reduced in diameter through a part of their length.

Amongst other points to be borne in mind when speeding up a standard engine are the strengths of certain parts which are called upon to bear the excessive stresses which result from such increased speed of rotation. In some cases it is advisable to substitute a steel flywheel for a cast iron one, where such is fitted, and especially is this so where the standard flywheel is vaned for cooling purposes—not that supplementary air induction of this kind is necessary in a high speed car. The pump and magneto drive should also be considered, and it is important that the attachment of the pump spindle to the driving gear wheel should be amply strong for the increased load it has to bear, as the resistance of many types of radiators is considerable.

A NEW SLEEVE VALVE ENGINE.

This Engine will be incorporated in the 25 H.P. Argyll for 1912.

DURING the past year two or three patent specifications have been published relating to sleeve valve engine patents by Argyll's, Ltd. In the new 25 h.p. engine the embodiment of some of these is to be found, and the drawings reproduced in Fig. I. give a good general idea of the construction. In this engine only one sleeve is used, and it oscillates as well as moving vertically, the movements in either direction being only slight.

Turning to the transverse section, it will be seen that the bottom of the sleeve is attached by a substantial pin joint to a short shaft which is capable of sliding in a hole bored eccentrically in a large skew gear. The amount of eccentricity is about 17 mm. so 34 mm. represents the vertical travel. The degree of oscillation must of course also be the same, but, owing to the pin being further from the centre than the sleeve, the actual motion of any point on the sleeve is elliptical. Regarding the engine from above, as the sleeve moves downwards from the top dead centre it will be rotated a few degrees in a clockwise direction, and on the ascent this movement will be reversed, so that, if it were possible to mark two spots on the cylinder and one on the sleeve, the sleeve spot would coincide with one of the cylinder spots for an instant during the down stroke, and with the other during the up stroke.

This really is the system on which the engine works. In the sleeve there are five ports, and in the cylinder six, but one of the sleeve ports is of double size. The centre double port and one additional port on each side only are shown, but in fact there are two sleeve ports on each side of the centre double one, as has already been mentioned. The cylinder ports, shown in full line on the left hand side of the centre, connect with the inlet pipe, while those on the right hand side connect with the exhaust. In diagram 1, Fig. II., the inlet is just about to open, that is to say, the sleeve is rising and moving anti-clockwise. Diagram 2 shows that the sleeve ports and the cylinder inlet ports are coincident and represent approximately the maximum opening. The exhaust ports, it will be noticed, are now sliding past each other without any over-lap. In diagram 3, Fig. II., the inlet is just closed, but the sleeve continues to rise, though it is now over the dead centre and commencing to rotate clockwise. The compression and firing strokes now take place and, at the conclusion or just before the conclusion of the latter, the position shown in diagram 5 is reached. Here the exhaust ports are just touching at their edges, the sleeve is travelling downwards at its maximum speed, and has almost ceased clockwise rotation. Diagram 6 shows the exhaust ports over-lapping at their maximum degree of opening, and the next stage in the cycle is of course the return to diagram 1.

Mechanically the engine is not complicated, the camshaft, which is driven by a chain, carrying four small spiral gears which mesh with four wheels of double the diameter actuating the sleeves. In the transverse section the oil pump drive is shown, and it is worthy of comment that this important fitting is placed high where it is easily accessible without draining the sump, and that a ball valve at the bottom of its intake guards against priming troubles. The cylinder head is reminiscent of the Knight engine, practically the same arrangement for ensuring the proper cooling of the sleeve being used, and the water spaces are everywhere ample. Lubrication is by splash from constant level troughs through cast-in passages, and there is a separate spray supply to the half time shaft, this being indicated in the transverse section of the engine. It is to be noticed that the big end dippers are altogether exceptionally large, but that there appears to be no arrangement for protecting the cylinders from over lubrication. Possibly the considerable height of the cylinder base above the crankshaft is sufficient, and certainly the engine is not unusually liable to smoke.

Speaking generally, the engine appears to be well designed, and to possess carefully thought out detail. The oil filler level float and the oil pump are quite exceptionally accessible and convenient. Shafts are of good diameter and well supported, and the design is particularly cleanly externally. Recently we observed the behaviour of one of these engines in the chassis for which it has been designed, during a period of some seven hours running, including some fairly stiff hill work. It is, of course, difficult to judge without speedometer observations, which unfortunately could not be made, but the engine seemed to give quite as much power as may reasonably be expected from a smooth running 100 mm.-130 mm. engine. We were particularly on the alert to observe any vibration, but were unable to detect any bad period, the greatest amount of vibration (in itself slight) appearing to occur at quite a low speed of revolution. Like all sleeve valve engines the new Argyll possesses the peculiar power of improving steadily with use: the accelerative powers were not conspicuously great at the commencement of the run, but were decidedly good after an hour's running. Unfortunately the carburettor was not tuned up properly to suit the engine, and it is impossible to say to what extent this affected its performance. Considering, however, that the engine was the first of an entirely new type, its performance can only be regarded as most promising. Needless to say it is almost noiseless in operation and, owing to the dashboard on the trial car being of a very temporary nature, noises could be detected which would be entirely inaudible under normal conditions.

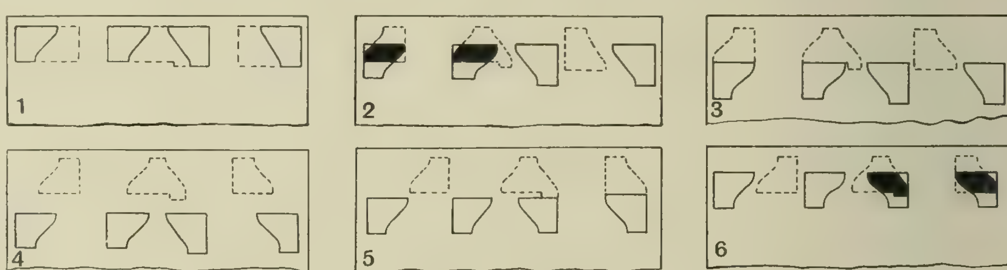


Fig. II.

Port opening diagram of Argyll engine. 1—Inlet about to open. 2—Inlet fully open (as shown by solid back). 3—Inlet just closed. 4—Firing. 5—Exhaust about to open. 6—Exhaust fully open.

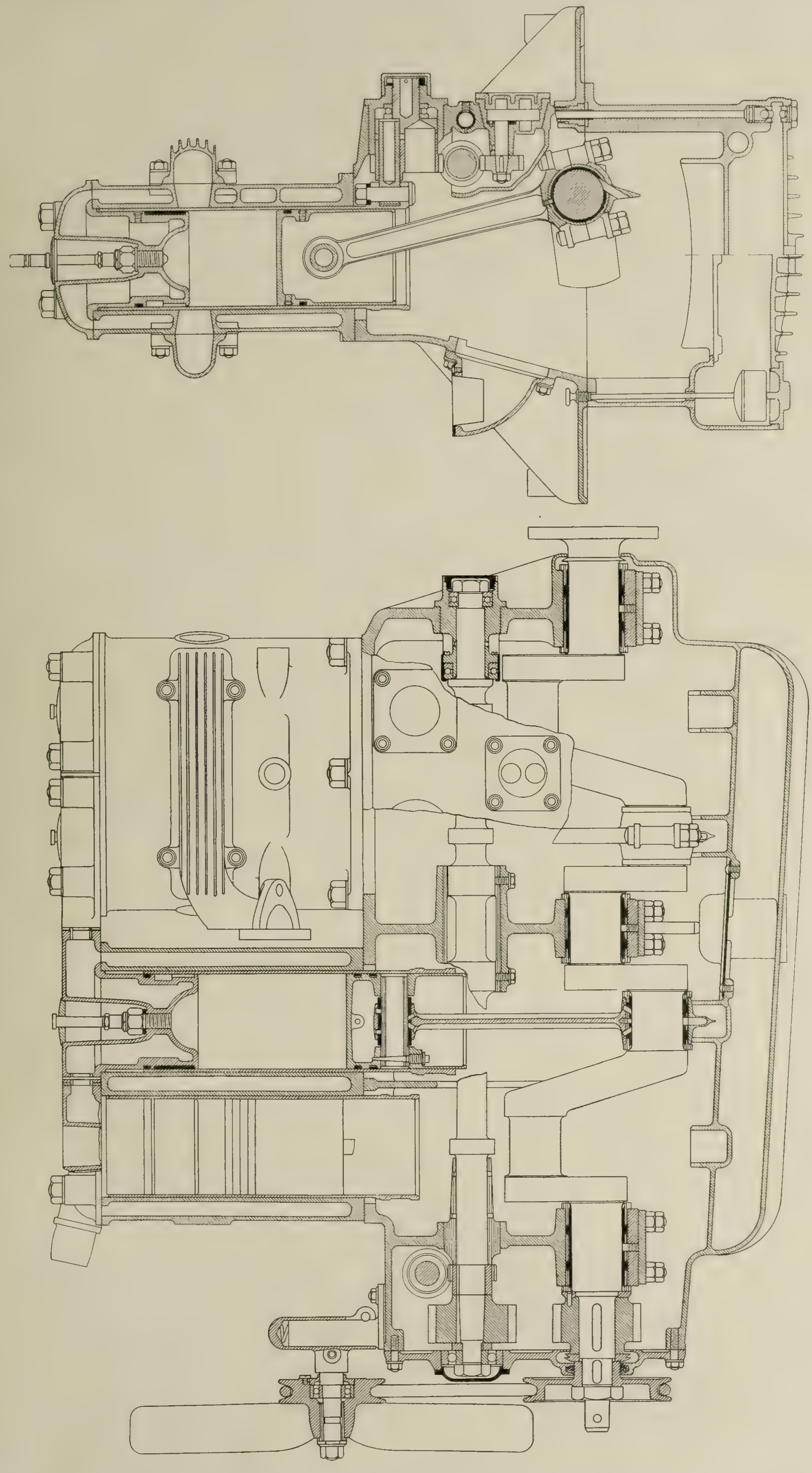


Fig. 1.

THE NEW ARGYLL SLEEVE VALVE ENGINE.

THE 25 H.P. SHEFFIELD SIMPLEX.

A chassis possessing a number of unusual features, and some examples of particularly good design.

PROBABLY there are few cars in which cost of production has been considered less than it is in the case of both the 45 h.p. and 25 h.p. Sheffield Simplex cars. The former has not undergone any very great alteration recently, but the smaller of the two has been almost completely re-designed, at least with regard to the engine, and it is really a particularly praiseworthy job. With a bore of 89 mm. and a stroke of 127 mm. the actual power developed is, of course, greatly in excess of the nominal rating, and the designers endeavour neither to sacrifice comfort to power nor *vice versa*. From Figs. I., II., and III., it will be seen that the cylinders are cast in blocks of three, and that the water spaces are very large indeed, the distance between adjacent cylinders being over 30 mm., and that the crankshaft is exceptionally big. This large crankshaft has been employed to overcome the unpleasant effects of periodic vibration through oscillation of the shaft or webs, and certainly during a short road trial in which the engine was speeded up to its maximum, no period could be detected, the vibration at all times being practically imperceptible. At the journals, of which there are 7, the diameter of the shaft is 68 mm., and the big ends are 45 mm. in diameter, the web being 75 mm. wide, and about 18 mm. in thickness. A chrome nickel steel is used, and the total length of main bearing is about 400 mm., while the big ends are 60 mm. wide. Not only is the shaft thus exceptionally robust, but great care has been taken in designing the crankcase to provide really stiff webs to each bearing, and this may also be observed by reference to the first three illustrations. Even the walls of the crankcase are thicker than is usual, the average being from 7 mm. to 9 mm.

Turning to the other parts of the engine, from which vibration emanates, the pistons and connecting rods are likewise light, although not exceptionally so. There is, besides, the vibration damping arrangement in connection with the camshaft, which will be described further on. In Fig. I. it may be noticed that the cylinders are separated from the crankcase by a cast partition, but that there is a fair sized chamber above this division and a 15 mm. slot is made in each vertical web above the main bearings. Therefore, there is an air space of considerable volume between the underside of each piston and the top of the web which separates the cylinders from the crankcase. This has been done to prevent the over-lubrication of the cylinders, the effect of the inter-connecting chambers being to allow the air displaced by the down stroke of one pair of pistons to pass along and rise up under a corresponding ascending pair, without being forced in the meanwhile to traverse the oil-laden atmosphere of

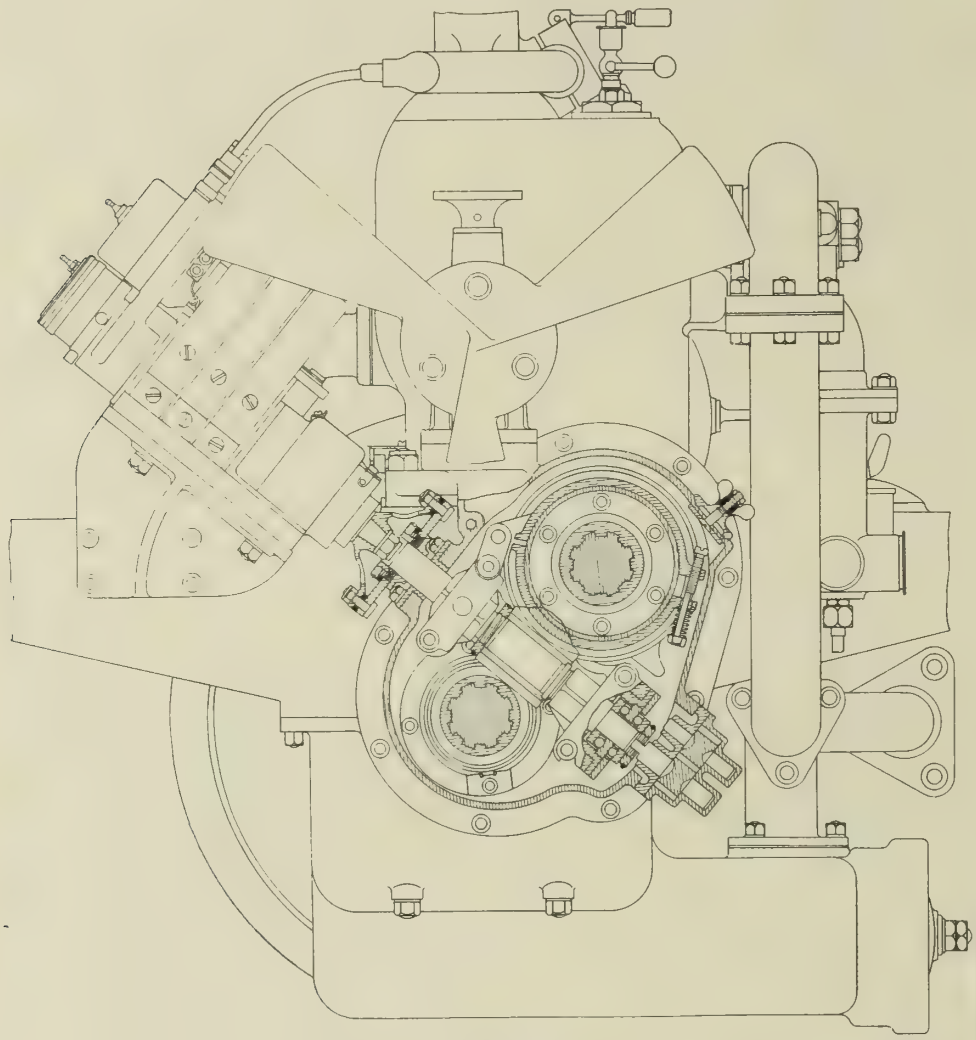


Fig. II.

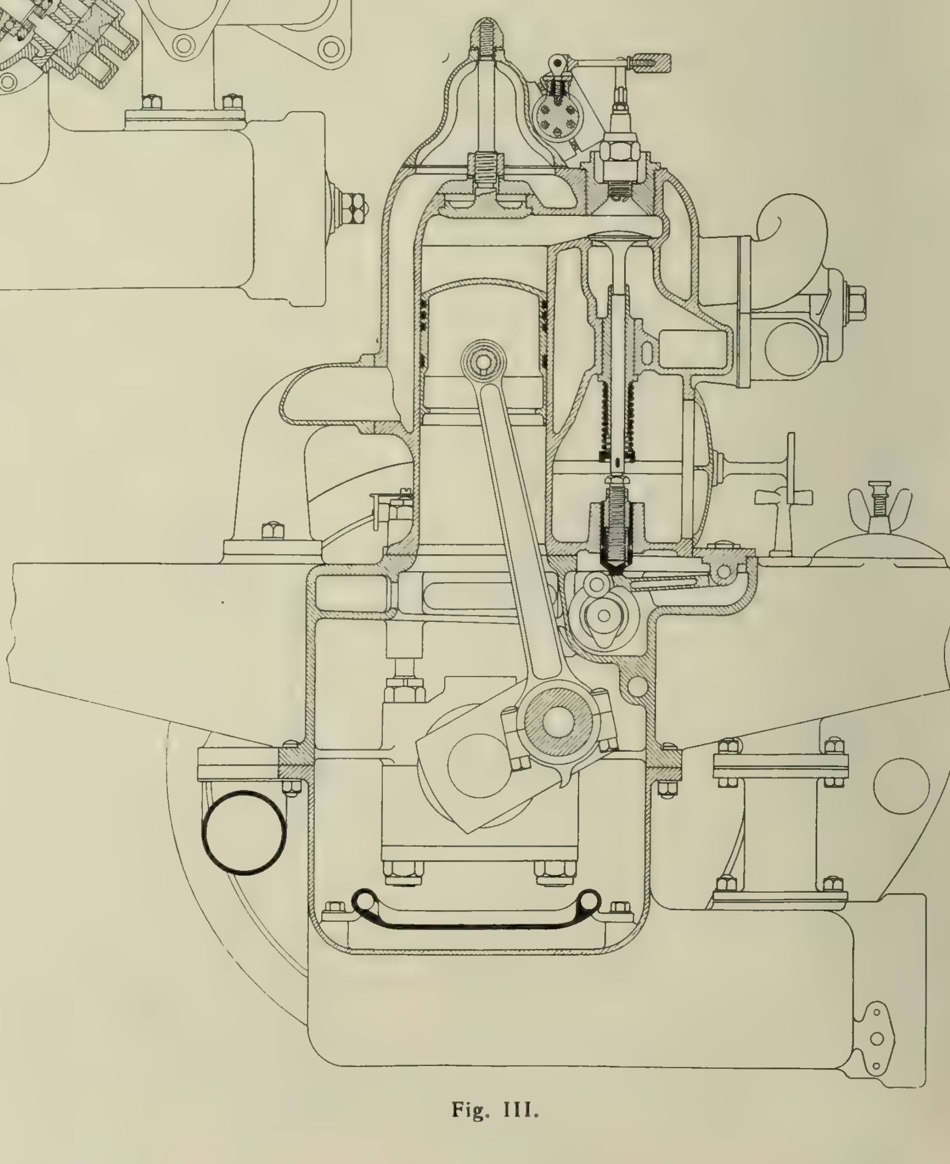


Fig. III.

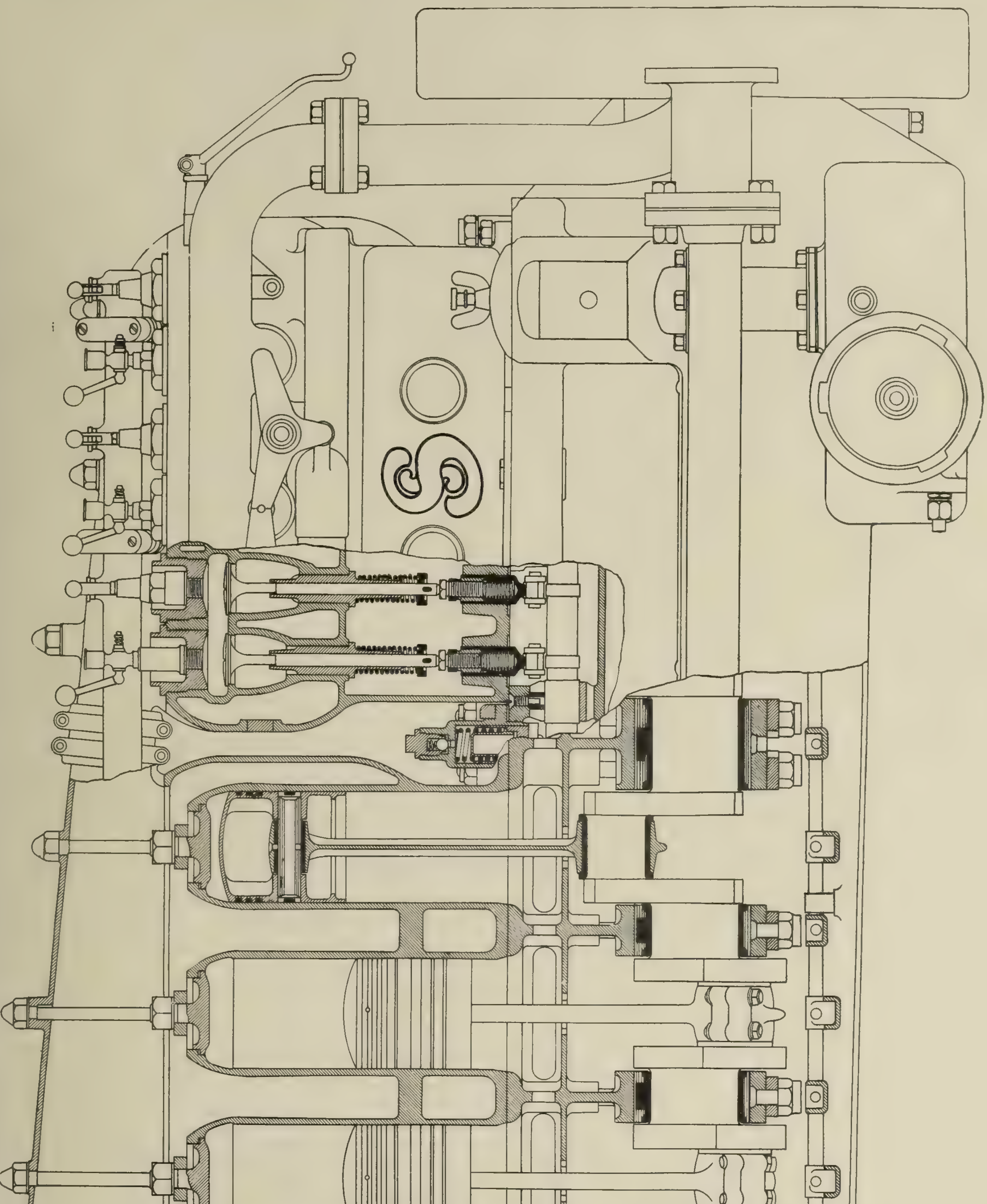


Fig. 1.

THE 25 H.P. SHEFFIELD SIMPLEX ENGINE.

the crankcase proper. It is claimed that this device has completely done away with smoking troubles, while allowing a copious supply of lubricant to be delivered to the main bearings and the big end troughs. Of course the chambering adds a trifle to the height of the engine, but this may be considered much more as an advantage than otherwise, because it makes for accessibility.

The oil pump is situated external to the crankcase, being driven from the end of the diagonal shaft, intermediary between the crankshaft and the camshaft, and it

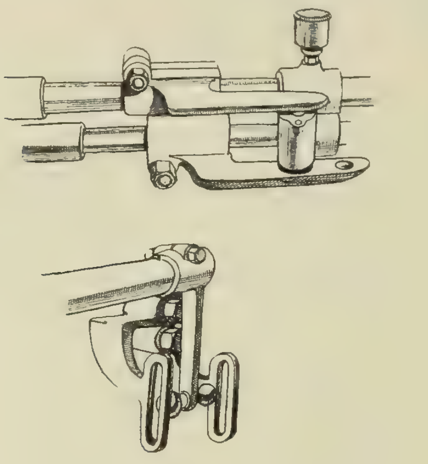


Fig. VII.

draws its supply from a deep sump at the flywheel end of the engine. The delivery is through a pipe supplying each of the main bearings, and this may be seen cast-in just beneath the camshaft in Fig. II Overflow from these bearings drips into gutters which stand at a higher level than the big end troughs, but are connected to them by another pipe as can also be seen in Fig. I. The filling cap is situated on one of the base chamber arms, and it consists of a large hinged lid with a cam fixing, the handle for which may be seen just behind the valve cover nut in Fig. II. There is an ordinary level tap, but the filter is exceptionally big, and is retained by an easily adjustable cover, the method of working which can be seen from Figs. I. and II. For the camshaft there is a separate oil bath with a separate filling cap. For the crankshaft white metal bearings are used throughout, those for the big ends only being cast directly in the connecting rods, the crankshaft bushes all being white metal dovetailed into brass. At the forward end of the crankshaft there is a small ball thrust partly to take the pressure from the skew gear and partly to resist any thrust due to the clutch when held out of engagement.

Of course, not the least interesting part of the whole engine is the means adopted for driving the camshaft because, although this is not new, still the fact that it is being retained in an unaltered form is proof of its satisfactory nature. Fig. III. shows the diagram of the timing gears and the details of the bearings for the intermediate pinion or worm, while it will be noticeable that there is a brake on the end of the camshaft consisting of two shoes, kept pressing against the drum formed integrally with the large timing wheel, by means of a light spring. The purpose of this brake is to apply a resisting torque to the camshaft so as to maintain contact between the teeth of the timing gears continuously, and certainly the engine is totally devoid of camshaft chatter, there being no particular speed at which the valves become very audible, as is often the case with high speed engines. Also, to avoid side thrust on the tappet guides, the cam acts on a roller carried by a lever 90 mm. long, and there is a cushioning spring inside the tappet to maintain contact throughout this portion of the gear. There ought therefore never to be any play between the crankshaft and any one of the valves.

The camshaft, too, is like the crankshaft, thoroughly well supported by ample bearings, and it is separated from the base chamber. Situated at the centre of the shaft there is an air pump for supplying tank pressure, this being of the ordinary spring plunger type cam driven, and with ball valve gear. Returning, however, to Fig. III. and the camshaft drive, the attachment of the oil pump is shown well and the adjusting ring for the thrust bearing. At the upper end the bearing is plain, and the shaft carries a leather disc at its top extremity, this being gripped tightly between flanges. From this leather the magneto coupling is made through a pair of discs with differential holes and adjustment can be performed with great delicacy.

The valves themselves are of 40 mm. effective diameter and of good shape, the diameter of the stems beneath the head being slightly less than that of the guide bore, to prevent any danger of sticking. The pressed-in guides are of very great length, being 110 mm. from end to end, while the stem diameter is only 9 mm. Care has been taken also to keep down the total weight of each valve, and the inlet passages are of ample area. In Fig. I. one of the dogs is shown which secure the inlet and exhaust pipes, and it will be noticed that this is applied neatly, and that both the pipes are raised sufficiently to render access to the tappets quite simple. All the other details of the engine are made sufficiently clear in the illustration, but one point which might be remarked on especially is the hollow crankshaft, the extent to which this is drilled out being shown in Fig. III. The operation is naturally a costly and troublesome one, but a good many pounds of weight are saved, and the shaft is, of course, a great deal more rigid than it would be if solid and of equivalent weight. Another point which makes for neatness is shown in the plan view of the chassis, being the way in which the cold water from the radiator is carried to the cylinders through a pipe cast solid with the crankcase. The final connection to the cylinders is by an aluminium Y pipe, and by an ordinary union to the radiator.

Turning now to Fig. IV., which shows the plan and elevation of the chassis, it will be seen that there is a disc clutch with a large number of plates (61 as a matter of fact) connecting to the single universal joint carried on the middle cross member. This clutch is exceptionally sweet in action even for its type, and this is probably to be accounted for by the fact that the total area of contact is considerably more than normal, so that the pressure needs only to be light. It may be recalled that the Lanchester company were amongst the first firms to adopt the plate clutch, and that precisely the same scheme was followed by them, namely, to have many contact surfaces and only quite moderate pressure. It will be noticed that the clutch shaft carries the inner member and is spigotted in the flywheel, wherefor the good behaviour of the clutch is not to be accounted for by unusual accuracy in alignment.

The universal joint has already been described in these columns and, as it has been unaltered, there is no need to deal with it in detail now. Undoubtedly it is mounted in one of the few best ways, being completely protected and also relieved of all tortional stresses. The method by which the ball end is attached to the propeller shaft casing by means of a split ring inside a retaining nut is neat and effective, and the hollow propellor shaft itself is another example of the way in which money has not been spared in the production of this chassis.

Turning to the next illustrations, Figs. V. and VI., showing vertical and horizontal sections through the gearbox and axle, the casting will probably appear to be light, especially when considering that it is of aluminium. However, the vertical section shows that this is more apparent than real as there is plenty of stiff webbing internally, the spigot bearing of the bevel pinion being particularly well supported, and possibly stress on the casing is lessened to some extent by the universal jointing of the inner ends of the driving shafts to the differential cage. Somewhat curiously, the conical sleeves are quite unusually strong, and the road wheels are supported with commendable rigidity. Not only are the main journals well spaced, but the easily adjustable double thrust bearings relieve them of all accidental stressing, as is the case with the front hubs also (see Fig. VI.) Regarding the brakes, these follow ordinary practice with regard to the shoes, but both sets are expanded and contracted by toggles so that the action is positive in both directions, although it is, of course, spring assisted in the usual manner. Owing to the large diameter, the brakes are very smooth and possess ample stopping power, despite the fact that neither is geared up.

For the gears no particular description is needed, their arrangement being

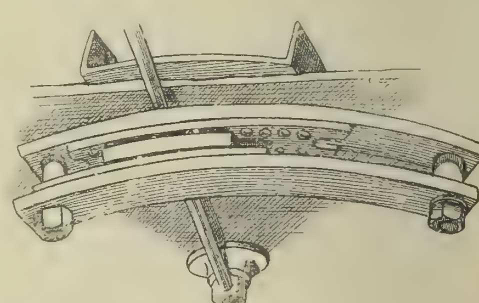


Fig. IX.

rendered perfectly clear by the drawings. The hollow shafts and the large bearings are noteworthy features, and the change speed control may be observed in the plan view of Fig. IV. Details of the striking gear are also shown in sketch, Fig. VII., and the stepped quadrant also in Fig. IV. Another neat feature to be observed in the latter view is the way in which the brake rods are carried through the front end hangers of the rear springs, and the compensating gear is thereby carried in a position comparatively easy of access through the

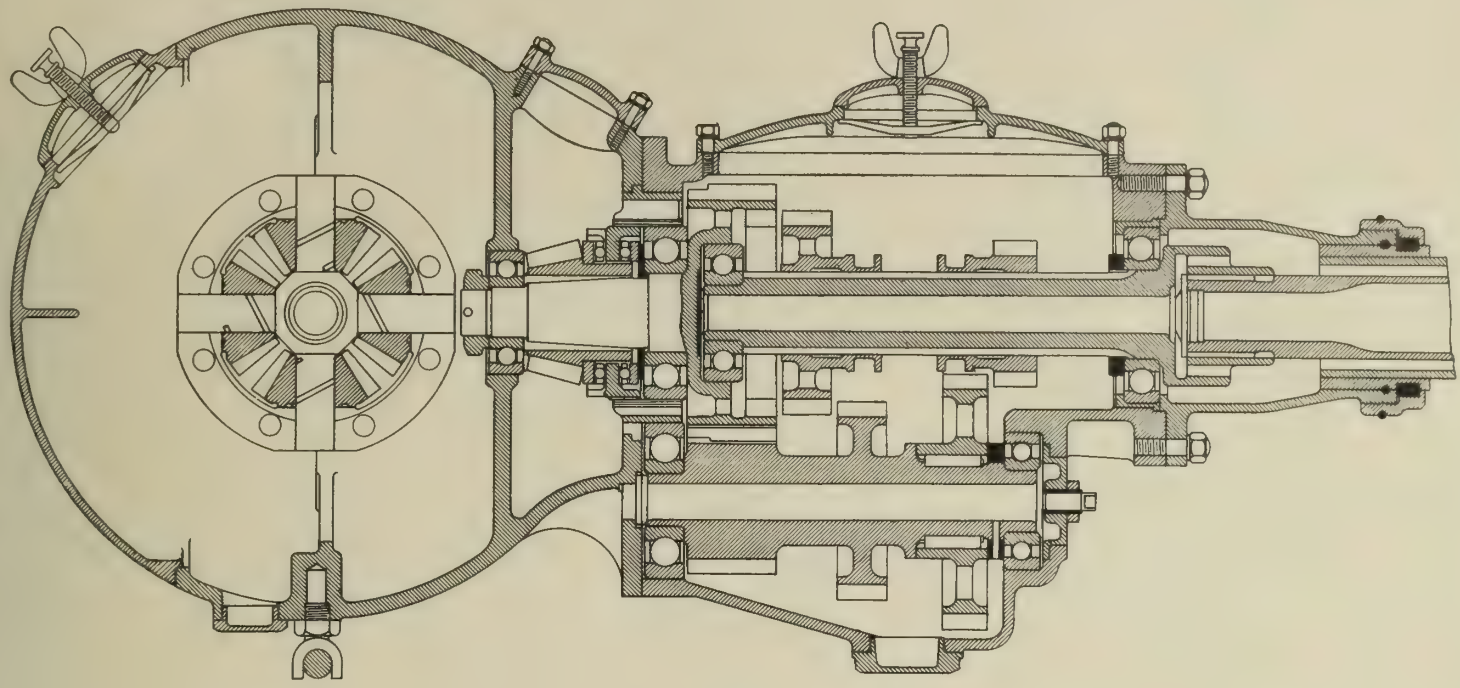


Fig. V.

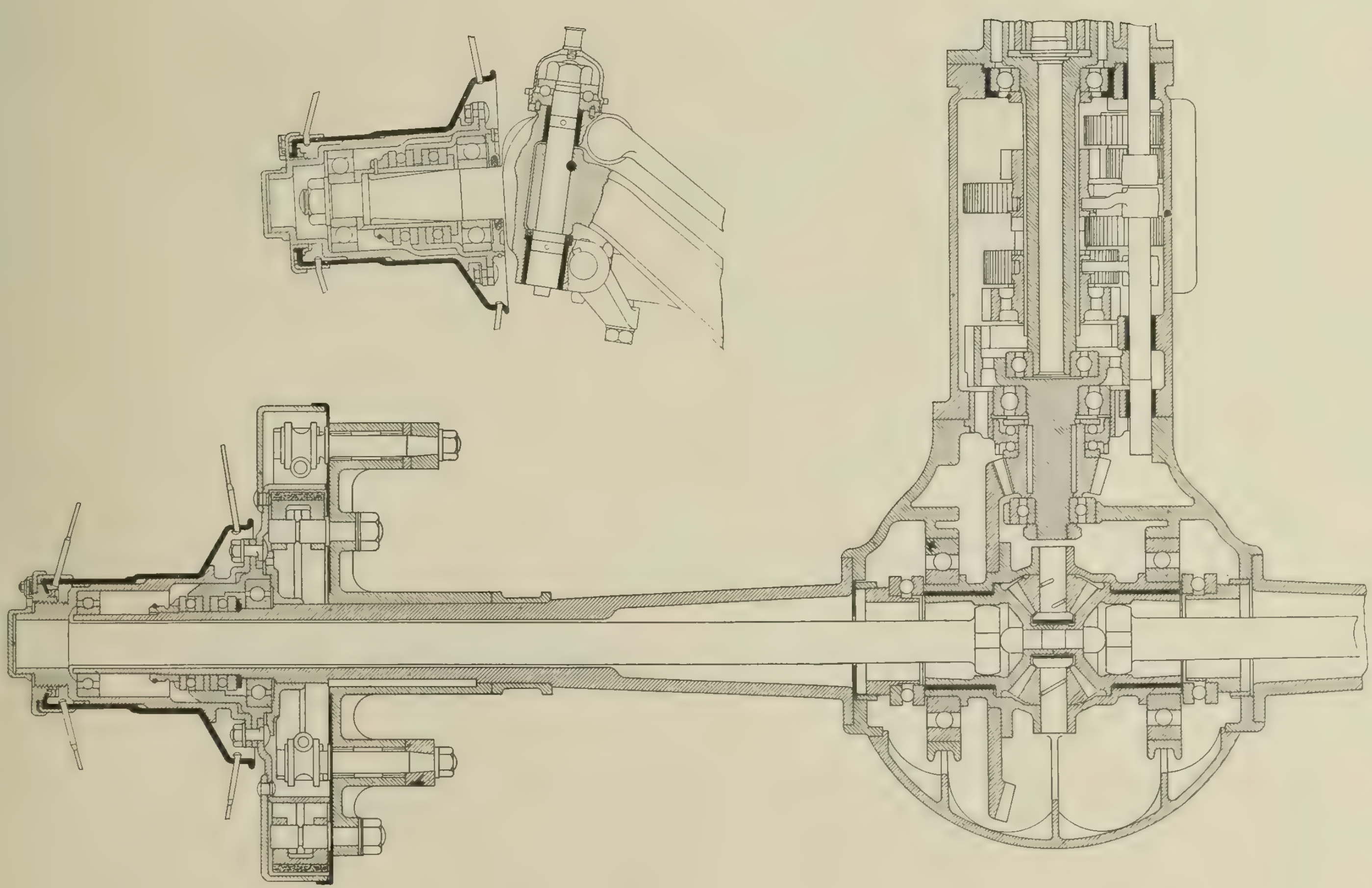


Fig. VI.

TRANSMISSION DETAILS OF THE 25 H.P. 6-CYLINDER SHEFFIELD SIMPLEX.

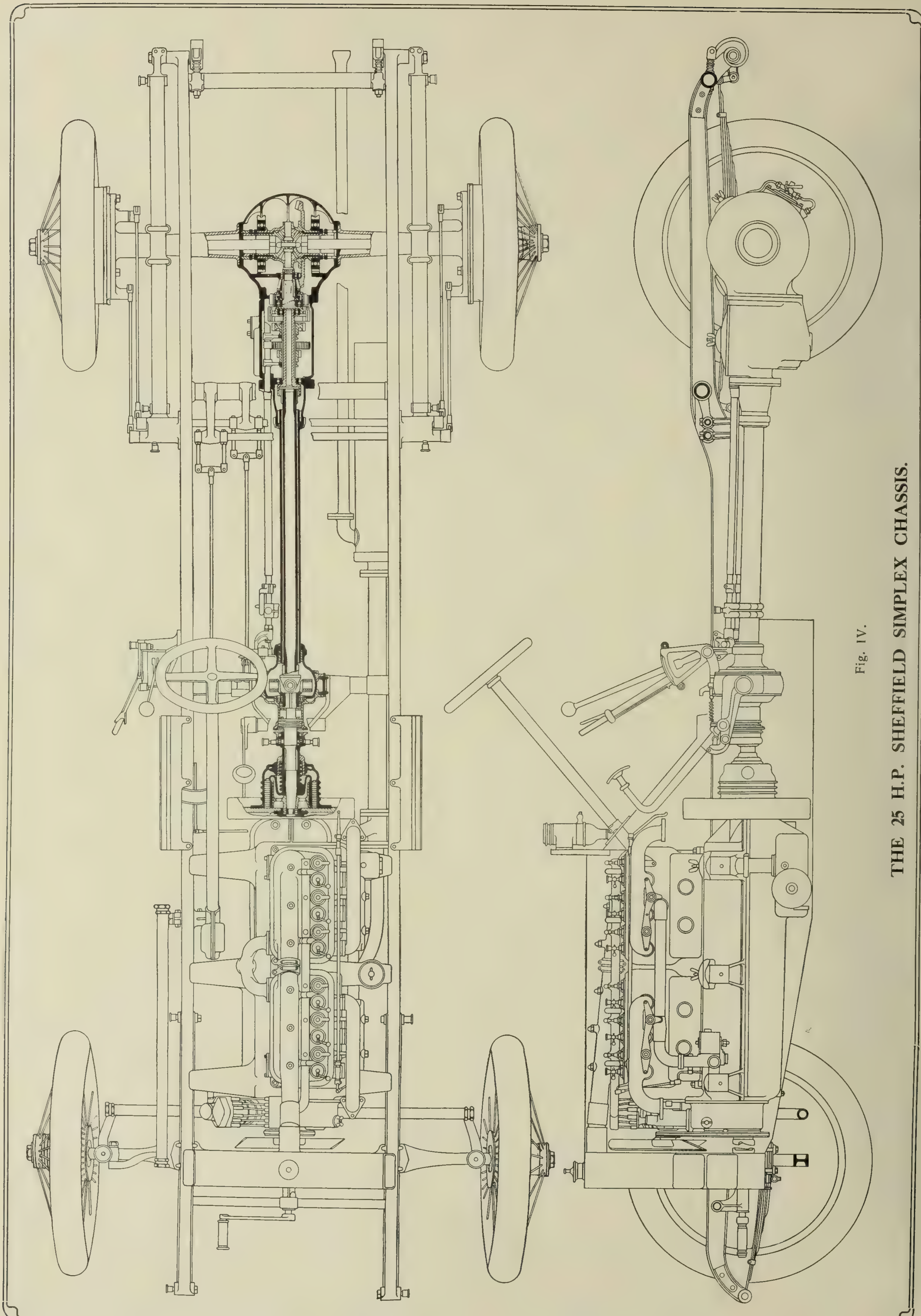


Fig. IV.

THE 25 H.P. SHEFFIELD SIMPLEX CHASSIS.

floor boards. The steering gear is laid out very cleanly, but is otherwise normal, and there is no other main feature of the design which calls for comment, unless it be the method of making up the central cross member with a large malleable iron casting containing the universal joint.

The control of the engine is by throttle pedal only, and this, as is well-known, operates transversely, sliding sideways on a sectional ball race, as is shown in Fig. IX. This system imposes the minimum of muscular strain on the driver, and it is easy to become accustomed to the arrangement. Once its movement has become semi-automatic one finds that the throttle can be kept at any desired degree of opening without any effort whatever, and this can really seldom be said of the ordinary accelerator pedal. One feels, however, tempted to suggest that the same free movement could be obtained by the use of a much more simple roller mounting, much cheaper, and quite as effective as the segmental ball race.

A possibly debatable point in the control is the combination of clutch pedal and brake, because, in order to apply the foot brake, it is necessary first to compress the clutch spring to its full extent. The clutch is, however, controlled from the pedal through a cam so that the pressure of the clutch spring is entirely relieved before the brake comes into opera-

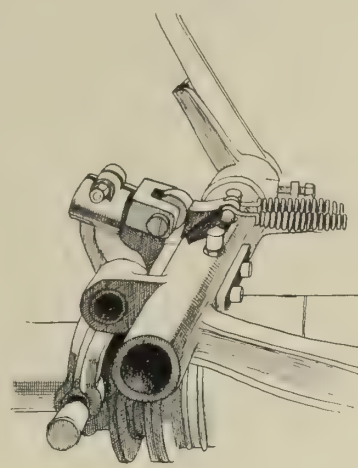


Fig. X.

tion. Thus, when actually applying the brake, only a very small proportion of the clutch spring pressure is still being resisted by the foot. This arrangement is shown in the sketch, Fig. X. Although we have not had the privilege of examining the horse power curve for any one of these engines, there can be little doubt that the power is rather exceptional for the size of engine at low speeds of revolution, and this is probably to be accounted for by the very great care which is taken in the fitting of every detail. At present the standard carburettor employed is the Zenith, but this is in itself scarcely sufficient to account for the unusual flexibility. For completeness in detail fittings this chassis may certainly be placed amongst quite a small number, and it gives the impression of being designed by men who have had considerable road experience. Geared at approximately 3/1/3rd to one on the top speed, with 880 mm. tyres, it possesses a good turn of speed. The springing is very good indeed, the Lever spring suspension shown in the chassis view being a standard attachment. Considering the size of the engine the climbing power on the top speed is certainly remarkable, but the most striking feature of the car on the road is undoubtedly its smoothness, which is exceptional even for a good six-cylinder. It certainly appears that the great rigidity of the engine has a very real and beneficial effect on the running.

CONDITIONS OF SUCCESS FOR CHAIN-DRIVEN CAMSHAFTS.

By Charles S. Renold.

THE silent chain drive for automobile engine camshafts and magnetos may now be said to have passed its first experimental stage and to have proved that, if certain conditions are complied with, it can be made successful. Exactly what those conditions are remains at present a matter for some controversy.

There are three sides to the question, three points of view to be considered: that of the builder and designer of the engine, that of the maker of the chain; and lastly, that of the man who buys the car. Now engine builders, in adapting chain drives to their camshafts and magnetos were inclined, naturally enough, to follow the line of least resistance. That is, they considered nothing more to be necessary than to substitute chain wheels for spur wheels, and after making the necessary alteration in centre distance, to put on the chain and leave it alone. The chain maker, however, knows that this is not sufficient. His interests moreover lie in the same direction as those of that third party, the owner of the car, for they both desire the chain to be not only successful at first, but to remain so. Freedom from breakdown, long life, ease of replacement—these desiderata touch both equally. It is to put the chain maker's point of view, and incidentally, that of the user of the car, that this article is intended. We wish to set out in a reasoned argument the conditions which must be observed for chains to be successful, to explain what these conditions are, and how the need for their consideration arises.

Now the chief point where the interests of the engine builder and the chain maker do not apparently coincide is the question of adjustment. At first sight it appears very much easier to arrange for a drive with fixed centres, but even here the advantage is more apparent than real, since such a drive requires two chains and four wheels set at pre-determined centre distances, instead of possibly one chain and three wheels, in which the position is conveniently adaptable to the other requirements of the design, and may be that previously used for spur gearing. Beyond this, however, the non-adjustable drive for camshafts or magnetos suffers from grave drawbacks, the nature of which it is our business to understand.

It may be stated at once that the arguments against the fixed centre drive are based largely on the defects of present-day chains, and their force cannot be judged on *á priori* grounds, but only after a close consideration of the defects and their magnitude. Such a consideration will show that as things are at present, these defects are unavoidable. It is, however, possible that such improvements may in the near future be made in silent chains that the arguments may be robbed of some of their force, but it is doubtful whether the scale will ever be turned in favour of the non-adjustable drive.

The Fixed Centre Drive.

We will now examine the conditions under which the fixed centre drive works, and the disadvantages from which it suffers, leaving consideration of the adjustable drive until later. A camshaft drive, especially with poppet valves, is an impulsive load, and the same is true of the magneto. The motion of the crankshaft, which drives the camshaft, is also likely to be impulsive, particularly in the event of any irregular firing. In addition to this, it must not be forgotten that a chain drive on a car, subjected to every kind of vibration, cannot be regarded as working under the same conditions as a chain on a fixed piece of machinery. In short, chains used for camshaft and magneto driving are subject to innumerable impulses, the most severe being those caused by the nature of the load. Now the effect of these impulses, unless the chain is fairly tight, is to produce *whipping* in the chain, which throws a much greater load than that of the power pull on the chain, and consequently causes it to wear rapidly, thus further lengthening it, and giving still greater encouragement to whipping, which is thus intensified, the effect being cumulative and of increasing rapidity, until finally the chain breaks, although the driving pull is much less than the breaking strength of the chain.

A further disadvantage of this type of drive is that slack in the chain can act as back-lash and allow variable timing of the valves or ignition. We are inclined to think, however, that this second disadvantage, important as it undoubtedly is in theory, is in practice overshadowed by the first one, i.e., the risk of chains wearing out quickly and breaking through whipping. It will be time enough to consider the second trouble when the first has been cured.

From the foregoing arguments it will appear that since so small an amount of sag causes serious whipping, the chain therefore, ought to begin its life either with no sag at all, or with the very minimum. This brings us to the question—how much slack, if any, is allowable in the drive to start with, and at what point does the permissible quantity become exceeded? As will be pointed out later, a slight amount of play in the chain is necessary for satisfactory running; which

may be put as being equal to a sag of $\frac{1}{8}$ in., this representing the necessary minimum. At the other extreme, judging by drives which we have measured, we find that with a centre distance of about 5 in., and wheels having, say, 19 teeth and 38 teeth, not more than $\frac{5}{8}$ in. sag is allowable. This means that it should not be possible to press the centre of the slack side of the chain more than $\frac{5}{8}$ in. away from the straight chain line, *i.e.*, the tangent to the pitch circles of the two wheels. With a chain of $\frac{1}{2}$ in. pitch, 50 pitches long, this represents an approximate elongation of .4 in., equal to .008 in. per pitch, or about $1\frac{1}{2}$%.

Now for an ordinary silent chain drive it is possible to use the chain until it has developed about 4% stretch, and even allowing for the severe conditions and the accuracy of gearing required for camshaft driving, it would still be possible to use the chain until it had about 3% stretch. From the above figures it will be seen that owing to the whipping and consequent risk of breakage, the chain has to be discarded when it has undergone only about half of the wear which is in it, and even from this residuum of life certain deductions must be made, as we are about to show. First, we would point out that as whipping increases, the *rate* of wear increases, due to the extra load produced. If whipping can be eliminated, not only would double the *wear* in the chain be available, but this would represent considerably more than double the amount of *life*, since with no whipping the wear would always be at the slowest rate. Secondly, having shown how far the wear may be allowed to proceed before the chain must be discarded, we must draw attention to the fact that not even all this amount of wear is available for chain life. Two facts are responsible for this, the effect of both being to reduce the available range of sag.

The first fact is that variations have to be allowed for in any machined article, and from the nature of the case we are considering, these must all be from that dimension at which the chain would join up with the minimum amount of slack, to something larger. By working to the finest possible limits, we find the variations in the total length of a chain may be equivalent to a difference of .001 in. per pitch from the shortest to the longest.

The second fact is, as stated above, that for satisfactory running at the high speed required, a slight amount of play in the chain is necessary even to start with, or no oil can enter or remain between the bearing surfaces, and galling or seizing will result. To give this play, the chain must have at least .125 in. of sag on the slack side. If the full amount of the manufacturing limit of .001 in. per pitch be taken advantage of, this sag will be further increased to .25 in. It will be remembered that the outside limit of sag allowable, as before stated, we put at .625 in., a limit which is reached when the chain is normally only half worn out. If, then, we make the necessary deduction of .125 for necessary sag, and the possible deduction of another .125 in. for manufacturing inaccuracy, it will be seen how little of the chain's possible life may be available.

Moreover, the most useful part of the chain's life, viz., when the rate of wear is slowest, due to the tension being correct, may be wasted where chains are on the long side, even though they are still within the manufacturing limit of accuracy.

The obvious comment on the foregoing arguments is that the limit of .001 in. per pitch is not fine enough, since if this limit were finer, less allowance need be made for manufacturing inaccuracy, i.e., the outside limit of $\frac{5}{8}$ in. sag would not be reached until the chain had worn to it, instead of being reached earlier through the accident of inaccurate manufacture.

Chains of "Dead Pitch."

The desirability of finer limits of accuracy brings us to the consideration of the possibility of producing chains of "dead length," which would all join up at exactly the right tension to start with.

Several methods have been suggested to us :—

(1) Make a chain to "dead size" with no variations. This obviously does not emanate from engineers, and requires no discussion. The limits within which we *can* hold the variations are dealt with fully later.

(2) Make the variations on the short side, and join up the chain by force. This can be done, but apart from the desirability of a slight amount of play in the chain, it means that every chain plate has to be sprung, which destroys the accuracy of the angles of the gearing faces, makes the chain run "hard" and hot, and more noisily than necessary. It is liable to stress the material of the plates across the arched back above its elastic limit, which distortion of the links will cause uneven running, since the chain combinations are of unequal strength, and each alternate pitch will pull out more than the others. Special appliances would be needed to draw the chain ends together when mounting, and replacement chains would, therefore, be very difficult to mount, except at the works of the car makers. It is an unmechanical proceeding, and puts unnecessary loads on the shaft bearings, chain studs and wheel teeth.

(3) Make the chain short in pitch and "run it in" to exact pitch under load. This is based on a misconception as to what "running in" does. It is found that during the first few hours of a chain's working life the wear is more rapid than later on. This rapid initial wear is due to bedding down of the stud and bush surfaces, and if "running in" were done at all, it should obviously be carried to the point where the bedding down is complete and the legitimate wear begins. If the chain, however, varies in its manufactured length, it cannot satisfactorily be brought to finer limits of variation by such a "running in" process, as the amount of increased length will in itself be variable. Such "running in," therefore, does not help to produce chains of "dead pitch" such as are needed for drives without adjustment.

It is of course conceivable that each chain could be "run in" until it reached a certain pre-determined length, but this would lead to one of two difficulties :—First, if the "bedding down" is completed before the chain has reached the predetermined length, it would be quite impracticable, commercially, to carry the process further, as after "bedding down" is complete the rate of wear is extremely slow. Thus it will take two to three hours to elongate the chain .001 in. per pitch, but it will take twenty to thirty hours to produce a further .001 in. per pitch increase. In the second case, if the chain reaches the standard length before the "bedding down" is complete, the gain due to "running in" will be almost immediately lost when the chain is set to work. "Running in," therefore, even if it were otherwise advantageous, would not save the fixed centre drive.

(4) Cut every set of wheels to suit its chain after it has been "run in." This could be done, but would only take care of pitch variations in the portions of the chain that lay on the wheels. It would not cure the variations of the parts of the chain between the wheels, and replacements would be almost impossible.

We come to the conclusion, therefore, that we must give up the idea of obtaining a chain of "dead pitch," and be content to work to the closest limits possible.

Elements of Initial Accuracy.

The limit of .001 in. per pitch, which we have stated, may not sound very fine in these days of fine grinding, etc., but let us consider on how many elements the accuracy depends :—

(1) Chain pitch, i.e., distance between centres of rivet.
(2) The angle of link face.
(3) Distance from angle of link face to centre of rivet.

The most important of these is chain pitch.

Accuracy of Chain Pitch.

We have already mentioned what limit of pitch accuracy we are at present able to work to, and in order to explain the difficulty of diminishing this limit, we will consider what a pitch element consists of viz.,

One link plate.
Two bushes.
Two studs.
Two clearances between stud and bush.

Variations may occur in the machined dimensions of each piece as follows :—

(*a*) *Variations of plate.*
 1. Pitch of holes.
 2. Diameter of holes (affecting force fit of bushes. See paragraph *d*).

(*b*) *Variations of bushes.*
 1. Outside diameter. (See force fits. paragraph *d*).
 2. Inside diameter (affects clearance between stud and bush).
 3. Eccentricity of hole (affects pitch).
 4. Variable hardness (affects the closing in of the bush when forced into the plate.

(*c*) *Variations of stud.*
 1. Outside diameter affects clearance in bush).
 2. Straightness after hardening (affects clearance in bushes of a combination of link plates. See paragraph *e*).

(*d*) *Variations introduced by forcing bushes into place.*
 1. The forcing fit may be too great,

due to—
Small holes in link.
Large bush.
Stiff bushes, which resist closing in, due to abnormal hardness or small inside diameter.

In these cases the bush may be closed in more than intended, thus reducing clearance, or the link may be distorted, opening out and increasing the pitch of the holes.

2. The forcing may be too slack, due to—
Large hole in link.
Small bush.

The bush does not close in as much as intended and clearance is too large.

(e) *Variations of clearance between studs and bushes.*

1. Due to variable size of hole in bush after assembling. (See paragraph *d.*).
2. Diameter of stud.
3. Straightness of stud.

The non-straightness of a stud has an effect in shortening the pitch of a combination of plates by absorbing some of the clearances. This is why wide chains are always shorter in pitch than narrow ones made of the same parts.

From the above list of possible variations in the chain pitch, it will be seen that to obtain a final accuracy to within .0005 in. on either side of the normal requires very fine limits indeed for the individual processes.

To sum up the case against the fixed centre drive—to be satisfactory, the chain must start life, not indeed rigidly tight, but with a very definite minimum of slack. At present chains cannot be guaranteed accurate enough for this. If, on the other hand, each drive is treated specially, *i.e.*, special cutting of wheels, special "running in," or special selecting of chains and wheels to suit each other, then neither the chains nor the wheels are truly interchangeable. Further, the chain must be discarded when a certain very moderate amount of sag has occurred. From this allowance moreover, the aforementioned slight initial slackness, which is necessary for a sweet-running drive, must be deducted, while the possibility of the chain being slightly "long" in pitch, owing to manufacturing inaccuracy, may mean a further deduction. In a word, lack of adjustment almost inevitably shortens the life of the chain by roughly one-half, and may reduce it even further.

The Effect of Adjustment.

From what we have said, it will be seen that two conditions are necessary for a camshaft or magneto drive to be satisfactory :—

(1) The chain must have a minimum of slack.
(2) The chain must fit the wheels.

Both of these depend in greater or less degree on the "Elements of Accuracy" we have discussed. We have shown that at present it is not practicable to produce chains which will always join up at the right tension at a given centre distance, but as a given amount of inaccuracy in the chain has a greater effect on chain slackness than it has on the fit of the chain on the wheel, it is quite possible to guarantee that all chains shall fit all wheels. This meets the second condition, and, if the first is met by adjustment of the centres, we have all the requirements of a perfect drive. Some of the advantages that result are as follows :—

(1) Risk of chain breakage is eliminated with the elimination of whipping.
(2) Both chain and wheels can easily be made interchangeable.
(3) The chain can be used for the whole of its possible life.
(4) The average rate of wear of the chain is slower, since each successive step in the wearing occurs under equally good conditions as regards tension. The wear occurs, so to speak, in arithmetic instead of geometric progression, as it would were no adjustment provided.
(5) By making only a certain amount of adjustment possible, the user of the car can be warned when the chain is nearing the end of its life.
(6) The chain is easy to handle, since no pulling or straining is needed to get it on the wheels, the correct tension being put on after it is joined up.
(7) The width of the gearcase can generally be made less than is necessary for drives without adjustment. In the "triangular" drive there is only one chain instead of two running abreast.
(8) The position of the camshaft need not be altered to a special centre distance, so chain wheels can replace gear wheels.
(9) The chain is "run in," i.e., the bedding down process is completed—on its own wheels during engine tests, and can then be taken up to the right tension before legitimate wear begins.

It is too early in the day to give any very general rule, but from what we have seen of adjustable drives we believe it will be found, if the chain is run during engine tests and then adjusted, no further adjustment will be necessary for many thousands of car miles, probably not until the car is sent in for overhauling. In fact, so far as the user of the car is concerned, he need trouble no more over adjustment than if none were provided.

Adjustment and Interchangeability.

Taking into consideration what we have said on the subject of initial accuracy, it will be seen that the provision of adjustment is of as much importance for starting a new chain at the right tension as it is for taking up future wear. This has an important bearing on the question of interchangeability, a point of considerable interest to the user of the car, for, even if the original chain had been put on at the right tension with fixed centres, the task of finding a second chain to join up exactly right at the same centres is well-nigh impossible for the private individual.

Method of Obtaining Adjustment.

It is of course impracticable to alter the relative position of the camshaft and crankshaft. There seem to be, therefore, only two possible methods of adjusting chain tension. The first, and most obvious, is to arrange an adjustable idler wheel pressing against the chain. With the silent chain such a wheel has to gear with it, and be in contact with at least three teeth; wheels bearing on the back of this type of chain being quite unsuitable. If only the camshaft has to be driven by chain, such an idler wheel would be the only way of providing adjustment, and would be quite satisfactory. If, however, the magneto is to be driven by chain also, a second method for obtaining adjustment can be adopted, by allowing the wheel on the magneto to take the place of the idler wheel, and making the magneto adjustable bodily. If there are two camshafts and a magneto, two triangular drives should be arranged, one adjusted by the magneto, the other by an idler. The fan or pump might possibly be arranged to be driven from the spindle of the latter.

It would generally not be advisable to connect two camshafts and the crankshaft by a single chain, as it would be difficult to arrange an adjusting wheel, either magneto or idler, that would not leave too small an arc of contact for the chain on one or other of the main wheels. If the disposition of the shafts made this possible, however, there is no objection to it.

It may be objected that a triangular drive puts unnecessary loads on the bearings of one or other of the driven shafts, according to the arrangement. This does not seem to have produced any trouble in any drive I have seen or heard of. Moreover, I would point out that any extra load of this sort, if it does occur, is likely to be much less severe than the sudden irregular loads, due to the whipping of a worn chain on a fixed centre drive.

Conclusion.

I have now put before my readers the full examination of the case as far as lay in my power. I cannot help feeling that the result forms an unanswerable argument in favour of adjustment as the chief condition necessary to successful camshaft driving by chain. Apart altogether from the question of "Initial Accuracy," into which we have gone fairly deeply, the simple fact remains that to deny the need for adjustment is to deny that chains elongate under wear. This denial cannot be made, nor can the ensuing consequence, namely whipping, be denied, nor the final consequence, namely breaking, to avert which the chain must be discarded when perhaps less than half-worn and certainly long before such a need ought to have arisen.

EDITORIAL NOTE AND ANOTHER OPINION.

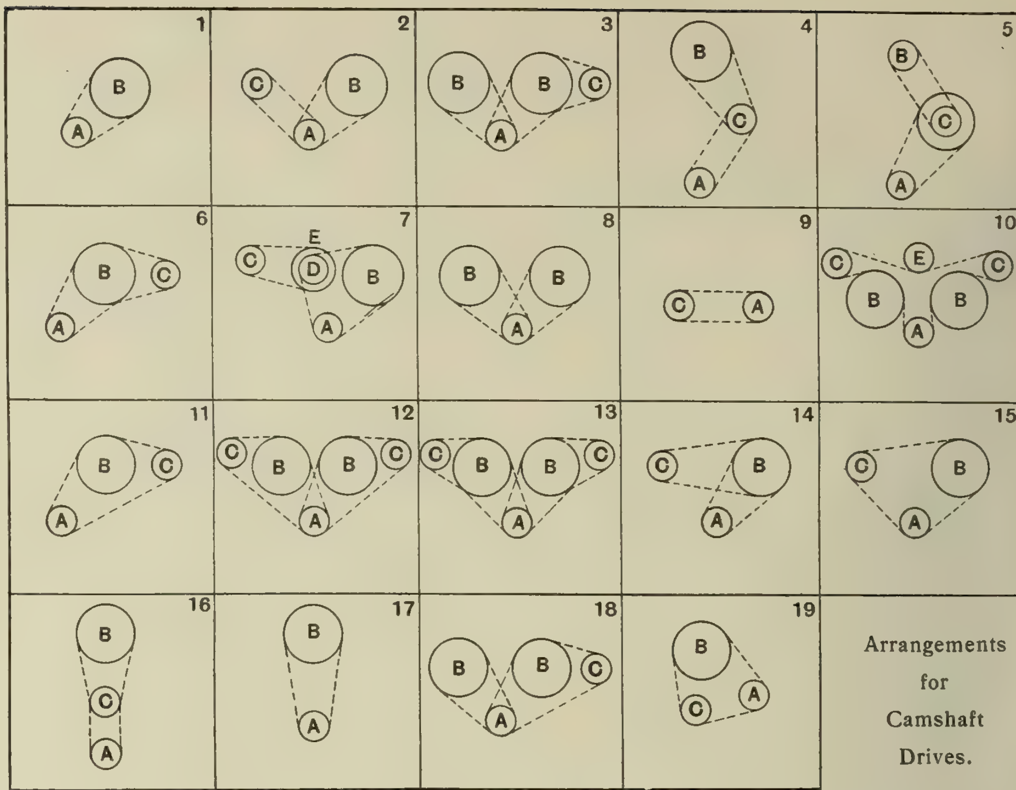
Our contributor, in this article, makes out a very good and clear case for the necessity for chain adjustment, and it is interesting to note that the Coventry Chain Company are of much the same *general* opinion as Mr. Renold, though they are much less emphatic. This question of adjustment for chain camshaft drives is of especial importance, because, if a number of engines are put on the market with drives which will not prove satisfactory, the chain may get a bad name for no fault of its own, and a real improvement languish into obscurity for want of proper understanding. In

an affair of this nature the opinions of chain makers are to be heeded, because chain making is so peculiar a branch of engineering. The Coventry Chain Company's views are as follows :—

The necessity for adjustment of chain wheel centres depends largely upon the nature of the drive and the type of chain employed, but whenever possible it is always an advantage to provide for this adjustment. With shop and general engineering drives running at comparatively long centres the sag in the chain will allow of the necessary adjustment by the removal of one pitch from the chain. On well-proportioned drives having short centres the type of chain used will influence the necessity for adjustment. With roller chains such adjustment should always be provided, but with chains of the inverted tooth type it will be seen that the elongation in the major portion of the chain is rectified by the chain's action in climbing up the sides of the teeth, so that only the stretch in the free portion of the chain has to be considered. As this portion of the chain is comparatively short compared with the whole length of the chain very little slackness will be apparent in this type of chain after considerable wear has taken place in the chain.

With camshaft drives of reasonable proportions, experience leads us to believe that adjustment of the centres of the wheels is not necessary, but that with auxiliary drives, such as that with the magneto and pump where the free portion of the chain is considerably longer than that in contact with the wheels, there can be little doubt that the adjustment of the centre distance with these drives is always advisable.

The acompanying illustration shows some arrangements of camshaft and auxiliary drives in use at present :—



Arrangements for Camshaft Drives.

1. This is the most usual type employed, and is by a single chain from the crankshaft to one camshaft with fixed centres for the wheels. In this case the magneto, pump or fan, are generally driven off the camshaft by gearing.

2. This shows two drives from the crankshaft pinion, the wheel B being on the camshaft, and wheel C being on the pump or magneto shaft.

3 This shows the drives for two camshafts on which are mounted wheels B, and a drive from one of the camshaft wheels to the magneto or pump wheels C, which may be driven from the camshaft.

4. Shows the most satisfactory solution for driving overhead camshafts with chains. The crankshaft pinion A drives the double pinion C, which is generally mounted on the magneto shaft. In all cases lateral adjustment should be provided for this pinion. The pinion C then drives the wheel B mounted on the camshaft.

5. This is similar to type 4, with the exception that the magneto pinion C runs at half the engine speed.

6. This has the magneto pinion C mounted on the same side of the engine as the camshaft in type 2. Lateral adjustment can be provided for the pinion C.

7. The crankshaft pinion A, drives the camshaft wheel B, by means of a chain running over the adjustable wheel B, mounted on a bracket which swings about the centre of C, so that the centres of C and D are fixed, whilst adjustment is provided for the camshaft chain. The wheel E may be of equal size to the wheel C or larger, according to the speed required at C. The adjustable axle of the wheels D and E may also be provided with suitable means for driving a fan.

8. The double crankshaft pinion A, drives by two chains the camshaft wheels B. and B.

9. This drive is by equal wheels, generally for operating pump and magneto and in some cases for driving a camshaft. The centres may be fixed or adjustable for the pump or magneto drives only, but should be fixed for the camshaft drive.

10. This drive, can only be operated by a roller or similar chain, on account of the reverse side of the chain being used to drive some of the wheels. The crankshaft pinion A, drives the camshaft wheels B and the pump and magneto wheels C, in the same direction of rotation as would be obtained with spur gears, this being in contrast to all other similar drives shown on this sheet.

The pinion E is adjustable in a vertical direction to take up the wear in the chain, the other centres being fixed. The very small amount of free chain between the pinions A and B is a decided advantage inasmuch as the regular timing motion of the wheels B is not particularly affected by the wear and elongation in this short length of free chain.

11. The disposition of these wheels is similar to those of type 6, but only one chain is used to drive all three wheels. In this case it is essential that the pinion C should be made adjustable, and that the free length of chain between the pinions A and C should be kept as short as possible.

12. This is a right and left-hand drive similar to type 11, the pump and magneto pinions C always requiring to be made adjustable.

13. The disposition of these wheels is similar to type 12, but two chains are used in place of each of the triangular drives.

14. The wheels shown here are similarly disposed to those shown on type 2, but the wheel B is a double wheel, and drives pinion C, which may be adjustable in a horizontal direction.

15. The wheels in this arrangement are disposed similarly to those shown on types 14 and 2, but are driven by one chain only. This drive cannot be recommended by reason of the small length of contact between the chains and driving pinion A. This drive requires a chain as wide as the two chains required on types 14 and 2. It is imperative that provision be made for the constant adjustment of the pinion C.

16. This drive for overhead camshafts is not as good as that shown in types 4 or 5, all the centres being fixed. The pinion C can be arranged to drive the magneto or fan.

17. This drive should be avoided wherever possible, even if it can be arranged for adjustment of the centres. The tendency of the chain on this drive is to leave the teeth on the lower portion of the pinion A, by reason of the weight of the chain accentuated by the driving effort of the teeth.

18. This is a combination of the drives 11 and 1. The centre C will require to be adjustable.

19. This is one of the few triangular drives that can be expected to give really satisfactory results. With the engine revolving clockwise, the camshaft B is on the left-hand side of the engine, the magneto C being placed below this. Adjustment for the pinion C is necessary, preferably in a vertical direction, and it will be noted that in this case the length of chain in tension between the driving wheel A and the camshaft wheel B is small.

It is assumed that inverted tooth chains are used on all the above drives, except where stated otherwise, and it should be noted that where one chain drives another by means of a double wheel or pinion, that chain will of necessity be wider than the driven chain when the intermediate pinion or wheel operates a magneto, pump, etc.

THE IMPORTANCE OF DETAIL.

Presidential Address by L. A. Legros, M.I. Mech. E. before The Institution of Automobile Engineers, October 11th, 1911.

Prefatory Remarks.

IN the first place, it has been frequently urged that it is desirable that meetings of a less conventional character should be held, meetings at which members would feel less restraint, and at which an extempore discussion should take place on a debatable matter rather than an elaborated discussion of a paper which has taken much time and thought in its preparation and requires a similar expenditure of thought and consideration in the framing of questions and in the direction of criticism in the discussion. Steps have been taken for providing that at least one such extra meeting shall be held during the session at which the proceedings will be of a less formal character, and at which a debate or discussion on a technical matter of general automobile interest will take place. It is to be hoped that these meetings will prove popular with those members who have hitherto refrained from taking part in discussions on the papers read at our regular meetings, whether from feelings of nervousness or from an objection to seeing in print a verbatim report of their spoken words. In this connection, I may add that the remarks even of the most experienced speakers of this institution, are, before they go to press, carefully edited by those members themselves. For this there is good reason, as in public speaking on general topics one frequently and unconsciously repeats oneself, and in speaking on technical subjects it is often more difficult still to order one's thoughts and frame one's sentences in such manner that they are clear without being tautological, whereas in writing, matter and phraseology can be easily re-read and corrected. To those members, therefore, who have felt these reasons to be an obstacle in the way of joining in the discussion, I would emphasize the fact that their fears are really groundless, and that ample opportunity will be afforded to them for putting into what they consider proper shape any remarks which they may make.

A second matter which has been raised is that of holding meetings at centres other than London. This question has been carefully considered by your council, and it does not appear possible to arrange for ordinary meetings outside the London area until such time as our membership shall have increased very considerably. At present we have flourishing graduates' branches in Coventry and Birmingham, in addition to that in London, and these, in their own centres, have held meetings and also organised excursions to works and places of interest in their respective neighbourhoods. I am sorry to say that the attendance at these meetings and excursions has not been so good as it should have been, but all three branches show signs of re-awakening interest, and steps are being considered for assisting this.

An excursion made by the London and Birmingham graduates by invitation of the Coventry centre to the Coventry district was, however, very successful. Numerous works were visited, and from the success of this visit, it has been proposed that, in the summer of next year, a visit of the whole of the membership, including the graduates, shall, if possible, be made to Paris to inspect some of the French automobile factories. It is expected that should this scheme meet with general appreciation, satisfactory arrangements may be come to so that the expense to graduates, who do not expect to see the whole of the sights of the city on the Siene, may be sufficiently low to enable a large number to take part in the excursion.

The question of a summer meeting of the institution to be held in some provincial centre at which papers could be read or debatable subjects discussed, has also been considered by your council, but for various reasons, not least amongst which is the desirability for waiting the result of the "extra meetings," it has not appeared to them advisable at present to depart from the usual procedure in respect of the ordinary meetings of the institution.

Last winter, at the request of Mr. Henry Hess, of Philadelphia, who is now one of our members, a reciprocal arrangement was made between this institution and the American Society of Automobile Engineers, whereby a member of either body finding himself in the country of the other is welcomed as a member of that body pro tem., from which it is hoped that good results may accrue.

In furtherance of this object, when it was learnt by chance that several of the members of the society were proposing to come across for the Olympia Show, your council thought that it would be a good opportunity to show them some hospitality and arrange a friendly meeting, and it was proposed to invite the visitors to a dinner.

For the purpose of entertaining our guests, an ample guarantee fund has been raised among the membership of the institution.

The Influence of Detail on the Development of the Automobile.

In every class of machinery, no matter how well known it may appear to be, either to those engaged in its manufacture or in its use, there are a large number of details of which many of the individuals connected with the construction have usually no knowledge, and of which the user frequently has less, in other words, he believes them to be quite different from what they actually are. From the ignorance of the small boy who thinks the boiler of the locomotive is completely filled with works, to that of the fireman who thinks the discolouration of his gauge cocks is due to the analysis of the metal—having heard that word used and believing it to be one of the ingredients in it—there are many other forms of ignorance which have contributed throughout to cause delay in the development and use of every class of machinery, and even at the present time many of these factors are still at work delaying that progress which might take place more rapidly were greater consideration given to the minutiæ of machinery.

To examine this subject systematically, we shall find that most detail has its origin in design, but that the design is frequently marred in execution, and that the executed work is subjected to abuse by the user and to wear by the conditions under which it works, and it is from the latter end of the story that the cycle of design must recommence, since it must take account of the possibilities involved in use and abuse, and of the certainties involved by wear and tear. The subject, of course, is one which does not admit of being dealt with in general terms for all classes of detail, but it is one which it is easy to illustrate by examples.

To take the first example which comes to hand, that of the steering gear, it is well known that the ordinary Ackermann axle affords a fair compromise for obtaining the intersection of the axes of the front wheels at a point on the axis of the back wheels.

This gear as usually made is fitted with a number of pin joints, all of which are liable to wear, and as wear proceeds, the two front wheels of the vehicle when it is travelling in an approximately straight line, which, after all, represents by far the greater portion of the distance it runs, will, whether the steering bar be in front of or behind the front axle, take up positions such that the horizontal diameters of the steering wheels would intersect behind the car. After a certain stage of wear has taken place, it is thought necessary to put the wheels in gauge again, and most mechanics, if left to themselves, would set the wheels properly and truly parallel. At this point we should ask ourselves whether this is the right thing to do; whether making things "exactly right" is after all the proper course to adopt, and whether the present example is not an illustration of a distinct advantage to be gained by making adjustments incorrectly in the first instance. If the limit for error in parallelism, determined by experience based on the wear of tyres, is half a degree (or, in the language of the shops, the wheels should not be more than one quarter of an inch out of parallel in a length equal to their diameter), and if, when this error has been reached, it is time to put them right, then why not set them half a degree (or a quarter of an inch) inwards to start with, so that they will start with a negative error no greater than the positive error permissible, and thus double the life will be given to the steering before it becomes necessary to take it up, assuming that want of parallelism is the only reason for the taking up?

Take another case: the bearings of an engine are made and fitted so that there is no shake or knock when the engine is turned round without the ordinary amount of lubricant. Such an engine will run very stiff until its bearings have become sufficiently worn to admit of the proper thickness of oil film for supporting the load. Under present conditions, with the limit gauges and more accurate

machine tools available, some of these factors are being incorporated in the design, as recorded by the drawings, but there are many factors which still escape and are not recorded, and it is left to the shops to do as their unwritten experience suggests to them may be right. The tendency in the bigger factories is to diminish the amount of responsibility left to the individual worker in respect to the employment of what, for want of a better term, may be termed "shop knowledge," and the reason may be found in the fact that whereas in the earlier days of engineering the same man both constructed and repaired, now, under modern conditions of output, the man who constructs is of a quite different class from the one who repairs, and the two classes are rapidly becoming almost out of touch with each other. Consequently that form of shop knowledge which was of such use to the mechanic of some years ago and which enabled him to put through work on the imperfect instructions of not very definite drawings, must to-day be replaced by positive information supplied by the designer and embodied in the detail drawings, figuring with limiting dimensions and supplemented by specifications.

In the broad and general consideration of detail, the first and most important point to be dealt with is that of standardisation, and, consequently, interchangeability. In spite of all the efforts of Whitworth and others in creating standards of size and form for screw threads and for other details, there are still numbers of manufacturers in the country who work almost as though such standards had never existed. That is to say, there are firms who will make ½ in. bolts 1-64 in. large or 1-32 in. large because the user will get a stronger bolt although he is buying the same size. This will account for the fact that many coach bolts and nuts are not interchangeable. Again, there are numerous makers of screws who claim that their product is within 1-1,000 in. or 1-500 in., as the case may be, but the accuracy of which does not run beyond the written or verbal statement, the actual bolts or screws having an error many times as great. The fit of screws between proper limits is quite as important as the accuracy of pitch and of shape of thread. A loosely fitting screw in machinery is subjected to so much vibration as is common with automobile vehicles will ultimately cause waste of time and trouble to the user, if not damage to other parts of the machine, wheras the too tightly fitting screw has its obvious disadvantages.

Standardisation is looked after by committees who formulate very excellent rules which should be followed by the manufacturer, but in many cases there is a want of uniformity in the resulting product which calls for better inspection at the start of operations and for the checking of the gauges to which the work is made. The admirable work now being done by the National Physical Laboratory in connection with all classes of standardisation cannot be overrated; but the importance of the independent checking of commercial standards by such an impartial central authority is not so fully appreciated by manufacturers as it might be.

Again to take an example, the ordinary pneumatic tyre is supposed to be interchangeable, that is to say, the same rims will do for any of the tyres made by the leading makers. The same pump connection serves for pumping up the inner tube, but it will be found that the same uniformity does not apply to the other details which go to make up the complete tyre on its rim. Security bolts, for instance, have various threads, causing the expenditure of much profanity on the road. The diameters of covers are not always in agreement with the rims within the usual limits, with the result that a cover may prove to be tight on the rim and may give considerable trouble in getting it into place. The checking of any of these dimensions is beyond the ordinary purchaser or consumer besides being outside his province, and the trouble caused by too great deviation from the standard dimensions is only discovered at a time when it gives great inconvenience. The question again is one of the various limiting dimensions of the rim and of the limits permissible in the cover under normal conditions.

In the wheels of pleasure vehicles ball bearings have been used for some years with increasing success, but their application was delayed through failures, in some cases due to over-loading, and in many others through imperfect provision being made in the casing of the bearing against the entry of water and mud. In fact, a considerable period of time elapsed before the various causes which contributed to the failure of ball bearings in road wheels were appreciated at their proper values; the several factors of the provision for taking end-thrust, the amount of the permissible radial load, the exclusion of water and dirt, and of uniformity in the quality of the manufactured ball bearings all tending to complicate the commercial solution of the problem. This matter of wheel bearings is essentially one of design, because the ball bearing, when worn, is, in general, beyond repair, so that the question of prolongation of the life of ball bearings is one which can only be referred back through the repair departments, who effect the renewal, to the designer. The presence of moisture, which has resulted in the failure of ball bearings in the road wheels, has also been found to affect those ball bearings which have been fitted to the crank shafts of some engines, and it has been found that a small amount of water in the lubricating oil will cause a sufficient pitting of the surface of the ball races and of the balls to result in premature failure.

In the engine many improvements in detail have been made, resulting in an enormous advance in respect of silence, speed of revolution, and power for piston area. Apart from such questions as multiplication of the number of cylinders, these improvements, however, have been confined to reduction in the weights of reciprocating parts, alteration of the arrangement and types of valves, modification of the shape of cams and of the size of the cam rollers, care in the selection of the materials and teeth of the gears used for driving the cam shafts, the replacement of low-tension by high-tension magneto ignition, and in general by improvements of detail.

It is, however, in the carburettor that the main problem of advance in the internal combustion engine appears at present to lie. Carburettors have been made giving over 50 ton miles per gallon on ordinary touring cars when running at speeds up to 40 miles per hour, and there appears to be no reason why such results should not be easily and regularly obtainable when the carburettor has attained a development as far advanced as that of the high-tension magneto. At present the tuning up of the carburettor is still frequently effected by the expensive method of running the car on the road, involving a considerable expenditure of the time of a skilled tester, the wear and tear of the whole machinery of the car, and the wear and tear of the tyres, which even if only old tyres are used, must be added to the other costs. It is true that on the road the conditions under which the carburettor is working are quite different from those of the testing bench. The forward movement of the car may give increased air pressure at the intake of the carburettor; the vibration of the car may appreciably alter the mean level in the float chamber and the amount of petrol which flows through the nipple. Usually these matters are adjusted by the tester by varying the size of the orifice in the nipple, but from an examination of the conditions which lead to the necessity for this adjustment, it would appear that frequently it is the level in the petrol chamber which requires adjusting quite as much as does the size of the orifice, and in but few carburettors is any provision made whereby the tester can set the level of the petrol to the desired height otherwise than by filing down the nipple or adding solder to the float.

In the clutch there is less complaint than was formerly common, in fact the peculiarities of leather, cone, and disc clutches have become sufficiently well understood by designers to render this detail one of those which now causes but little difficulty; in the case of metal disc clutches, the difficulties first met with in their use were mainly due to the imperfect knowledge on the part of the user of the proper conditions under which to work them, and, in this case, it is the improvement in the mechanical education of the user that has permitted their continued employment.

In the last few years the question of the reduction of the noise of motor vehicles has been almost entirely dealt with in the engine and gear box, apart from the change from chain drives to live axles. In the gear box noise was found to be produced by errors in the shape of the gear teeth, which caused irregularity in the velocity of the driven shaft accompanied by separation of the driving surfaces at speeds beyond a certain minimum. The improvements in gears have entirely been improvements in detail; the involute form of tooth has been retained and the angle of inclination to the tangent of the path of the point of contact has seldom been varied; on the other hand, not only have the cutters been made of greater accuracy than those employed for the construction of other classes of machinery, but methods have been adopted, such as those for developing gears by hobbing, which of themselves produce an approximation to the true form of tooth much more accurate than was obtainable by older methods. Again, the distortion of the gear wheels which may occur in cementing and in case-

hardening has been more thoroughly appreciated, and precautions have been taken by manufacturers which have resulted in a much smaller error in the finished product. In the back axle a source of noise has remained in the bevel gears, which even though made on developing machines, are liable to the introduction of more error than is the case with spur gears. Machines have already been devised, and some are obtainable, for correcting the errors in spur and other gears by grinding, and, should it become necessary to run gearing of light weight transmitting large powers at still higher linear speeds, it may be necessary for manufacturers seriously to consider the subject of grinding the bevel gears as well as the spur gears to the final degree of accuracy required. The back axle difficulty can of course be overcome by the use of a properly designed worm gear, and here again it is detail of design which fully determines the difference between the unsatisfactory and the satisfactory.

In the ordinary touring car there is still one detail which looks as though it should be altered before long, and that is the want of alignment between the propeller shafts and the shafts in the gear box when the car is under its normal load. It would appear that a simple modification should be possible by which the whole length of shaft would be in alignment from the front of the engine to the centre of the back axle when under normal load. At present the chief difficulty appears to lie in the lubrication arrangements for the engine. Now the angle of inclination of the shafting, if it is made lineable, is but small, and is in fact much less than that of any of the gradients up which the engine is required to work at full load. If the uniform lubrication of the engine were assured for a larger range of angle, covering the total inclination of the engine to the frame added to that of the maximum gradient to be ascended, this difficulty would disappear. In the case of the transmission on commercial vehicles, the chain has been found far from unsatisfactory, especially since it has been possible to obtain chain cases which are at the same time simple and sufficiently oil-tight to ensure the chain running continuously in an oil bath. Under such conditions the chain is much more silent, its life is increased to such an extent that the cost of chains as a factor in the running becomes negligible, and chains running in proper oil-tight chain cases can now be guaranteed for a life of over 25,000 miles.

In the case of public service vehicles, the improvements made in detail are immediately noticeable on the London streets, where some of the earliest taxicabs are still running side by side with the latest types. In motor-omnibuses the contrast is still more marked between the old pattern with the chain gear and the new pattern with the bevel drive, and here we have a paradox, for some of the more silent omnibuses, though they have no chains in the transmission from the gear box to the back axle, yet have a greater number of chains running at much higher speeds continuously within the gear box itself. The chain itself, therefore, should not be held to blame for the noise, but the cause should be attributed to the faulty method of application of the chain.

Provision for wear and tear is now made more ample than it was in the early days of the self-propelled vehicle, when the motor was frequently constructed in the form which may be called the "sandwich" engine. In this a single plane joint divided the upper from the lower half of the casing, with the half bearings contained in each of the respective casing halves, so that when it was required to take up such wear as had taken place, it was necessary, after dismantling the engine, to take a cut over the whole surface of one half of the casing, or else to replace entirely the whole of the brasses in the bearings. This arrangement, which would not have been tolerated for a moment by a constructor with ordinary engineering experience, had, however, one great advantage, and that was that it ensured oil-tightness in the casing. Later, when engine cases were first designed so as to make provision for taking up the wear of the main bearings, difficulties in securing oil-tightness were met with which required to be overcome by various arrangements of detail for preventing the loss of oil, or for ensuring its return to the crank-chamber, and, amongst others, in many engines it was found necessary to provide a vent for the escape of such gas as leaked into the crank-chamber, a provision seldom necessary with the sandwich engine. The provision of this air vent might with advantage be adopted in some gear boxes and back axle casings, in which the warming of the lubricant which unavoidably takes place when power is being transmitted causes sufficient expansion of the air contained in the casing to force a small quantity of oil continuously along the shafts, and to cause not only wastage, but, in some cases, the unintentional oiling of the brake surfaces.

Experience has now determined the amount which should be provided for wear in the brakes; brake surfaces have been increased, and provision has been made for an ample range of adjustment in the brake gear. In the earlier designs of autombiles, the designer seldom compared the new or maximum form of the brake shoes or drums with the worn out or minimum thickness, with the result that frequently it was necessary, when adjusting the brakes, to cut the rods for length and re-thread the ends, or to set the levers in order that the necessary adjustment could be effected.

The popular denunciation of mechanical traction on account of its alleged danger, usually based on the single factor of its speed, is contradicted very effectively by the figures given in the police statistics for the total number of accidents caused by vehicles which have occurred in the metropolitan area within the last three years for which the returns are available, that is, from 1907 to 1909. During this period the total number of street accidents caused by all classes of vehicles involving injury actually shows a slight decrease, while the population is estimated to have increased by 1.5 per cent. per annum, and the number of automobile vehicles registered has increased by about 19 per cent. per annum. Considered on the basis of the population, the accidents per million increased by 50 per cent. in the last six years (1891 to 1897) prior to the advent of the automobile; they attained their maximum in 1907, and have decreased by 2½ per cent. in 1908 as compared with 1907, and by nearly 5 per cent. in 1909 as compared with 1907.

Accidents, caused by automobile vehicles, excluding those due to the human fallibility of drivers and others, may arise from neglect, from defective material or from defective design. The elimination of the first of these causes is largely a question of management and of the responsibility carried thereby; defective material may be guarded against by proper specification and efficient supervision, but questions of defective or inadequate design, especially in regard to such vital details as the stering gear and the brakes, call, in certain cases, for examination by an independent body. There are many public vehicles now on the road covering long distances from their base, and though these may have passed the local police inspection and be considered adequately fitted for local conditions, yet outside the area of operations of the local authority they may be required to run under very different conditions of maximum gradient and of road surface. From some of the serious accidents which have occurred to such vehicles, it would appear necessary that the approval of the design and the inspection of these vehicles prior to their going on the road should be performed by a staff having special technical knowledge. The experience of this staff, as in the case of others appointed to examine into boiler explosions and railway accidents, should be supplemented by an inquiry into the cause of all fatal automobile accidents due to mechanical failure, as in the case of the above-mentioned classes of fatality. These enquiries and inspections should be performed by a department of the Board of Trade represented by an expert official.

Accessibility is a question which has had a great influence on the design of the automobile, and in some instances may have determined the type which has set the fashion, and fashion in the automobile vehicle plays a more important part than it does in any other class of machinery within my experience. The necessity for frequent access to the engine, to its ignition gear, to its carburettor and to its valves has ensured the placing of the engine in the front of the car where it could be quickly and most easily reached with the minimum of disturbance to the main portions of the vehicle. This fashion in position of the engine is likely to die very hard, so accustomed have we become to giving up the front of the car to the engine.

Accessibility may be divided into two main heads; first, accessibility to those parts which frequently require adjustment requiring no special skill, such as the adjustment of the brakes, of the strength of the clutch spring, of the spark of the ignition devices, and the like, most of which have already been dealt with by the designer in arranging them, and second, accessibility to details requiring skilled attention. In the latter class come the overhauls of engines, gear boxes, axles, etc., and the influence of commercial and public service vehicles on this branch of the subject is only now commencing to make itself felt. The importance of being able to remove parts of a car, unit by unit, that is, engine, gear

box, back axle, etc., has now become recognised by those responsible for the vehicles of public services such as those of the motor omnibus and cab companies, since the conditions of working such services are much more closely allied to those of the railway and the tramway than are the conditions of the private car or the commercial vehicle. The easy removal of these units complete, and their interchangeability with other similar units on the same class of vehicle is a large factor in economically keeping a fleet of public service vehicles upon the road.

Apart from the two broad questions just mentioned, a third and very important factor is that of accessibility to the various parts by those tools used in making the adjustments. With the necessity for keeping down weight has come the reduction in the size of nuts below those selected by Whitworth for a material the use of which in automobiles only occurs in body work—reduction in the size of nuts has been accompanied by reduction in the width of flanges—the whole of the work has become more cramped and the clearance between the faces of the nut and other adjacent surfaces has been greatly reduced. To put the parts together in the first instance may require the use of special spanners, owing to the fact that the designer has not laid a scale-tracing of an ordinary spanner on his drawing and ascertained that it can be effectively used, that he has not tried the clearance between the corners of the nut and the adjacent surfaces to ascertain whether a box spanner will overcome the difficulty, or, if so, that he has not allowed for the height of the box spanner. Ignorance of these factors contributes heavily to the repair bill, especially if such inaccessibility is assisted by the super-imposition of small details, and particularly piping, which requires removal before the main parts become accessible. If these features were considered in design, bolts would often be substituted for studs, long bosses would be cast on parts to enable the nuts or bolt heads to be reached; channels would be milled across faces into which bolt heads could fit to prevent them from turning, and such parts as guardings and covers would be so made that their detachment would be dependent on very few devices, and those of kinds easily secured and readily locked.

The difficulties connected with lubrication in the engines of cars have been already alluded to briefly, but far more important is this subject in the case of the marine motor, in which continual changes of inclination are taking place, and even greater still is its importance in the case of the aeroplane engine, in which the engine shaft is inclined considerably from the horizontal for long periods of time, and in which the question is often further complicated by the use of engines with fixed crank shafts and revolving cylinders.

In the case of engines employed for locomotion on the water or in the air, we come again to the carburettor question as one of the greatest importance for the immediate future. It is a question of such moment that a large proportion of the energy of research workers might well be devoted to it, and if their efforts were assisted by the loan of modern engines adjusted by the makers and supplemented by details of the power and consumption obtained on the test bench, the work would be much facilitated. The work done by Willans on steam engines in the testing house at Thames Ditton has left its mark on high-speed steam engine design. It is therefore reasonable to hope for similar results from well-directed research on the modern petrol engine.

The existence of the automobile as a practical comercial machine has been shown by others to be largely dependent on materials previously neither readily obtainable nor extensively used, such as rubber, aluminium and petrol; the same applies to the steels used in construction, which, though previously known, were not only difficult to obtain in the necessary commercial forms, but their proper heat treatment was but imperfectly understood owing to lack of research and of experimental data. In the early days of engineering steel was steel, that is to say, there was wrought-iron or cast steel (tool steel), and mild steel is a comparatively recent product, but with the large number of special steels used in the construction of the automobile, it has become difficult to distinguish by simple tests any one quality from another. Under these circumstances it is necessary that the automobile manufacturer should adopt a system in his works for marking the different qualities of steel (as by painting them a different colour at one end of each bar, for instance), in order that a store-keeper or other unskilled worker may be able to issue or receive the proper quality for any detail required.

Leaving the vehicles of the present, and the position which detail has taken in their development, we may pause to ask ourselves what are the possible detail improvements which will influence the future evolution of light automobile machinery, whether for transport by earth, water, or air, and what effect will they have on the design and construction of the vehicles of the future.

Fuels.

Regarding fuel as a storage of energy, we have in petrol nearly 50 per cent. more energy per unit weight than is stored in coal, and, moreover, we have it in a more convenient form, owing to the advantage which a liquid possesses over a solid.

Among so-called improvements may be cited solidified petrol, but it is difficult to imagine what possible advantage a solid which is troublesome to handle can have over a liquid which can be readily led from its reservoir to its destination through a pipe by gravity, or by pressure if gravity will not suffice.

Among gaseous fuels, acetylene has been proposed, and though this compound is to some extent endothermic yet it has not a sufficiently high thermal value to render it a competitor of petrol, particularly as it is unsafe when compressed. It gives, however, a wider range of explosive mixture when mixed with air than the other well known hydrocarbons, and the mixture of acetylene and air fires at a lower temperature than is the case with other gases. In considering the applications of such gases of high calorific value as acetylene or hydrogen, it is only necessary to make a rough calculation to realise that at present no saving, but, on the other hand, a great increase of weight would result were they generated on the vehicle owing to their small weight relatively to that of the compounds used in their production. Moreover, the same applies to the storage of compressed gases, which can only be considered commercially practicable in the case of town gas applied to heavy vehicles engaged on runs of but short mileage between fixed charging points.

The fact that acetylene gives a larger range of explosive mixture than other well-known gases, and that the mixture fires at a lower temperature would point to a possible saving in the weight of the ignition apparatus. This saving would, however, be but small, and affects but little the question of the total energy obtainable within a limit for the combined weight of fuel and engine.

Lubrication.

In the desire to reduce the weight of the transmission gear, the diameter of shafts in the early vehicles was reduced to the minimum, and, in order to obtain the requisite area of bearing for carrying the load, the bearings in the gear box were of necessity made long. The spring of the shafts under the heavy loads to which they were subjected resulted in bending to such an extent as to reduce the thickness of the oil film locally below that necessary for efficient lubrication, with the result that in many of these earlier cars, difficulties arose in maintaining the bearings in efficient order. This difficulty has been largely overcome by the use of the ball bearing, which, as it takes up less length of the shaft, reduces the effective span between the supports and diminishes the spring. But this is not the only advantage given by the ball bearing; still more important is the fact that it is capable of working satisfactorily with a greater error of alignment than is possible with a plain bearing. The ball bearing working under suitable conditions, and provided it is not overloaded in the first instance, appears capable of running almost indefinitely when immersed in an oil bath and kept free from small pieces of abraded metal. In fact, in the gear box, ball bearings generally give less trouble than on other portions of the car. Although the ball bearing has such marked advantages when treated in a suitable manner, yet under conditions less favourable, such as those of the road wheels, where a bearing may be called upon to stand excessive and obliquely applied loads, failure is much more easily produced, especially if accelerated by the penetration of water even without dirt, into the bearings as previously mentioned. The effect of water in destroying the smoothness of the surfaces leads to rapid disintegration, and once the ball bearing has begun to fail either by the breakage of the balls, or of the race, its end occurs more rapidly than is the case with the plain bearing.

Increased Motor Efficiency.

The improvements in the efficiency of motors have been almost inseparably linked with improvements in carburettors; nevertheless, improvements in the motor itself have to no small degree contributed to the advance in the amount

of power obtainable per unit of weight of motor and in the efficiency of the motor itself as a thermodynamic machine. The consumption of fuel per brake horse-power hour in the petrol motor has now been reduced to 0.63 lb. Allied to the question of efficiency is the question of obtaining small commercial motors capable of working with a less highly inflammable fuel, such as ordinary paraffin oil. Many attempts have been made to effect this by slight detailed modifications of the engine or of its carburettor, but the problem is one on which mechanical engineers have already spent vast quantities of time and money, the Priestman and the Hornsby-Ackroyd oil engines being examples. The problem has, however, been tackled recently in a different manner by a method which combines the carburettor with a gas producer.

It is well known that if a candle be blown out, the gas which rises from the wick is inflammable, and that a light held some distance above the wick will ignite the mixture of gas and air resulting from the incompleted combustion, and that the flame will travel down the ascending column of gas, re-igniting the candle. A similar principle underlies the action of the suction gas producer now used on many gas engines.

A paraffin carburettor has now been produced which resembles a gas producer in so far that a portion of the oil supplied to it is partially burnt, but, unlike the case of the ordinary producer in which anthracite or other coal is used, a portion of the heat generated in the partial combustion may be utilised to vaporise an excess of the fuel so that a mixture of producer gas enriched with oil vapour can be admitted to the engine. By keeping the percentage of vapour sufficiently low it is possible to avoid the difficulties which occur by the clogging of rings and the deposition of carbon when paraffin oil is used direct in the ordinary petrol engine.

Further hope of using heavier and cruder fuel lies in the adaptation of the Diesel principle to the ordinary form of automobile engine. The efficiency of the Diesel engine has now been brought to so high a pitch that engines of 500 horse-power are built with a guaranteed consumption of crude oil as low as 0.42 lb. per brake horse-power per hour at full load. The Diesel engine, however, as at present constructed, is unduly heavy, and some time must elapse before it has been sufficiently modified to be of suitable weight without having lost the efficiency obtainable in its heavy form. Not only is the Diesel principle in its 4-cycle modification a possible rival to the ordinary Otto cycle engine, but the same may occur with the two-cycle Diesel, to which some considerable amount of attention is being devoted at the present time. Another rival in the field to these engines at some future time will undoubtedly be the internal combustion turbine. In 1884, in a lecture on turbines given by Professor Unwin before the Institution of Civil Engineers, the difficulty in the way of the construction of a steam turbine was tersely summed up by him in the following words :—

"So soon as we can find a material strong enough and durable enough to stand an excessive speed of that kind (1,000 feet per second) so soon we may have steam turbines much smaller and cheaper and not less efficient than ordinary steam engines."

After the successful progress made by the steam turbine, and its development as predicted by Professor Unwin, it has gone on until it has passed the reciprocating steam engine in economy. A very similar prediction could be made at the present date in regard to the internal combustion turbine, modified in this respect, however, that temperature as well as velocity are the factors to be considered.

Many attempts have been made to obtain transmissions which are either variable in speed or not merely variable in speed, but which also give facilities for storing excess of energy in the motor, and utilizing this stored energy when required. The latter class has been fully dealt with by my predecessor in the chair, and the former is being developed at the present time by another former President of this Institution on lines which promise immediate fruition.

Improvements in Materials of Construction.

Not only have steels and aluminium alloys been improved in their ultimate tensile strength and in those other physical qualities necessary for the safe employment of such materials in constructional work, but increased knowledge has been obtained as to the proper treatment which these materials should undergo in order to give the best commercial results. The heat treatment of steels, whether for oil tempering or case-hardening, the proper treatment of aluminium alloys in order that the properties of the original mixture may be retained in the castings, and the diffusion of knowledge of rubber treatment and vulcanising outside the highly specialised tyre factories, are instances in point. There are, however, certain possibilities in the case of steel which may yet have to be considered. It has been found that metallic tantalum has very great hardness and power of resisting wear by abrasion; it is recorded that an attempt to drill a tantalum sheet by means of a diamond drill run at 5,000 revolutions per minute for 72 hours, resulted in a penetration of the metal to the extent of $\frac{1}{4}$ millimetre only, and was accompanied by considerable wearing of the diamond tool. Now, if we suppose that the surface of a prepared piece of steel could be treated with tantalum in the same way as it can be treated with carbon in the ordinary case-hardening operation, and that, in fact, it could be superficially coated with a firmly adhesive coating of metallic tantalum, it should be possible to reduce any bearing surfaces so treated to dimensions hitherto unapproached under the heaviest loading. In other words, whereas the ball-bearing may be narrower than the plain bearing carrying the same load, a bearing dependent for its rubbing surfaces on a material so much harder could be made still narrower, and accompanying the narrowing of the bearings a large reduction in weight could be effected in many instances.

The treatment of ductile materials has up to the present taken the form of drawing into wire and tube, and rolling into plate and bar, so that the forms available have been those of uniform cross-section or of uniform thickness. For many purposes the tube is an extremely efficient constructional factor; the ordinary bicycle affords one of the best examples of its utility in obtaining a light and rigid structure capable of carrying a heavy load while subjected to considerable shock; but nearly half the difficulties in bicycle construction, apart from those of the bearings (solved, as in the case of the motor car, by the ball-bearing) were encountered in the difficulties of making tubes of sections other than cylindrical, of making them tapered, and, above all, of joining the various elements in such manner that strength was not sacrificed at the joint. Attempts to use tube for heavier and larger constructional work have also been made, and a company was formed for producing tubular frame railway goods wagons. The facilities at present existing for preparing and joining the ends of channel sections and the absence of such facilities in the case of tubes, militated greatly against the success of the type. For the standards carrying the trolley wires of the electric tramway, tapered steel tubes are used, and it is a matter probably only of a few years before tapered tubes will become more generally available as constructional material. The instances to which their use would apply are numerous and obvious.

The distribution of a given mass of metal into that form in which it will carry the greatest load in bending when employed as a beam of given span, or the greatest load in compression when employed as a strut, is in each case dependent on ability to resist buckling due to failure under compression rather than to actual failure under compressive stress. The stiffening of beams and struts against such failure is a matter studied by the big bridge builders, and it is in large bridge work that one finds the problem of tubular construction seriously considered. In the case of light struts and structures, particularly the parts of aeroplanes, were it possible to construct such parts of cellular form and of metal of high tensile and compressive strength, it is obvious that the weight of many structures could be reduced greatly below that which is at present attainable. The difficulty in the problem lies in the joining of one piece of thin metal to another without sensibly impairing its physical qualities. For certain classes of work of large section, electric welding has been available, and by its aid good work has been done; for others acetylene welding is being tried at the present time, and it would appear that there is a field open for research on the construction of beams and struts of minimum weight in which the elementary forms are produced from thin sheet or tubes joined by such methods as are commercially available.

In concluding this broad and imperfect survey of recent progress and the possibilities of the immediate future, it appears to me that the development of the automobile vehicle for the next decade lies most largely in the hands and heads of the younger members of the profession who are engaged in the design of vehicles and their accessories. Along the lines laid down by their observation and the intelligent conclusions drawn therefrom, the evolution of the vehicle will in a great measure take place.

CASE HARDENING BY MEANS OF COMPRESSED GASES.

Abstract of a Paper given by F. Giolitti (Turin) and F. Carnevali (Turin), given before the Iron and Steel Institute.

IN a paper, published about two years ago,* we communicated the results of certain preliminary experiments carried out with the view of establishing what influence, if any, variations in the pressure of carburising gas (in the special case considered, carbon monoxide) exert on the characteristics of cementation zones.

These preliminary experiments had already furnished proof that the characteristics of cementation zones vary in notable fashion concurrently with the variation in pressure of the carburising gas, and more particularly in the case where carbon monoxide reacts on ordinary steel in the presence of free carbon, in the sense that an increase of pressure produces (all other conditions remaining unchanged) an increase in the depth of the cementation zone obtained within a given time, and an increased concentration of carbon within the zone itself.

Apart from the practical conclusions to be drawn from such results, the results in themselves constituted a fresh, irrefragable proof of the direct and dominating intervention of carburising gases (and especially of carbon monoxide) in the process of cementation. In view, however, of the *ensemble* of theoretical knowledge recently acquired concerning the chemical reactions which take place in the course of case-hardening by means of case-hardening agents with a carbon monoxide base, and more particularly concerning the conditions of equilibrium of such reactions, it was only natural that we should not stop short at these first conclusions, but that we should endeavour to extend them by dint of more complete and more exact experiments.

With this object we thought it advisable, first of all, to modify some of the structural details of the apparatus utilised in the earlier experiments. The differences between the old and the new apparatus, apart from modification in the dimensions of various parts, so as to obtain a fuller utilisation of heat and more uniformity in the heating process, are those devised in the methods of closure. The new methods, the arrangement of which is clearly shown in Fig. 1, while retaining the old facility of rapidly setting up and dismantling the apparatus, allow of maintaining the vessel perfectly gas-tight. This is an essential towards an exact knowledge of one of the variables which (as we have shown in previous papers) possesses preponderating influence on the results of experiments of this kind—namely, the velocity of the current of carbon monoxide which flows into the case-hardening chamber.

A and B (Fig. I.) are clamps attached to terminals (insulated electrically and gas-tight) which pass through the wall of the cast-iron receiver C, and conduct, to the nickel-wire spiral D, the current which is destined to produce the heating effect. This spiral is wound round the porcelain tube E, which can be easily detached from the apparatus and put back again, as it is merely enclosed in a fireclay tube (F) of larger diameter. The space between this tube F and the wall of the cast-iron vessel is packed with asbestos. The gas (carbon dioxide, CO_2) enters the apparatus by the tube G and leaves it by the tube H. By the tube of specially strong porcelain (I) is introduced the thermo-electric couple which is destined to measure the temperature along the entire length of the case-hardening chamber: beside it are placed the blocks which are to undergo case-hardening (L) completely surrounded by granular carbon.

The experiments were carried out in the manner previously described, the case-hardening being accomplished by means of a mass of wood charcoal traversed by a slow current of carbon dioxide. We demonstrated on another occasion that this gas supplies with great rapidity, and without any possible excess of carbon monoxide, a mixture of carbon dioxide and carbon monoxide of a concentration exactly corresponding to the equilibrium with free carbon, under the conditions of temperature and pressure employed in the operation. We made use of steels of different composition in the form of tiny cylinders, measuring 10

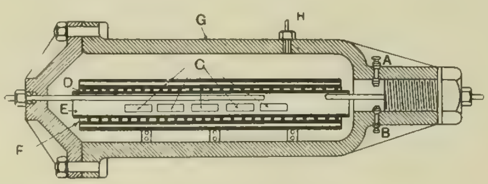


Fig. 1.

millimetres in diameter and from 70 to 100 millimetres (2.8 to 4 inches) in length, which we were careful to bury completely in the customary granular carbon. We indicate farther on, in the case of each experiment, the precise conditions of the operation, and, above all, the temperature, duration, pressure of gas, and velocity of the current of gas flowing through the apparatus. In regard to this last point, it must be noted that the numbers which we shall tabulate do not refer to the gas (dry carbon dioxide) which entered the apparatus, but to the gas which issued therefrom, measured under ordinary pressure. Now, since this latter gas—in equilibrium with the carbon at the temperatures at which we worked—was almost entirely made up of carbon monoxide (containing not less than 2 or 3 per cent. of carbon dioxide), its volume was about double that of the carbon dioxide introduced into the apparatus, measured under the same conditions of temperature and pressure.

With regard to the temperature, the difficulty of insulating thermally the heating apparatus, and more particularly the necessity of water-cooling the cast-iron walls of the vessel, made it practically impossible to maintain the temperature rigorously constant during the entire process of cementation. As will be seen later on, we can only determine for each case-hardening operation a sufficiently ample interval during which the temperature remained constant.

The steels employed were of the following compositions:—

1. Ordinary soft carbon steel, which we shall subsequently designate "Carbon Steel."

	Per Cent.
Carbon	0.11
Manganese	0.54
Silicon	0.05
Sulphur	0.02
Phosphorus	0.04

2. Soft steel with 2 per cent. of nickel, which we shall subsequently designate "Nickel Steel 2."

	Per Cent.
Nickel	2.03
Carbon	0.10
Silicon	0.26
Manganese	1.38

3. Soft steel with 5 per cent. of nickel, which we shall subsequently designate "Nickel Steel 5."

	Per Cent.
Nickel	5.02
Carbon	0.118
Silicon	0.20
Manganese	1.53

4. Steel with 25 per cent. of nickel, which we shall subsequently designate "Nickel Steel 25."

	Per Cent.
Nickel	24.92
Carbon	0.17
Silicon	0.10
Manganese	3.46

5. Steel with 2.3 per cent. of chromium, which we shall subsequently designate "Chromium Steel."

	Per Cent.
Chromium	2.33
Carbon	0.41
Silicon	0.15
Manganese	1.02

6. Chromium-nickel steel, which we shall subsequently designate "Chromium-Nickel Steel."

	Per Cent.
Chromium	1.50
Nickel	3.17
Carbon	0.33
Silicon	0.06
Manganese	1.15

In all these steels the percentages of sulphur and phosphorus were less than 0.04.

The accompanying Table I. embodies a synopsis of the conditions attending the various case-hardenings, as well as some remarks (the value of which will be appreciated later) concerning the state of the surface of the various steels after case-hardening.

In order to follow the process of concentration of the carbon in the carburised zones, we availed ourselves of the microscope in the case of the steels Carbon, Nickel 2, and Chromium; thanks to this microscopic investigation fairly precise data (especially in the case of the two first-named) were obtained and are set forth in the second table.

For the other three steels, Nickel 5, Nickel 25, and Chromium-Nickel, in regard to which microscopic examination,

*F. Giolitti and F. Carnevali, "Researches on the Manufacture of Cementation Steel—V. (Case-hardening by means of Highly Compressed Gases)." *Atti della R. Accademica delle Scienze di Torino*, vol. xlv., February 13, 1910.

as is well known, could not yield reliable results, we carried out quantitative determinations of the carbon in the material obtained by machining away successively, and collecting separately, "co-axial" layers of about a quarter of a millimetre thick from the small case-hardened cylinders.*

We carried out also in the same way quantitative analyses of carbon on the layers of the case-hardened cylinders of Chromium steel, because we wished to determine exactly the extraordinary increase in the carbon content in the outermost thin layer which had been revealed to us by the microscope. The results of these analyses are embodied in Table III.

One primary fact which meets the eye from the experimental data here tabulated is the influence which variations in the pressure of the carburising gas exert on the depth of the case-hardening and on the concentration of carbon in the carburised zones. This fact fully confirms the results to which our previous experiments had led, and proves that they also apply to the special steels of different types on which we have now been working.

It is hardly necessary to examine in detail the numerical data set forth in the foregoing pages, in order to demonstrate the accuracy of this statement; for it will become perfectly evident on mere comparison of the data which we now publish and on comparison with those previously published in sundry memoirs concerning the case-hardening of carbon steels and special steels with the same case-hardening mixture as that utilised in the experiments here recorded, but utilised at ordinary atmospheric pressure. The effects of greater pressure of the carburising gas—effects which are more particularly manifested in the increased concentration in the carburised zones—confirm the conclusion at which we arrived on former occasions concerning the direct intervention of carbon monoxide in case-hardening effected by means of the "mixed" case-hardening agent.

Another fact is plainly to be deduced from a comparison between the results of Experiment VI. and those of Experiment VIII.; between those of No. IX. and those of No. XII.; between those of No. XVII. and those of No. XX.; and between those of No. XXI. and those of No. XXIII. Such comparisons, indeed, show how an increase in the velocity of the current of carbon dioxide tends to cause a diminution in the intensity of case-hardening to such an extent as to eliminate (in the four series of experiments just enumerated) the effects of increase of pressure. We have seen, however, these latter effects manifested with noteworthy intensity in those cases where comparison is made between carburised zones obtained by working under varying pressures (ordinary pressure and pressure ranging from 15 to 25 kilogrammes per square centimetre) but with currents of carbon dioxide of constant velocity.

The cause of this phenomenon is that in case-hardening accomplished by means of a cementing mixture based on the simultaneous action of carbon dioxide and free carbon, a state of complete chemical equilibrium is never really attained; while the ultimate characteristics of the carburised zones depend in great measure on the relations between the velocities of the various reactions which take place during the process of case-hardening.

A third phenomenon of surpassing interest, both from the theoretical and from the practical point of view, a phenomenon which, according to the data tabulated in the foregoing pages (see especially Table I.), occurs with notable frequency and intensity in the course of case-hardening operations carried out under high pressures by means of a case-hardening mixture based on carbon monoxide, is

Table I.

No.	Steel Used.	Duration of Heat.	Pressure of Gas (in Kilogs. per sq. cm.).	Range of Temperature within which the Case-hardening was performed.	Velocity of Gaseous Current (CO_2) — Litres per hour for every sq. decim of Carburised Surface.	Remarks on the Condition of the Surface of the Steel after Case-hardening.
I.	C	3 hrs.	15	900°-955° C.	1·5	Surface unaltered.
II.	,,	3 ,,	15	1020°-1050° C.	1 5	,, ,,
III.	,,	2½ ,,	25	890°-960° C.	1 5	Thick stratum of compact oxide.
IV.	,,	3 ,,	25	980°-1015° C.	3	Considerable oxidation.
V	Ni 2	3 ,,	15	955°-975° C.	1·5	Surface unaltered.
VI.	,,	3 ,,	15	1035°-1045° C.	1·5	,, ,,
VII.	,,	2½ ,,	25	905°-955° C.	1 5	Slight oxidation.
VIII.	,,	3 ,,	25	1030°-1050° C.	3	Surface unaltered.
IX.	Ni 5	3 ,,	15	850°-890° C.	1 5	,, ,,
X.	,	3 ,,	15	945°-995° C.	1·5	,, ,,
XI.	,,	2½ ,,	25	840°-930° C.	1 5	Thick stratum of compact oxide.
XII.	,,	3 ,,	25	875°-915° C.	3	,, ,,
XIII.	Ni 25	3 ,,	15	870°-930° C.	1 5	Surface practically unaltered.
XIV.	,,	3 ,,	15	1000°-1045° C.	1·5	Surface unaltered.
XV.	,,	2½ ,,	25	870°-950° C.	1·5	Thin stratum of non-compact oxide.
XVI.	,,	3 ,,	25	942°-980° C.	3	Slight oxidation.
XVII.	Cr	3 ,,	15	935°-965° C.	1·5	,, ,,
XVIII.	,,	3 ,,	15	1035°-1060° C.	1 5	Thick stratum of compact oxide
XIX.	,,	2½ ,,	25	900°-965° C.	1·5	,, ,,
XX.	,,	3 ,,	25	1010°-1035° C.	3	,, ,,
XXI.	CrNi	3 ,,	15	810°-850° C.	1·5	Considerable oxidation.
XXII.	,,	3 ,,	15	875°-915° C.	1·5	Thick stratum of compact oxide.
XXIII.	,,	3 ,,	25	810°-845° C.	3	,, ,,
XXIV.	,,	2½ ,,	25	850°-900° C.	1·5	,, ,,

Table II.

Heat No. (See Table I.)	Thickness of the Carburised Zone in Millimetres.	Thickness of the Eutectic Zone in Millimetres.	Thickness of the Hypo-eutectic Zone (up to about 0·4 per Cent. of Carbon) in Millimetres.
I.	0·4	0·3	0 5
II.	1·3	0·8	0·4
IV. . . .	0·7	0·6	0.4
V. . . .	0·25	1·1	0 35
VI. . . .	0 9	0·8	0 7
VIII. . . .	0·9	0·7	0·8
XVII. . . .	0·1	1·0	?[1]
XVIII. . . .	0 15	1·0	?
XX. . . .	. 0·8	1·7	?

Table III.

Heat No. (See Table I).	Concentration of Carbon. In the First Layer (0·25 Millimetre).	In the Third Layer (Depth = about 0 7 Millimetre).	In the Fifth Layer (Depth = about 1 Millimetre).
	Per Cent.	Per Cent	Per Cent.
IX. . . .	0 71	..	0·12
XIII. . . .	0·57	0.54	..
XXI. . . .	0·45	...	0·54
XVII. . . .	2·22	...	1·03
X. . . .	0·99	..	0·29
XIV. . . .	0·90	0.32	...
XXII. . . .	0·76	...	0 49
XVIII. . . .	3·1	...	1·39
XII. . . .	0·73	...	0·36
XVI. . . .	0·61	0·37	...
XXIII. . . .	0·54	...	0·56
XX. . . .	2·37	...	1·40

*In order to plane off the successive layers of the carburised steels of Experiments IX., X., XII., XX., XXI., XXII., and XXIII., it was found necessary to reheat them for five hours at a temperature of about 550° C. in a neutral atmosphere. We have already shown (on another occasion) that reheating under such conditions in no respect modifies the character of the carburised zone.

[1] For chromium steel having the composition of that adopted by us, the precise microscopic determination of the hypo-eutectic zone is extremely difficult.

the superficial oxidation of the case-hardened steel. Before examining in detail this phenomenon, it will be perhaps as well to set forth certain theoretical considerations.

An oxidation of this character had already been observed by Charpy in 1909, when he exposed to the action of pure carbon monoxide at 1,000° C. chromium, manganese, and various chromium steels (containing from 1.99 to 7.71 per cent. of chromium) as also chromium-nickel steels (containing from 2.08 to 6.45 per cent. of nickel, and from 0.70 to 7.04 per cent. of chromium). This investigator had noted that when he used these special steels in the form of filings, the carbon monoxide was decomposed by the chromium, and there resulted "simultaneously the oxidation of the chromium and the carburising of the iron: the two elements behaved as if they were isolated" (p. 513). He adds further (p. 514) that "when, instead of working with metals reduced to filings, one utilises pieces of tolerable dimensions, the same phenomena do not recur; the oxidation of the chromium is restricted to the surface-layer, beneath which case-hardening proceeds in the regular course by diffusion." From his experiments generally, Charpy concludes that "the action of carbon monoxide at 1,000°, which is a case-hardening one in the case of iron, as also in the case of tungsten and perhaps of nickel, is therefore oxidising in the case of chromium and manganese." We have already pointed out on various occasions how these conclusions deduced by Charpy from his extremely interesting experiments were too incomplete and too dogmatic in their simplicity, and were especially faulty therein that no account had been taken of the results of the important researches concerning the action of carbon monoxide on the metals of the iron group, which had been published by Schenck some time before.

In point of fact, Schenck's researches, of the results of which we have availed ourselves largely in the course of our technical investigations on the case-hardening of steel, had already led us to range within a single category the action of carbon monoxide in the various metals of the iron group, and thereby to simplify enormously and elucidate perfectly the theoretical treatment of the progression of reactions which take place in systems consisting of single metals, or carbon monoxide, of carbon dioxide, or carbon, and of the products of oxidation and carburisation of various metals.

Now, our experiments, the results of which have been summarised in the foregoing pages, allow us to give a sufficiently concrete shape (though certainly not definite and far from complete) to the considerations which may be developed on the basis of Schenck's results, in order to elucidate the general laws which determine the process of the case-hardening of special steels, by means of those case-hardening mixtures the activity of which is due to the specific carburising action of carbon monoxide.

But, before examining from this standpoint our experimental results, it would seem necessary to refer briefly to certain physio-chemical laws connected with the action of carbon monoxide upon metals, in order to see how far these laws are applicable in the present instance.

If we consider the very complete diagram of isothermal equilibrium drawn up by Schenck, in the case wherein the metal exposed to the action of carbon monoxide is iron alone, and at temperatures at which solid solutions (mixed crystals) of carbide of iron cannot be formed with the iron (that is, below 1,700° C.), we notice five curves of equilibrium corresponding to five distinct reactions. Representing, as usual, along the axes of the abscissæ the concentrations (X) of carbon monoxide within the mixture consisting of carbon monoxide and carbon dioxide (varying from 0 to 1), and along the axes of the ordintes the total pressures (P) of the above-mentioned gaseous mixture, the five curves are those represented in Fig. 3, and designated by the numerals 1, 2, 3, 4, and 5.

Let us recall here briefly the meaning and the equation of each of these curves:—

Curve 1.—Cubic hyperbola $\frac{x^2}{1-x} P = \mu$ (μ constant), corresponding to the conditions of equilibrium of the system (Fe_3C, Fe, CO, CO_2).

Curve 2.—$\frac{x^5}{(1-x)4} . P = \theta$ (θ constant), corresponding to the conditions of equilibrium of the system (Fe_3C, FeO, CO, CO_2).

Curve 3.—Straight line $\frac{x}{1-x} = \eta$ (η "constant of reduction"), corresponding to the conditions of equilibrium of the system (Fe, FeO, CO, CO_2).

Curve 4.—Straight line $\frac{x}{1-x} = k$ (k constant), corresponding to the conditions of equilibrium of the system (FeO, Fe_3O_4, CO, CO_2).

Curve 5.—$\frac{x^2}{1-x} . P=Z$ (Z constant), corresponding to the conditions of equilibrium of the system (C, CO, CO_2).

If, on the other hand, we work at a temperature higher than 700° C., carbide of iron may yield solid solutions with iron, and in the diagrams of equilibrium must appear, alongside the five curves already enumerated, the curves of equilibrium of the systems resulting from the foregoing by the addition of the mixed crystals of iron-carbon to the above-mentioned constituents. To each coefficient of the concentration of carbon in the mixed crystals corresponds an equilibrium curve, the equation of which (in cases where the mixed crystals are present in sufficient quantity to prevent their concentration from varying appreciably with the variation—within certain limits—in the pressure of the gaseous mixture) is still that of a cubic hyperbola of the form $\frac{x^2}{1-x}. P=v$, that is, analogous to the curve representing the systems (Fe, Fe_3C, CO, CO_2) and (C, CO, CO_2). Among all these cubic hyperbolas, constituting a sheaf or bundle, and all passing through the point (x=1, P=0),* there is one which presents especial interest from our point of view, wherein the process of case-hardening having been accomplished with the "mixed agent," all the reactions are invariably completed in the presence of a notable excess of free carbon ("granular" wood-charcoal). This curve is that which corresponds to the special concentration of carbon in mixed crystals, whereby the constant V has (at equal temperatures) the same value as the constant Z of the equilibrium equation representing the system (C, CO, CO_2). It is obvious, from all that we have stated, that this curve coincides in its entire length with the equilibrium-isotherm of the system (C, CO, CO_2). If we assume that all the operations have been carried out with sufficient slowness to attain invariably the state of perfect equilibrium (this, as we know, never precisely happens in practice, but we may admit a sufficient approximation if we content ourselves with deducing from our argument conclusions of a qualitative nature), then the final concentration of carbon in mixed crystals will always be that which corresponds to a value of the constant V, equal to the perfectly well-defined value which, at similar temperatures, is assumed by the constant Z corresponding to the equation $2CO \rightleftarrows CO_2+C$. We denote by Σ those mixed crystals wherein carbon is thus especially concentrated.

From what we have said above it may be inferred that, in the qualitative investigation of the reactions which take place during case-hardening by means of a case-hardening mixture based on the simultaneous action of carbon and the mixture of carbon dioxide and carbon monoxide in equilibrium therewith, we may eliminate all but one of the equilibrium-curves corresponding to the various concentrations of carbon in mixed crystals.

In other words, in the instance of case-hardening carried out with our "mixed agent," the problem assumes the same simplicity as that which characterises it (as Schenck has demonstrated) when its investigation is limited to temperatures (lower than 700° C.), at which true case-hardening does not take place, because the mixed crystals of iron and carbide of iron cannot form. We may, then, in a preliminary proximate study of the course followed by the phenomena in our experiments, confine ourselves to the consideration of the diagram built up of the first five curves.

Since in these experiments we have always started from gaseous systems rich in carbonic anhydride, operating in the presence of wood charcoal (to which form of carbon curve 5 of our diagram accurately applies), therefore in the event of no variations of temperature occurring, and on the supposition that conditions of metastability are excluded, the only portions of the diagram that should be taken into account are those on the left and below curve 5. We will begin with the examination of such a case. From the diagram it is immediately evident how the succession of phenomena varies fundamentally with the variation of pressure, and in this wise:—

(*a*) At pressures higher than that which corresponds to the point O† (intersection

*Some of these curves, marked *a*, have been reproduced in Fig. II

†It should be noted that, in our experiments, the pressure is kept constant during the entire operation, from the moment of the arrival of the fresh gas in the apparatus. For, if the quantity of gas were limited, it is well known that the initial minimum pressure of the carbonic anhydride, at which the formation of the magnetic oxide takes place, would be lower than the pressure corresponding to point O.

of curves 4 and 5), the formation of mixed Σ crystals and of magnetic oxide of iron takes place simultaneously in the presence of wood charcoal.

(*b*) At pressures ranging between that which corresponds to the aforesaid point O and that which corresponds to point Q (intersection of curves 3 and 5), there is simultaneous formation of protoxide of iron and of mixed Σ crystals, as before in the presence of wood charcoal.

(*c*) At pressures lower than that which corresponds to point Q, the mixed Σ crystals alone form, as always, in the presence of wood charcoal.

Therefore it is only at pressures lower than those which correspond to the point Q that the process of case-hardening with the "mixed agent" can take place independently, that is, unaccompanied by oxidation of the metal. Therefore, reasoning from what has already been said, we may foresee that, if the pressure of the gas is sufficiently raised in the case-hardening with the "mixed agent," phenomena analogous to those observed by Charpy in the case of chromium steels will also take place with pure iron or with the ordinary carbon steel.

This is abundantly confirmed by our Experiments I., II., III., and IV. (Table I.). In the first two of these, the pressure (15 kilogrammes per square centimetre) is lower than the pressure corresponding to the equilibrium point Q (for the temperature at which the case-hardening is completed); while, in the last two, the pressure (25 kilogrammes per square centimetre) has a value higher than that which the ordinate of Q assumes at the temperature selected for the experiment (900° to 1,000° C.). In point of fact, in the first two experiments the mere process of case-hardening is all that takes place; whereas in the last two the process of case-hardening (formation of the mixed Σ crystals) is accompanied by considerable surface-oxidation of the metal.

The condition of the small case-hardened and oxidised cylinders of steel demands a little closer investigation. We refer more especially to the cylinder that has undergone case-hardening in Experiment III.

That cylinder is coated with a compact

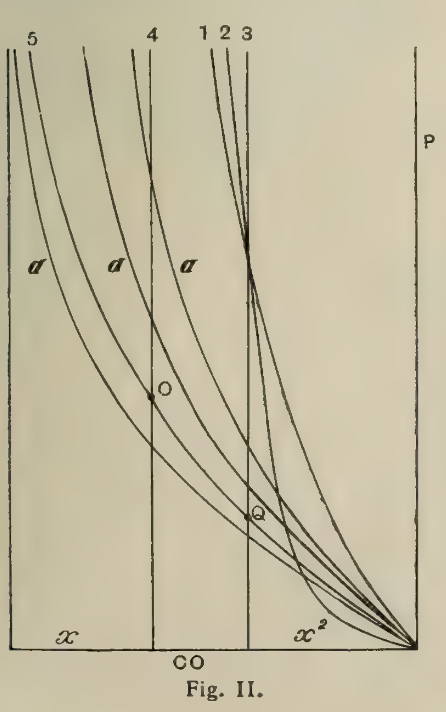


Fig. II.

layer of oxide, about 0.7 millimetre thick, wherein the fragments of granular carbon which were in contact with the surface of the metal have remained firmly encrusted. The metal below the oxide layer is intensely carbonised.

This assemblage of facts, which at the first glance may appear curiously contradictory, is nevertheless perfectly explicable, if we take into account the considerations already developed in this paper. It is illustrated by Fig. III., which represents, at a magnification of 65 diameters, a section (perpendicular to the axis of the cylinder) of the steel which immediately underlies the oxide layer. In this section, which has been cleansed with a 5 per cent. alcoholic solution of picric acid, it is plainly seen that the steel has undergone case-hardening, and that the concentration of the carbon attains, near the external surface of the carburised zone, a percentage of about 0.85.

Examination of the material which constitutes the external layer, easily detachable from the cylinder, shows it to consist of magnetic oxide of iron (Fe_3O_4), with which small quantities of carbon are intermixed.* It is, in fact, attracted energetically by a magnet. On analysis, it is shown to contain 69.2 per cent. of iron; the smaller percentage of the metal, as compared with that occurring in the pure peroxide (72.41), is evidently attributable to the presence of the occluded carbon.†

The position of the two points of unchangeable equilibrium, O and Q, whereof we may now appreciate the significance, varies concurrently with the variation in temperature and in composition of the steel which is subjected to cementation. Let us examine briefly the sequence of these variations.‡

With regard to the changes in the two pressures of equilibrium that are assignable to variations in temperature, we may observe that the three curves, 3, 4, and 5 (see diagram, Fig. 3), deviate all three towards the right when the temperature rises, in such wise that the variations in the ordinates of points O and Q depend on the relations between these deviations. These relations vary according to the composition of the steel, but generally they are such that the ordinates of points O and Q (minimum pressures of oxidation-cementation for each of the two degrees of oxidation) increase concurrently with rising temperature. Whence it follows that the higher the temperature of

*While it is easy to separate the oxide from fairly large fragments of wood charcoal, it is impossible to separate it from the pulverulent carbon (or charcoal dust) which is invariably mixed therewith.

†It is not improbable that a little ferric oxide also has been formed, as a consequence of some oscillation of temperature which may have lowered for a time the pressure of oxidation to Fe_2O_3 below the pressure at which the experiment was conducted.

‡From the practical point of view the only one of these two points that presents real interest is the lower (Q), since to it corresponds the minimum pressure at which the reduction of the metal to ferrous oxide (FeO) begins to take place. The second point, O, corresponds simply to a change in the "degree of oxidation" of the metal—in the sense that, at pressures higher than that corresponding to it, the stable oxide is no longer the ferrous oxide (FeO) but the magnetic oxide (Fe_3O_4). It is plain how, from the practical standpoint (in the present state of our knowledge, at all events), this change fails to present any particular interest.

case-hardening the higher the pressure beyond which the oxidation which accompanies the case-hardening process begins to take place (in its two degrees). In other words, the higher the temperature the wider the range of pressures within which the case-hardening may be accomplished with the firm assurance that no oxidation of the metal will take place.

A comparison of the results of several of these experiments, taken two by two from among those carried out on identical steels, confirms this assertion,* and the same fact is again clearly demonstrated by the following experiment:—A small cylinder of ordinary soft steel, 60 centimetres in length and 1 centimetre in diameter, is subjected to case-hardening for about three hours in the accustomed apparatus and with the customary "case-hardening mixture," at a pressure of 15 kilogrammes per square centimetre. Care is taken to embed it along its entire length in granular carbon. The temperature, maintained constant between 980° and 1,035° C. over a belt of about 20 centimetres near the medium part of the cylinder, decreases gradually (in accordance with the structure of the apparatus which we have described) towards the two extremities, until it falls at the very ends to about 500° C. The surface of the cylinder which is thus treated—case-hardening being unmistakable in all that portion where the temperature has exceeded 800° C.—is absolutely unaltered in its central hottest part (which is also the most intensely case-hardened), while in those portions that have been subjected to a lower temperature the surface is coated with the characteristic layer of compact oxide in which are embedded the fragments of charcoal. Beneath the oxide layer the steel is still case-hardened, in all those portions, at least, where the temperature has exceeded 750° to 800° C.

The two pressures of equilibrium vary also, as we have already pointed out, concurrently with the variation in composition of the steel which is subjected to case-hardening. Since in this case curve 5 undergoes no deviation (for it must in every instance correspond to the conditions of equilibrium of the gaseous mixture and the wood charcoal), the displacement of points O and Q will be exclusively determined by the displacement of the two straight lines 3 and 4 (see diagram, Fig. II.). Now, it is well known that these two straight lines deviate the more towards the left, the purer the metal which is being subjected to the action of the gaseous mixture. And since the deviation of these lines carries with it the displacement of points O and Q, it follows that the purer the metal the wider is the range of pressures (lower than those corresponding to point Q) within which we may conduct the cementation with the firm assurance that oxidation of the metal will not take place.

Among the metals which frequently alloy with iron and are capable of forming mixed crystals with it, are manganese, chromium, and nickel. The first two (manganese and chromium) are

*Thus, for example, the results of Experiment VII. may be compared with those of Experiment VIII. (and let it be borne in mind that in the latter the velocity of the gaseous current is double what it is in the former).

less pure than iron; therefore, as their concentration in special steels increases, the temperature remaining unaltered, the two pressures O and Q (above which cementation is perforce accompanied by the two degrees of oxidation of the metal) become gradually lower and lower. For chromium steels, a comparatively small proportion of chromium (from 2 to 3 per cent.) suffices, because the above-mentioned pressures, at temperatures which

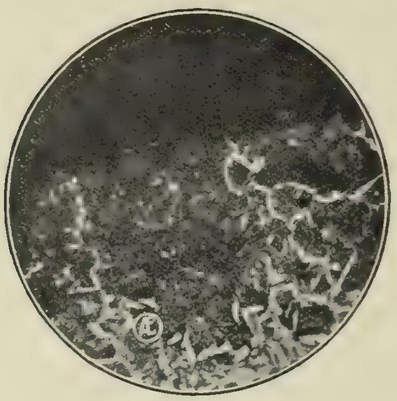


Fig. III.

are not too high, decrease below ordinary atmospheric pressure. This explains the results recorded by Charpy, and demonstrates that they are by no means due to the fact that the gaseous mixture of carbon dioxide and carbon monoxide acts separately on the iron and chromium.

Nickel, when alloyed with steel, has an effect precisely contrary to that due to chromium and manganese, imparting to the solid solutions into which it enters with iron the character of a purer metal than undiluted iron. Therefore, the higher the proportion of nickel in the steel subjected to case-hardening the higher are the pressures which may be reached in case-hardening with the "mixed agent" without any risk of oxidation of the metal.

In special steels also the phenomena which we are considering produce effects

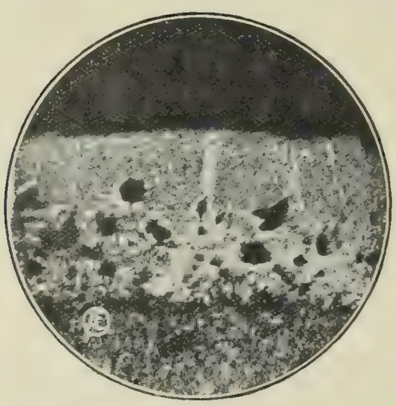


Fig. IV.

analogous to those which we have described in the case of ordinary carbon steels. Thus, for example, the photomicrograph reproduced in Fig. IV., represents (at a magnification of 185 diameters) a strip of the external border of a plane section, perpendicular to the axis of the cylinder of chromium-nickel steel, case-hardened in Experiment XXIV.† Here also the case-hardened metal is coated with a layer of oxide in which particles of carbon are embedded.

The photomicrographs reproduced in Figs. V. and VI., taken under the same conditions as the foregoing, at magnifications of 65 diameters after elimination of the layer of oxide coating the cylinder, illustrate the intensity of case-hardening undergone by the chromium steels which formed the object of Experiments XVIII. and XX., while at the same time these steels, as we already remarked in Table I., become considerably oxidised.‡

The same facts are noted in connection with nickel steels, when the operation is conducted at pressures sufficiently high to produce oxidation of the metal. This is well shown in the photomicrograph reis well shown in the photomicrograph reproduced in Fig. VII. (Experiment VII.), taken under the customary conditions and magnified 65 diameters. From all that we have said it is evident that the process of the case-hardening of special steels by means of the "mixed agent" is subject to the same conditions as those which attend the case-hardening of carbon steels. The differences for the various steels are purely quantitative, whether in regard (in accordance with

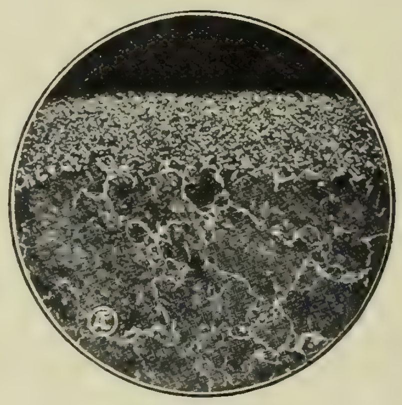


Fig. V.

what we have already said) to the pressures of oxidation, or in regard to the concentration of the mixed crystals in equilibrium, along curve 5, with the gaseous mixture $(CO+CO_2)$ which in its turn is in equilibrium with the charcoal.

All the considerations developed in the foregoing pages remain valid until phenomena of metastable equilibrium cease to occur, and on condition that during each case-hardening operation the temperature is maintained constant. Now both the first condition (to which should be added the certainty that the low velocity of the various reactions shall not impede the attainment of the status of equilibrium) and the second are extremely difficult to realise in practice; in fact, it may be said that they never are to be found in industrial practice. The consequence of the imperfect attainment of one of these conditions (and above all of the second), or of both, is the possibility that, during one or more of the phases of case-hardening, the point representing the system in which the reactions take place will be situated in a quadrant of the diagram where stable equilibrium could not have been reached. For instance, it is quite possible that, as a result of fall in temperature, the above-mentioned representative point will be found in the right-hand quadrant and above curve 5 (see diagram, Fig. II.) In such a case, reactions different from those which we have been so far considering

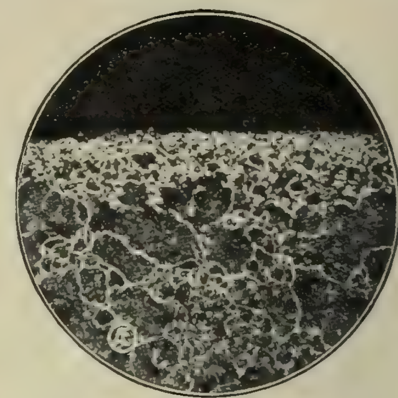


Fig. VI.

may occur; as, for example, the decomposition of carbon monoxide, accompanied by the liberation of pulverulent carbon. Moreover, the representative point may meet one of the curves 1 and 2 (the significance of which has already been indicated), and these, in their turn, may undergo considerable deviations in consequence of changes in temperature. In this last case, the reactions which may take place are much more complex.

Before concluding this note, we desire to draw attention to the fact that, for the considerations set forth in the foregoing pages, many useful practical axioms may be deduced for the purpose of carrying out, on a well-reasoned basis, the case-hardening of special steels with the "mixed agent." In proof of which we may merely cite one example: the case-hardening of steels with rather high percentages of chromium (more than 4 or 5 per cent.) with "mixed cement" at a temperature not higher than 1,000° C. (a limit which in many cases cannot be ex-

Fig. VII.

ceeded) is accompanied by considerable oxidation of the metal, even when the process is conducted below ordinary atmospheric pressure. Now, the results of our experiments, and the considerations developed in the foregoing pages, suggest immediately a method of avoiding this oxidation; it will suffice, indeed, to reduce the pressure of the carburising gaseous mixture $(CO+CO_2)$ below the minimum pressure of oxidation (Q) characteristic of the particular metal at the temperature at which it is being case-hardened. This can be in many cases easily accomplished by "diluting" with air the carbon dioxide which is circulating through the mass of granular carbon. In this way the carburising gaseous mixture acts under partly reduced pressure, since it is "diluted" by nitrogen.

†Etched by an alcoholic 5 per cent. solution of picric acid.

‡We shall have occasion, ere long, to deal in another paper with the causes of the special distribution of carbon within these case-hardened steels (in which as we have seen, the carbon tends more particularly to accumulate towards the surface).

MOTOR OMNIBUS MANUFACTURE.

Some Impressions of a visit to the works of the London General Omnibus Company.

By H. Burchall.*

One of the impressions a visitor to the works of the L.G.O.C. comes away with is that floor space is used to the very best advantage. No department appears crowded, and yet the output of finished chassis, twenty per week, is greatly in excess of what one would expect from a works of its size. This output is more easily appreciated when one considers the short time the well-known "B" type 'busses have been on the road, and the number of routes worked solely by this type. The output is all the more remarkable as only one department—the machine shop—is working overtime.

If the number of chassis built per week is to be maintained, it becomes a source of speculation as to whether the whole of the factory will be necessary after the full complement of 'busses is built. Estimating the number which could profitably be run in London at, say, 2,500, it follows this number would be built, at the present rate of working, in two and a half years. The output will probably fall off in proportion as the number of vehicles on the road increases, to compensate for the making of parts for replacement. If then, another year is allowed for the manufacture of the 2,500, it would appear that the life of a 'bus is estimated at three and a half years, or probably about 70,000 miles.

This appears to conform with American ideas regarding output and the advisability of scrapping machinery rather than doing extensive repairs.

Repair and overhaul work is not done at the chief works, but each depot does its own repairs. This method of working is facilitated by the supply to the depots of a complete set of the blue prints required for their own particular chassis. Every part of the chassis—stamping, casting, forging, etc., bears a number corresponding to the number on the drawing. By this means any difficulty in ordering spares should be avoided.

In a works dealing with heavy and bulky parts, such as are found in 'bus work, one would expect to see some form of tramway for the quick and easy handling of material in the various stages of manufacture. The passages between the various shops do not appear suitable for any form of small-wheeled trucks, as they are open to the weather, and are at present composed only of earth, and naturally become very uneven and dirty on rainy days. However, the absence of some such method of handling may be due to the unfinished condition of the works. In the shops, too, the arrangements for handling do not seem as complete as in many factories dealing with much lighter work.

Details of Manufacture.

The works seem to be run on a scientific basis and not on rule of thumb methods. A laboratory is equipped as a part of the factory organization; this feature, although fairly common on the Continent, is rather the exception than the rule in this country. The company evidently believe not only in the analysis of material, but also in the advisability of experimenting. A good example of this is to be found in the change speed mechanism, on which, it is reported, huge sums of money were spent before the silent chain gearbox was finally adopted. It is evident that this experimenting has been most successful in developing a transmission for a heavy 'bus running on solid tyres which, for quietness and sweetness, can be equalled by very few makers of touring cars.

It is usual in most factories to find some process or detail in manufacture not generally adopted. In this works unusual methods of working are more numerous. For instance, in the hardening shop it was noticed that in order to avoid hardening the parts of an article which were required soft, they were copper plated before being packed in the carbonising material. This method, which is almost universal in America, is looked upon as quite satisfactory.

Oxy-acetylene welding is used on quite a number of details amongst which might be mentioned the oil trays on gearboxes and back axles. The hot-air jacket for the carburettor is also welded to the exhaust pipe by this method. The foundry is equipped for casting in brass, gunmetal and aluminium, all iron castings being obtained from outside specialists. Die castings are made in aluminium, and the clean finish of the products is particularly noticeable. The engine-testing department is equipped with an engine from one of the older type 'busses, driving shafting from which engines can be run in by a belt round the flywheel before being tested under their own power.

Peculiar Conditions which Modify the Design.

Probably one of the chief reasons that has led to the production of such a fine piece of engineering as the latest "B" type chassis is that the designers are not limited by any question of popular practice, or whether the chassis will sell. Nor do they have to consider alterations in the design for a new season's model. The only limitations imposed upon them are those of the police—which, after all, are only what the designer should impose upon himself, and the prime cost of the vehicle. That the number to be made will run into thousands, when once the design is accepted, has a very great bearing on cost.

The chief chassis features upon which the police insist are maximum weight, silence, clearance, safe design of steering gear and brakes, and the prevention of smoky exhausts and oil droppings. Although all these stipulations are made for the convenience of the public, they are very good for the development of the chassis. Vibration is the cause of noise, and wear is the effect, so it follows broadly that the quieter a chassis runs the longer it will last. High clearance acts for the company by producing a design which is accessible without the use of a pit. The advisability of safe steering design is too obvious to need comment, and the prevention of oil droppings and smoky exhausts leads to economy, both in oil and, indirectly in maintenance, through less trouble from carbon deposit.

In running a fleet of vehicles at no great distance from a base, and having them back in the garage in skilled hands at least once each day, it is possible to make provisions against damage or maladjustment by unskilled drivers, which are impracticable under ordinary conditions. In the latest type of 'bus, the most likely part to suffer from undesirable attention is the magneto, and this is encased in an aluminium box which is locked up. This appears rather a daring step, but goes to prove the wonderful reliability of a modern high-tension magneto if left alone.

The whole design of the 'bus tends to reduce the number of things requiring attention by the driver. The engine control is by pedal only, and the lubrication is quite automatic. A 'bus engine is run light for a considerable portion of its working time, and to economise in petrol a carburettor has been fitted having two jets, one of which is specially suitable for running at low revolutions and without load.

Particulars of Design.

In reviewing the design, probably the point most worthy of comment is the precautions taken to make the 'busses silent. Silent chain drive is adopted for the camshafts and magneto pinions, and the valve tappets are quietened by the insertion of a rubber buffer between the valve stem and the tappet, the design of which is particularly neat, it being made so that admission of oil to the rubber is prevented. Incidentally, valve covers, now so popular on touring cars, and which were fitted to the "X" type (the last model but one) have been abandoned on the latter type, the reason given being that they do not make the valves any quieter, that appearance is no advantage, and that it eliminates an unnecessary part.

The gearbox, although very silent, and allowing a neat change of gear, appears to be rather costly, as the chain wheels, which run on ball bearings, are locked to the shaft by internally cut teeth meshing with ordinary spur pinions. The road wheel bearings are of a most unusual design, having a floating sleeve rotating between the plain hub and axle. The drive to the back wheels is taken through hexagon ended shafts, not by the more usual castellated type, and the frames are not made in accordance with generally accepted standard practice, but of wood with steel flitch plates; the reason for this is not quite apparent, particularly as the output is so great. The leather-faced cone clutch has been retained, but its diameter is very great, and laminated springs are used to engage the clutch instead of the more common coil spring.

The piston and connecting rod appear to be very heavy for a 110 x 140 engine, but otherwise the design does not call for any special comment, while the performance of the 'busses on the road is sufficient testimony to the care and thought expended on the design and manufacture.

*Hon. Sec. the London Graduates' Section of the I.A.E.

VARIOUS USEFUL JIGS & FITTINGS.

By C. T. Schaefer, M.S.A.E.

The following is an endeavour to show some solutions to the various small problems which frequently occur during the manufacture of motor car components. They are interesting when compared with the various machining articles which have appeared from time to time in the "Automobile Engineer." Fig. I is a drawing of a jig used when testing cast iron pistons for porosity after machining. As will be seen, the piston in question is held down on its seat by the spider and is seated on a soft washer to prevent any leakage, the spider itself being adjusted by a hand screw working in a collar hinged at either side by a couple of pin joints. When satisfactorily bedded down, water is admitted to the piston through the $\frac{5}{8}$ in. pipe in the centre of the base, leading to a $\frac{3}{8}$ in. hole drilled at right angles to the delivery pipe, itself fitted with a tap for adjusting the water pressure. On the right-hand side of the admission pipe will be observed another hole of smaller dimensions drilled down into the base until it reaches a pipe leading to the outside of the casting. This pipe being also fitted with a tap allows the escape of the water when each piston has been tested. Of course, when filling the piston it is necessary to allow all air to escape from the interior, and accordingly a small $\frac{1}{8}$ in. pipe is erected in a vertical position inside, and made of such a length that it almost touches the top of the piston. A lead is then drilled through the base of the jig until it connects the socket of this pipe to the outside air, at which point a stop tap is placed, which is closed immediately water shows signs of its presence from the tube. In operation the jig is exceedingly simple and easy to handle, while any porosity very quickly shows with the minimum amount of the mess usually associated with water tests.

Fig. II. is a jig which has been evolved for machining the spring seating of a live axle car. In this case a large boss has been machined on the jig to take the axle bracket, when it has been bored, and the whole spring bracket is held thereon by a large U washer and hexagon nut. The reason for using this particular type ot washer is that half a turn of the nut will allow the washer to be removed and, the hole in the spring bracket being larger than the hexagon nut, the whole bracket can be withdrawn without disturbing any further portion of the jig. The lever is adjusted by the two hand screws plainly seen in the vertical section, while the full thrust of the milling cutter is taken on a couple of set screws which are in the upright part of the jig and visible in the plan view of the whole arrangement. Such an arrangement is neat and extremely easy to set up, especially as it is provided with a tongue which will fit into the groove on the bed of the milling machine and prevent any inequality or bad levelling up. On the other hand there is a possibility that the careless workman might set the spring bracket, by means of its adjusting screw, in such a manner that the flat surface would not be obtained and the job scrapped in consequence. Once the jig has the job actually in position it is perfectly rigid and most unlikely to give in any direction under the action of the milling cutter.

Now that jaws are so much used in brake and change lever connections a great many firms have found it very hard to settle which method of machining is the least expensive and most rapid. Fig. III. shows a simple form of open-ended box jig suitable for such a joint as the one shown in the drawing. The jaw is simply placed in position, held at one end by the adjustable bush which drills the hole for receiving the rod, and it is then held up in position by a wedge inserted underneath the jaw and holding the latter in such a position that the drill bush in the jig comes exactly in the correct position for drilling the pin hole in the jaw. The idea of the larger adjustable drill steady bush is to allow for any inaccuracy which may occur in the stamping from which such details are likely to be machined, as otherwise with an immovable bush there is likely to be considerable inaccuracy in the drilling, and the whole job will most probably get scrapped. Such a jig can be manufactured from 2 in. x $\frac{3}{8}$ in. mild steel with a few machine screws, and the total cost is extremely low. It is perfectly satisfactory for dealing with the drilling operations on a jaw such as the one described, and is of such dimensions that it can quite easily be held by the operator without the necessity of adjusting clamps to the various grooves of the drill table.

In connection with these jaws, there is another small operation on which a great deal depends, viz., the drilling of the hole which is to receive the split pin in the large coupling pin used at the end of the jaw. If these are drilled without a jig sooner or later the man who buys the car in question is sure to find a lot of shake and to be displeased with the car accordingly. On the other hand it is equally likely to be drilled in such a position that a split pin of the proper dimensions cannot be inserted in the hole, the troubles of which arrangement are obvious at once. With the aid of a small piece of hexagon tool steel and an ordinary taper pin, it is possible to construct a small jig which will deal adequately with the operation in question.

Fig. IV. shows a jig constructed in this manner. It will be noticed that the pin is made slightly taper at the end which has to be drilled, and is inserted into the jig so that the head rests on a collar and is itself held firmly in position by the action of the taper pin and the wedging effect of the taper joint pin end. The drill is, of course, run through the jig in the ordinary manner. It is impossible for this arrangement to drill a hole otherwise than in exactly the same position, provided the joint pin has been correctly machined beforehand. Even should the taper pin work slightly loose, it is exceedingly unlikely that the joint pin would shift during the short time necessary for drilling the hole.

Fig. V. shows a small device which can be used for winding coil springs and manufacturing them at a very small prime cost. It takes the form of a small piece of metal with a circular orifice. This fits on the bar on which the wire is being wound, and has a small groove which may act as a steady during the winding. As a general rule it will be found that such a device will be of great value when it is necessary to manufacture springs in an ordinary automobile factory, though fortunately such an occasion is extremely rare.

As a fixture which may be used for steadying an exceptionally long camshaft during the machining of the cams when the latter are solid with the shaft, Fig. VI. is a good example. Here the six-cylinder camshaft shown above is located by vee blocks and a couple of hexagon bolts to each of the bosses seen along the fixture. Between each boss there are grooves of suitable dimensions that take the rough circles from which the cam must subsequently be milled. In the sides of the base there are grooves to take the ordinary form of holding down arrangement shown in the end sectional view of the device in question. It is intended that each cam shall be formed separately and that the brackets which hold the cam-shaft in position on the bosses shall secure the cam in the firmest possible manner at either side of the cam which is being machined; while a further set of clamps are provided so that these can be put in place by the operator during the time the milling cutter is busy on the previous cam. Of course, the advantages of such a jig depend entirely on the form of camshaft which is adopted. It is not nearly so necessary to provide an elaborate fixture if the camshaft has been stamped roughly to the form which it will assume when machined, and as this is a practice likely to become prevalent in this country, it is possible that the particular type of fixture described will not be of much service.

Fig. VII. cannot exactly be described either as a jig or a fixture, but it is of sufficient interest to warrant attention, and shows one of the ways by which costs may be reduced and the productive speed of a machine run up. It will be observed that care has been taken to fit the turret with attachments capable of holding a couple of tools at suitable distances apart. The job shown in the illustration is that of completely machining a rough flywheel casting. In the first diagram the turret is machining the boss which will later on receive the flywheel bolts, the inside face of the flywheel and the edge of the outer rim, while the ordinary tool carrier takes care of the machining necessary on the outer rim itself. Diagram 2 shows the turret revolved two faces and the roughing cutter in its carrier machining the cone surface necessary to receive the clutch. At the same time the ordinary tool carrier has been moved until that tool which machines the back of the flywheel has been brought into operation. The third diagram shows the finishing tool necessary for the clutch surface, whilst diagram 5 indicates the time at which the centre hole is bored.

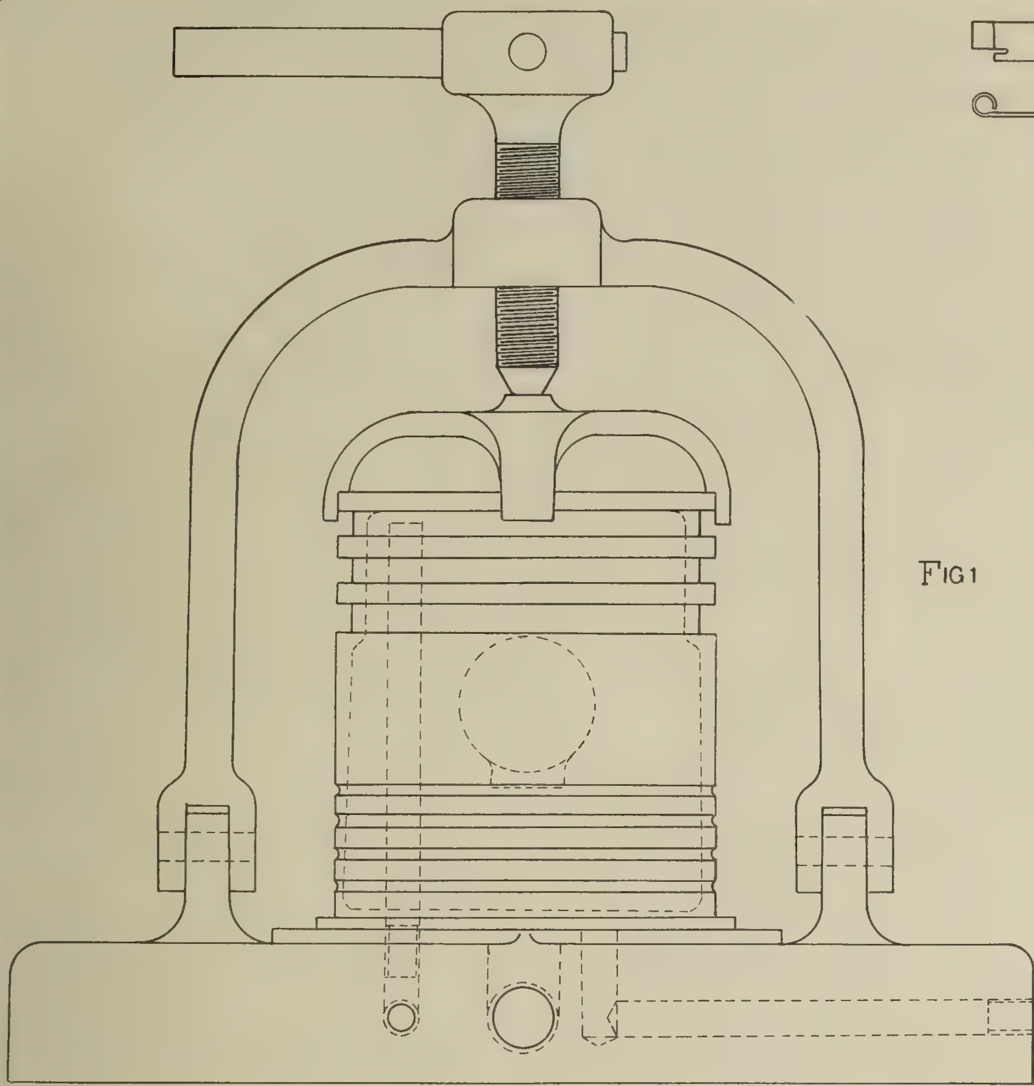


Fig I

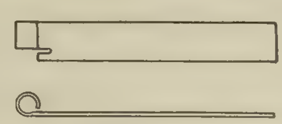


Fig V

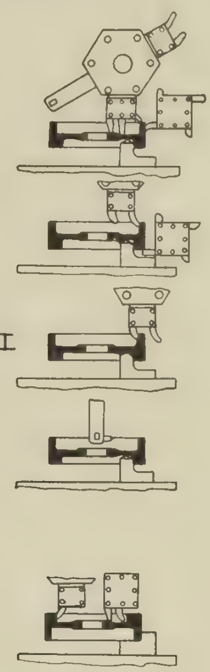


Fig VII

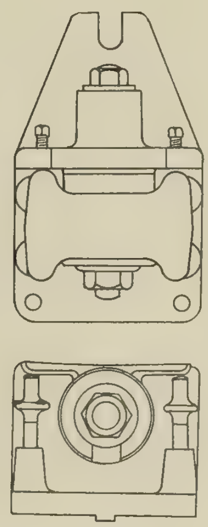


Fig II

Fig IV

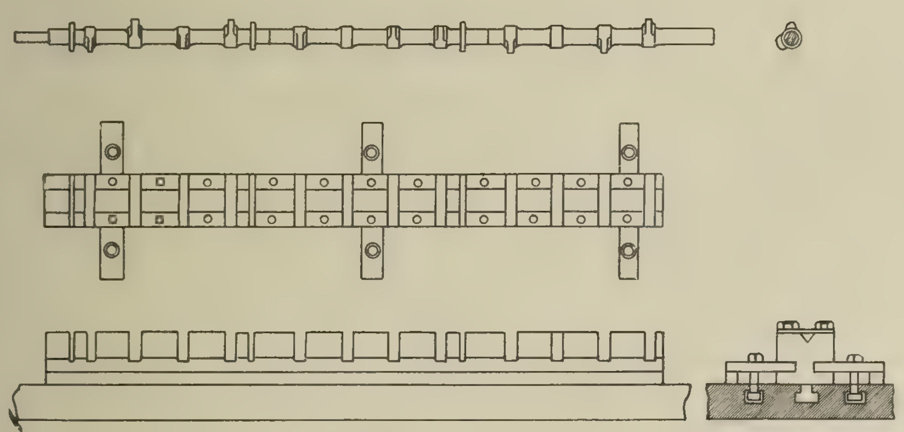


Fig VI

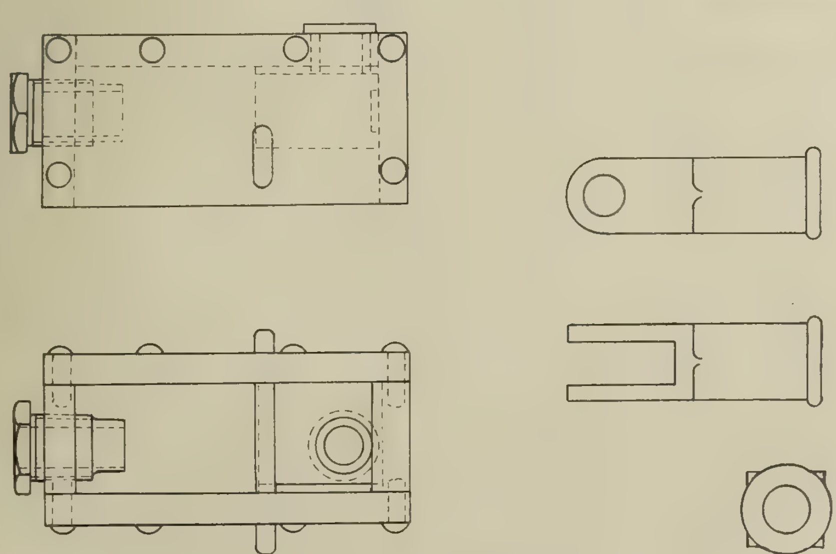


Fig III

VARIOUS USEFUL JIGS AND FITTINGS.

(see page 536).

VISIT TO ENGLAND BY AMERICAN AUTOMOBILE ENGINEERS.

On Monday next (November 6th), a considerable party of members of the Society of Automobile Engineers will land in this country, where they will be the guests of our own Institution of Automobile Engineers. The latter body are not only entertaining the visitors at a dinner to be held on Saturday, November 11th, but have been successful in arranging a joint meeting on the evening of Wednesday, 8th, when Mr. H. E. Coffin, of the Hudson Motor Car Co., will read a paper on "Chassis Design." It is hoped that some of the visitors will also give their impression of things European as shown at Olympia.

Many prominent manufacturers here have invited the party to visit their works, and the programme arranged by the Reception Committee of the I.A.E. is as follows:—

Nov. 8th.—Paper by H. E. Coffin at the Institution of Mechanical Engineers, 8 p.m.

Nov. 9th.—Visit to the works of the London General Omnibus Co., by permission of Mr. Iden, the chief engineer.

Nov. 10th.—Visit to the National Physical Laboratory, Teddington.

Nov. 11th.—Visit to the cab garage of W. and G. Du Cros, Ltd., and the dinner given by the I.A.E.

Nov. 13th.—Inspect Humber factory at Coventry, and also the works of Alfred Herbert, Ltd.

Nov. 14th.—Visit the Daimler works at Coventry, and possibly also those of Rudge Whitworth, Ltd.

Nov. 15th.—Inspect the Wolseley works at Adderley Park, Birmingham, and various other factories.

Nov. 16th.—Visit the Sunbeam works at Wolverhampton, and travel to Manchester.

Nov. 17th.—See the works of Hans Renold, Ltd., and return to London.

On Sunday, November 12th, the visitors will be the guests of the Ford Motor Co., who are taking them to Brooklands track, which will be opened to them by courtesy of the B.A.R.C. and Major Lindsay Lloyd, while on Saturday, November 18th, the party cross to Paris, under the guidance of Mr. R. W. A. Brewer, where several works will be visited, including Panhard and Levassor, Lemoine, and De Dion. The party propose to disperse about November 23rd.

As an example of the great activity of the American engineers, the organisation of such an extensive excursion may perhaps be taken as typical. Undoubtedly it is proof positive of the important place automobile engineering is taking in the work of the world.

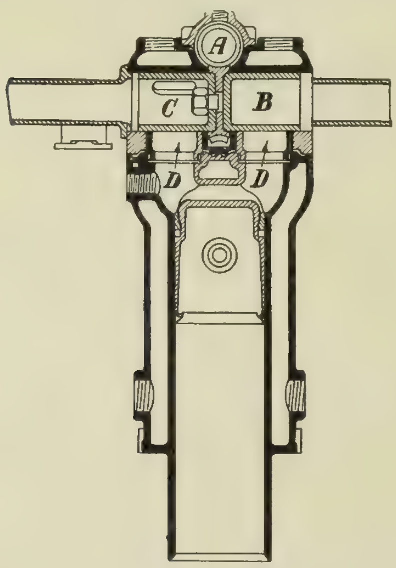
The joint meeting between the visitors and the I.A.E. ought to provoke one of the most interesting discussions in the whole history of motoring, for never before have accredited representatives of the two nationalities met together in any number. We believe this occasion is likely to prove the commencement of a closer understanding between engineers in our own particular branch—such an understanding as can only help towards the increase of knowledge in mutually useful directions.

RECENT AUTOMOBILE PATENTS.

By Eric W. Walford, F.C.I.P.A.

Rotary Valve Engine.

The cylinder is provided with a detachable head, across the top of which runs a shaft A carrying a worm gear which meshes with a worm wheel, to which is attached two tubular rotating valve members B and C. Each of these is slotted

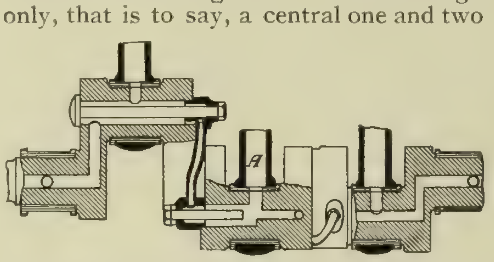


and open at one end, the open end communicating with the induction or exhaust pipe as the case may be, whilst the slot is adapted to register with the passage leading to the combustion chamber at the correct periods. The casing containing the worm gears is fed with lubricating oil, which passes therefrom to the crank chamber. The valves B and C are maintained gas-tight with the cylinder by means of throat pieces D, which are pressed up against the cylindrical valves by spring pressure.

No. 25,546/10. F. Lamplough.

Crankshaft Lubrication.

In six-cylinder engines in which the cranks are arranged in three bearings only, that is to say, a central one and two

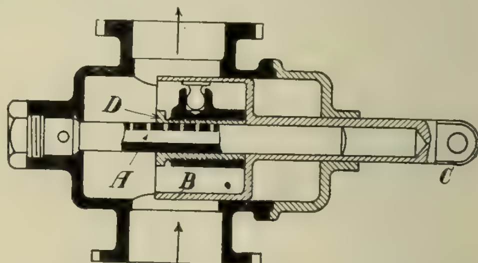


end ones, difficulty is experienced in feeding lubricating oil under pressure to the centre crank A of each set of cranks. To the outer cranks the oil can be fed from the main crankshaft bearings and forced radially outwards along the crank web to the crank pin. Centrifugal action prevents the oil passing down the next crank web to the middle crank bearings unless the oil pressure is very high. This is overcome by connecting the centre crank pin with the adjacent pins by means of tubes A, which are curved and of the same radius throughout, which is that of the throw of the crank pin. By this means the oil passes in a circular path and is unaffected by the centrifugal pressure.

No. 10,629/11. Société Anonyme des Automobiles Delaunay Bellville.

An Interesting Carburettor.

The carburettor is of the single lever type, petrol being supplied to a horizontal tube A through which it issues by small holes drilled through the upper wall of the tube. The air passes through the carburettor in the direction of the arrows, being controlled by a sliding piston B, one edge of which controls the air inlet and the other the gas outlet, the piston being operated by a lever attached to C. Adjustably connected to the piston and taking a bearing on the tubular jet A, is a slide D, which uncovers the jet outlets in proportion as the throttle is opened. Relative adjustment of the slide to the piston throttle enables a certain variation of results to be obtained, and furthermore, adjustment to suit different engines, or to provide different results, can be obtained by varying the spacing of the jet nozzles.



Thus the first few holes uncovered would be arranged close together to provide quick acceleration, and so forth.

No. 27,531/10. E. P. Everest.

Balancing System For Crankshafts.

This patent specification deals with the balancing of high speed engines to prevent the synchronous vibrations which occur. As the remarks of the inventor in his specification are of considerable interest, I cannot do better than quote them at some length:—

"The present invention relates to improvements in high-speed reciprocating engines, and to an improved method of apparatus for eliminating certain forms of vibration in the running of high-speed multi-cylinder engines.

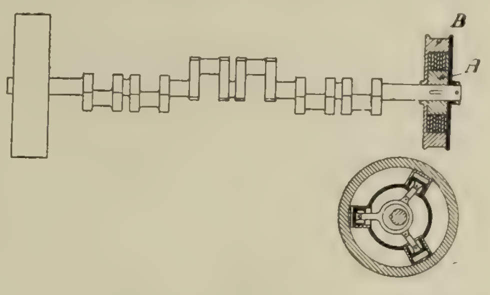
In apparatus requiring means for the elimination of torque variation it has been common to employ heavy flywheels for the purpose, such fly-

wheels being attached to their shafts by non-positive connections, such as springs, or cylinders and pistons co-operating through an elastic fluid, such elastic means being adapted to absorb excess energy during certain periods and to restore such energy to the system during certain succeeding periods. Again, when a heavy flywheel has been employed for eliminating torque variations in apparatus subjected to sudden stoppages, it has been common, in order to prevent rupture of the shaft, to attach the flywheel to its shaft by a device capable of slipping, such as a friction clutch.

In high-speed multicylinder internal combustion engines the elimination of torque variations is effected in the usual way by the use of a heavy flywheel secured to the crank shaft. Such crank shafts present a new problem to the designer, however, for at certain speeds of rotation it is found that objectionable, and sometimes dangerous, torsional vibrations are set up, and the object of the present invention is to eliminate such vibrations.

A crankshaft for an engine with several throws is a very elastic body, and is commonly loaded at one end, or near one end, by a flywheel of great moment of inertia, and is further loaded at different points along its length by inertia masses in the form of pistons, connecting rods, etc. These inertia masses act to a certain extent in the manner of a second flywheel, and this is especially the case in a six-throw crank. The crank shaft, therefore, is equivalent to a torsionally elastic shaft with a distribution of bodies having moments of inertia along its length, and it will be seen that under these conditions, torsional vibrations may be set up by any suitable exciting torque.

In multicylinder reciprocating engines, particularly internal combustion engines, such torques exist in the intermittent impulses of the explosions, and in the inertia effects of the re-



ciprocating masses. The piston and connecting-rod in reciprocation bring about a series of four alternations of torque on the crank shaft during each revolution. If the vibration were strictly of the character which exists when a single length of shaft is fitted with two flywheels rigidly attached, the resulting vibration would not be felt by the casing of the engine, because the whole of the torque variations in such a system are self-contained in the rotating elements, and therefore the present invention does not apply to devices where parts in pure rotation are concerned. Where, however, the second rotational part exists partially or wholly in the form of reciprocating elements, the individual variations in the energy of these reciprocating elements result in considerable vibration being set up, following the vibrations of the rotating system, but communicated to the stationary system of the engine. I have found that this form of vibration causes great trouble in the case of six-cylinder engines, and results in the production of violent vibrations, which are transmitted to the body of the car at and above certain definite speeds.

The object of the invention is to destroy the synchronous vibrations above referred to, and so to render an engine of light weight and comparatively non-rigid construction, free from vibration periods such as are commonly met with in engines at present in use.

The invention is carried out by providing the crankshaft at the front end with the drum A, on which is mounted a flywheel B, coupled to the drum by a plate type of clutch which affords considerable frictional surface, and it is preferred that there be an oil film between the contracting surfaces. As an alternative, the flywheel B is connected to the crankshaft through a form of hydraulic clutch which comprises radial pistons, cranks and connecting rods, the flywheel being thus connected to the shaft through the liquid in the clutch.

No. 21,139/10. F. W. Lanchester.

CORRESPONDENCE.

SOME FACTORS IN TYRE ECONOMY.

Sir,—Whilst comparing the merits of the wire wheel with the old type of wooden wheel, Mr. Mackle might at the same time have taken into consideration the hollow steel wheel, such as the Sankey, which is probably more popular than the wire wheel, and undoubtedly has many features which the wire wheel does not possess. It is lighter than the wire wheel, stronger laterally, does not require such a long hub or such ugly arrangement of the outer spokes as does the wire wheel, and possesses the great advantage that the wheel can be applied to the axle either side out. This at first thought may not seem a big advantage, but if one considers that the majority of front wheels are arranged at a slope, and that consequently the wear takes place along one edge all the time, it will be at once seen how additional mileage can be obtained by turning the wheel round, which at the same time runs the tyre in the opposite direction, reversing the stresses.

I make the practice of doing this with the Sankey wheels on my car, and I know the advantage of it.

With reference to the Daimler Co.'s hire department, it has been stated that the tyres in question were not run at the same pressures throughout. Before Mr. Mackle's figures are taken as accurate it would be interesting to have a definite statement from the Daimler Co.'s hire department as to whether the air pressures were the same, and, if so, under what pressures the tyres were run.

I hold no brief for the steel wheel. I have them fitted to my present car, and I have had considerable experience of both wire and wooden wheels. The security of attachment of the steel wheel and its ease of cleaning are great points in its favour.

STEEL WHEEL.

TWO CYCLE ENGINES.

The next ordinary meeting of the Society of Engineers (Incorporated) will be held at the Institution of Electrical Engineers, Victoria Embankment, on November 6th, at 7.30 p.m., when our contributor, Mr. R. W. A. Brewer, will read a paper on the above subject. Mr. Dugald Clerk, F.R.S., who is an authority second to none on the above subject, has promised to be present and to open the discussion. Engineers interested in the subject of the paper can obtain tickets of admission from the Secretary of the Society, 17, Victoria Street, S.W.

The following is a resumè of the paper :—

The paper opens with reference to the early work of Mr. Dugald Clerk dating back to the year 1877, and the development of the Clerk two-cycle engine, and attention is drawn to the system of sub-dividing the charging side of the engine into two separate portions—an air pump and a fuel pump. The large development of the Clerk engine is then referred to in the consideration of the modern two-stroke engines actuated by producer or blast furnace gas. The facilities for scavenging such an engine are briefly referred to.

Coming now to the modern development of the two-stroke engine, a certain number of advantages which this type of engine has over the four-stroke engine are enumerated and discussed, and it is shown how that theoretically certain methods of operating the charging of the working cylinder are superior to other methods, and special reference is made to the distribution of the inlet and exhaust ports and the direction of flow of the working fluid through the engine during its cycle of operations. An example is taken of one type of engine in which the charging and scavenging pumps, as well as the working piston, are comprised in one reciprocating part, with the necessary valves. Various arrangements of cylinder and pistons are then discussed, and the possibility of obtaining complete scavenging with the various arrangements.

Reciprocating and rotary charging devices are then touched upon, and examples are given of engines working on both these systems. The thermal efficiency of the two-cycle engine is dealt with at some length, and comparative figures are given of several types of two-cycle engines. The author gives the results of a number of tests which he has carried out with a certain type of two-cycle engine, and by means of the exhaust gas analysis, makes various deductions as to what has taken place within the working cylinder, the loss of charge which has passed through the engine unburnt, the excess of air in the explosive moisture, and the mean effective pressure in the working cylinder referred to the b.h.p. The paper concludes with a description of several types of two-stroke engines which are being built, some of which have given satisfactory results in working

I.A.E. GRADUATES' SECTION.

The first paper of the session was read before the London section by Mr. C. E. G. House, the subject being "The History of Automobile Construction." The author said that the following were amongst the more important detail improvements seen during the last few years :—

Compression is much higher than formerly. Ignition has entirely changed. The magneto is supreme, the coil and accumulator having been almost entirely superceded except for the Lodge system. Low tension is now unknown, as is tube ignition and flame ignition. Carburettors are all of the jet type, with one exception, which is of the wick type. The control is much improved, the auxiliary air and spark advance being automatically operated in most designs. The systematic study of carburation and the vast number of experiments made have greatly improved the sensitiveness of the carburettor to engine speed. Lubrication is now very reliable and the cooling efficient. Wheels have tended to become smaller and tyres larger. The steering mechanism is now very easily moved by one hand, and the details have been greatly improved. The sliding system with gate selecting mechanism has superseded older patterns, and the design of the teeth and shafts is much better. The system of final drive has gone from chains to bevels and finally to worm gearing, which appears to be slowly ousting bevels at the present time. The springs are now of a much flatter camber, are broader, and are usually supplemented by light springs or vibration dampers. The frames are almost universally of pressed steel. The brakes are much more efficient, and in a few examples are fitted to front wheels in addition to the rear wheels.

By far the most successful concert ever organised by the graduates of the Institution of Automobile Engineers took place on Friday, 13th October, at the Holborn Restaurant. Mr. F. W. Lanchester, the president of the Institution, was in the chair, and was supported by Dr. H. S. Hele-Shaw, F. Leigh Martineau, Esq., and about 130 graduates and friends.

During the evening a presentation was made to the late hon. secretary of the London branch of the Graduates' Section, this taking the form of an illuminated address.

STEELS.—Managers responsible for the buying or application of the various forms of new steel to a works will be interested in the pamphlet issued by the Pennsylvania Steel Co., which states thoroughly the advantages of a new chrome nickel alloy known as "Mira" steel. Details of the analysis and heat process used in the preparation of this steel for special purposes are given, together with a series of curves dealing with the various temperatures and quenching.

FACTORY EQUIPMENT.—With shops which are mostly fitted with electrical machine driven plants, it is occasionally necessary to thoroughly clean the motors, and in all cases to avoid the entry of over much dust into the machines themselves. For this purpose a small electrically driven compressor has been put on the market by Lacy Hulbert and Co., 91 Victoria Street, W. The pump is in portable form, and is specially designed for thoroughly cleaning the generator by the vacuum cleaning process. The same firm also make stationary air compressors of various types, pamphlets of which can be sent on application. The one dealing with road tools is probably the most useful to the automobile trade as containing a considerable number of particulars of a type of pump commonly used on motor cars.

TWO NEW GRINDERS.

The illustration below shows two new grinding machines recently put on the market by Chas. Churchill and Co., Ltd., and exhibited by them at the Electrical Exhibition. The left-hand view shows the new self-contained motor-driven plain grinder, and the other two are of a universal and chucking grinder. The former has a capacity up to 10 ins. by 60 ins., and has a quick hand traverse for moving the grinding wheel head.

The ratchet wheel feed is arranged to give a feed at each reversal of the table from .0025 in. to .004 in. in diameter of work, and gearing is introduced below this with an auxiliary shaft to the hand wheel, so arranged as to give a ratio of 8 to 1 between the speed obtained by the fine hand motion and that obtained by the quick hand motion. At the same time this does not interfere with the position of the automatic trip, the fine wheel feed tripping automatically at the same place as it was set before using the quick hand motion. An indicator is fitted which shows the position of the wheel head on its slide, so that the operator can bring the wheel from, say, a large diameter down to a smaller one, without even looking at the position of his wheel, the indicator on the quick hand motion showing the position of the wheel.

The machine is entirely self-contained and requires no overhead countershaft or long arm drum, and is driven by a motor placed at the back of the machine. To drive the grinding wheel the belt from the motor pulley is taken up to a self-contained countershaft carried in a bracket on the wheel head, tension being maintained on the belt and allowance for the travel of the wheel head by jockey pulley, while the countershaft runs on ball journal bearings.

The second machine has a capacity of 12 ins. by 36 ins., and may be compared to the ordinary plain or universal grinding machines as the turret lathe compares with the ordinary engine lathe. It is designed to carry three wheels, two of which may be heavy wheels for external work, and the third for internal work. The two heavy wheels are carried on spindles built integrally with the turret, and, for internal work, any spindle may be immediately fitted to suit heavy or light work, all internal spindles being self-contained and arranged to fit the bracket mounted on the turret.

Primarily, the two main spindles are intended to carry a face wheel for the grinding of shafts or other pieces of work externally, and, secondly, a cup wheel for the side grinding of flanges or similar work. On work where the cup wheel is not required it may be removed and a face wheel substituted the same size as that carried on the first spindle, thus enabling the use of roughing and finishing wheels on work where high finish is required, instead of using a wheel which may not be the best for either purpose. On the other hand, wheels of different grade may be carried for the grinding of different metals, obviating the changing of wheels and consequent loss of time and waste of wheel in returning.

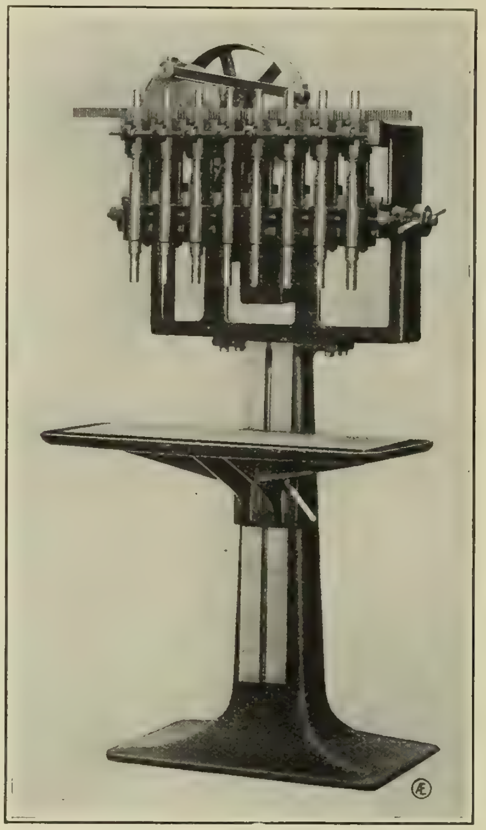
On chuck work, the tailstock can be removed and the machine operated purely as a chucking machine. The work headstock, which is very heavy, and carries a hardened and ground spindle, has a swivel base and is graduated in degrees. Various attachments are supplied, the equipment including a large face-plate, four-jaw chuck and face chuck with draw-back collet.

CONSTRUCTIONAL WORK.—In order to facilitate standard designs in shop girder work, Messrs. David Rowell and Co. have issued a booklet containing all their various types of girder.

A MULTIPLE SPINDLE VALVE GRINDING TOOL.

The subject of the illustration below is an ingenious machine made by the Foote-Burt Company, for the grinding in of valves in four-cylinder engines where the valves are all on the same side.

The eight spindles are adjustable as to position, and are driven by a reciprocating rack, which causes them to make a turn and a quarter in alternate directions, while at every fifteenth stroke the valves are lifted from their seatings during a period corresponding to two revolutions. Thus the process of grinding by hand is reproduced entirely automatically, and quite unskilled labour is sufficient to take charge of the machine. To change a set of cylinders is not a long operation, so the capacity of the tool is very considerable. The following are the principal details and dimensions :—



Automatic valve-grinding tool for four-cylinder monobloc engine.

Distance face of column to centre of spindle	$8\frac{1}{2}$ in.
Morse taper in nose of spindles	No. 1.
Minimum distance nose of spindle to top of table	6 in.
Maximum distance nose of spindle to top of table ...	30 in.
Maximum travel of spindles ...	$2\frac{1}{4}$ in.
Minimum centre to centre of spindles	2 in.
Maximum centre to centre of spindles	$3\frac{1}{2}$ in.
Working surface of table ...	12 in. x 30 in.
Spindles withdraw every ...	15 revolutions.
Countershaft speed	150 r.p.m.
Spindle speed	375 r.p.m.

CATALOGUES RECEIVED.

MERCHANT AND EVANS have sent us a list of the various articles manufactured and supplied by the firm. These include universal joints, spring fitted greasers, Hele-Shaw clutches and various standard parts mostly suitable for the automobile trade in America.

ELECTRIC LIGHTING SYSTEMS. — A further system of dynamo car lighting has been put on the market by the Apple Electric Co., and the description is embodied in a pamphlet containing illustrations of the dynamo fitted in many ways to various cars, and also a general description of the system under which the dynamo works, and a brief history of the manufacture of the particular type of lighting employed by the company.

GOVERNORS.—Wilson Hartnell and Co. make a speciality of the manufacture of governor gearing suitable for engines running electric lighting, or for plant in connection with an automobile factory. Iti s claimed that the governor in question has been extensively used on turbines, and contains some data which will help in the decision as to which size of cover is most suitable for any particular job.

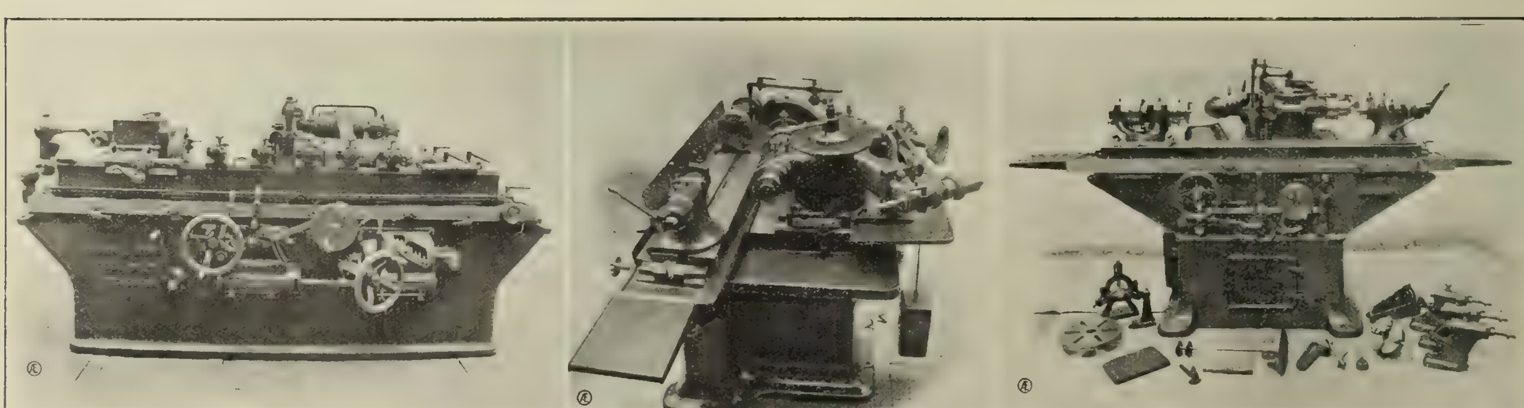
DROP FORGINGS.—In order to assist customers, the Anderson Forging and Machine Co have issued a complete specification of the standard forgings for which they possess dies. In the main this contains tie-rod ends and knuckle joints, of which every kind is illustrated, together with universal joint jaws, crankshafts decision as to which size of governor is most suit-automobile factory.

FACTORY PLANS.—Those firms which require a certain number of small-sized stationary engines will be interested in the catalogue of the Hillman Motor Co., which gives particulars of every engine suitable for this type of work, together with the conditions of business, guarantees and full list of accessories which might be necessary for any one particular engine.

LATHES.—Further to their recent list of milling machines a well illustrated catalogue has been brought out by James Archdale Co., Ledzan Street, Birmingham, giving full particulars of all the capstan engine and screw cutting lathes manufactured by them, which is illustrated, together with its accessories and a complete description, which may facilitate the choosing of any machine likely to be of use for some particular job. At the end is a list of steadies, tools and chucks which can be used in connection with their machines. The whole catalogue is extremely well got up.

SHOP EQUIPMENT.—Greenwood and Batley are the manufacturers of almost every kind of shop equipment and shop machines necessary for factory equipment. A pamphlet has recently been issued by them setting forth the advantages of their cold rivetting machines, with some examples of their work, and also a full list of the automatic crank shaping machines supplied by the same firm.

IGNITION.—Although accumulator ignition is not usually fitted to modern machines, it is interesting to glance through the list of the Dayton Engineering Laboratory, which gives particulars of every type of distributor, wiring and contact maker, together with the particular timings necessary with their plant, and some account of the up-keep, with instructions when repairing any minor defects which may crop up during ordinary running.



A new self-contained motor-driven plain grinder (left-hand view) and a new universal chucking grinder (two right hand views).

THE

AUTOMOBILE ENGINEER.

A technical magazine devoted to the theory and practice of automobile construction.

VOL. I. No. 19. DECEMBER, 1911. PRICE SIXPENCE.

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CONTRIBUTIONS.

Articles of a technical nature relating to the design or construction of automobiles for land, air, or water, will be carefully considered by the editor. Matter must be clearly written or typed on one side of the paper only, and a stamped addressed envelope must be enclosed for return. No responsibility can be accepted for the safety of contributions although every reasonable care will be taken.

Correspondence on interesting subjects, which come within the scope of this journal is cordially invited. All letters must contain the name and address of the writer, but not necessarily for publication. If a reply is desired by post enquiries must be accompanied by a stamped addressed envelope.

CONTENTS.

THE OLYMPIA SHOW.

In this issue no direct reference to design as shown at Olympia last month will be found, but the whole subject will be dealt with fully in the "Automobile Engineer Year Book," which will be published in about four weeks' time. In making this second extra annual issue we have been guided by the requests of many of our readers that the information should be more condensed, and in tabular form wherever possible. We trust that the 1912 "Year Book" will be a real improvement on the last, but shall welcome any suggestions for its further expansion, more especially as regards the improvement of the tables of data. In compiling these care has been taken to include those from the ordinary engineers' pocket book which are required most frequently, and to add a variety of other tables having special application to automobile work which have not previously been published.

There are, of course, numerous details of design concerning which the data available is insufficient to enable any formula or table to be drawn up, and it is intended to add these year by year as rapidly as possible.

THE OPEN MOUTH.

With this issue, No. 19 of THE AUTOMOBILE ENGINEER we end our first volume, and in future shall commence a fresh one with each January issue. In the first issue we set forth our aims and aspirations, chief of which was the hope that the new journal might prove to be of real value to designers and constructors of automobile vehicles. We believe we have been able in some measure to carry out our mission, but we would like to take this opportunity of thanking all those firms and individuals who have so freely placed information at our disposal for circulation amongst their colleagues.

In describing chassis designs we have criticised freely where it seemed that criticisms were deserved, and just as we have always endeavoured to make such comment of a constructive nature, so has it been accepted cheerfully—or even welcomed. It is often said that no secret can be kept that is worth keeping, and it is a curious commentary on this dictum, that the few manufacturing firms who have refused to give us information concerning their practice are mostly of small account.

In this connection the remarks of Mr. H. E. Coffin, of the American Society, in the paper which he read before the I.A.E. last month—reported on another page in this issue—acquire a special interest, because the opposition to free circulation of knowledge was much greater in America a few years ago than it is in this country to-day. In the course of the discussion which followed the reading of the paper Mr. Coffin laid stress on this fact, saying that the present spirit of co-operation had been brought about only by much time and labour. When the first efforts were made by some enlightened engineers to get the others of their craft together to discuss things generally, every maker in America almost "was proceeding behind locked doors, whereas now there is probably not a single factory in the United States where visitors are not welcomed."

Fortunately we can make an almost equal claim for England, and signs are not wanting that even the conservative industries of France and Germany are waking to the fact that progress in ideas is being made without. Commercially the refusal of the French trade to continue to hold the annual show has proved disastrous. The closed mouth and the locked door represent an equal stagnation of intellect. The growth of our own Institution of Automobile Engineers and the American Society is a sure and certain sign—if, indeed, such were needed—that it really is to the advantage of everyone to discuss freely with competitors his troubles and his discoveries. This means that there is a centralisation of research work going on equal to that of academic scientific knowledge, and that no slackening in the rate of improvement in automobile construction is to be expected for some time to come.

At the close of the year it is instructive, perhaps, to speculate in what direction development may most confidently be anticipated. Twelve months ago tendencies in design were much more easy to define than they are now, because, whilst last year the high speed, long stroke engine was a comparatively newcomer, and one might safely prophesy the ultimate supremacy of this type, it is to be doubted whether the wise limits of bore and stroke have not now been passed in many instances. One thing, however, is certainly general, and equally certainly the effect of public discussion, and this is the much smoother running of the long stroke engine which has been brought about by lightening the reciprocating parts, increasing the rigidity of shafts and bearings, and by rotational balancing. Where improvements seem most likely to be made during 1912 are in carburation and the further elimination of vibration from engines—together, perhaps, with continued development of new valve mechanisms. The greatest changes are now needed, and are to be hoped for, in variable speed transmission, for some thing is needed badly which will (by giving equal silence of operation and great ease in changing) discourage the present tendency on the part of the ordinary driver to use the direct drive to the uttermost limit of low engine speed. The chain drive "gear" box is, of course, an attempt in this direction; whether it is the final solution it remains for time to show.

AMERICAN COMPETITION.

Being the last of a series of articles by a member of "The Automobile Engineer" staff, who recently made an extended tour in the motor manufacturing districts of the United States.

In concluding my article on American manufacturing methods in the last issue of "The Automobile Engineer" I undertook to make some pronouncement on the subject of competition between this country and America, more especially with regard to the British over-seas dominions and possessions. But, first of all, as regards this country, I am tempted to agree neither with those who maintain that the coming of the American car is likely to prove infinitely harmful to the Home industry, nor with those who, with equal obstinacy, profess to believe that it will do no harm at all. Firstly, to assume that the cheap American cars (and it is the very cheap ones which are at present being landed in England in the greatest quantities) will prove incapable of standing up to European usage for three or four years, or more, is almost absurd. The cheap European cars mostly have very small engines and are quite as ill-cared for in detail as any of the American productions, which have, on the other hand, large slow speed engines and are usually somewhat better proportioned in their transmissions. Speaking generally, the cheap American car is likely to give equally good service as similarly cheap European vehicles. Lately one or two of the middle class American manufacturers have commenced to export into England and they are sending over cars which, although old-fashioned in many respects, are quite capable of running well over many years. Against the American, however, is that fact that his cars invariably lack the refinements to be found on European machines of slightly higher price. Absence of ball bearings in the steering gear, stiff control levers, somewhat uncomfortable body work, poor paint and poor accessibility are the faults of the American car which would first impress themselves on a European user. I think it is extremely unlikely that a man who commenced his motoring with an American car would ever buy a second, at all events for use in Europe. He would find that when the time arrived to acquire a new car the extra convenience and extra comfort to be obtained by spending £50 to £100 more in the local market would be well worth while. Of course, American design is rapidly catching up with European development, but the big out-put in the United States makes it impossible that it should ever quite catch up. At the present moment many American manufacturers have settled definitely the principal features of their 1913 models—models which will not come into the market until the late summer of 1912.

If the so-called American invasion is threatening any real danger to the British motor industry, the way to combat it is not by the scamping of quality or the wholesale reduction of price. The competition is far more a matter of salesmanship than of engineering skill. Many of the small cars which have been prepared to oppose American importations are far more likely to destroy the reputations of their manufacturers, and ultimately seriously to injure their businesses, than they are to obtain a footing amongst that particular class of customer which the importer is seeking. Mr. H. E. Coffin, in the paper which he so recently read before the Institution of Automobile Engineers (and which is reported elsewhere in this issue), maintained that the cheap car had proved of the greatest benefit to American makers of better class vehicles, because it enabled thousands of men to become motorists who could not otherwise have done so, and that these same men subsequently became buyers of dearer and better cars. The view is held very generally in America and, up to a point, it is quite sound. Where it fails is when one comes to the makers of really good quality single and two-cylinder cars. I believe I am correct in saying that in America there is no such thing as a good car of this type, and it is a matter for argument whether a really high grade car with a single or two-cylinder engine is actually better than a cheap four-cylinder. It depends very much upon the work for which the car is needed, but everything points towards the early extinction of everything but four-cylinder or six-cylinder types. Really the British maker of single or two-cylinder cars is being hit quite as hard by his own 15 h.p. or 12 h.p. model as he is by any outside competition.

Supposing, however, that the British manufacturer was really anxious to obtain a footing in the cheap car trade which has begun in America, I am convinced that the American way of producing the cheap car is better than the European. In every case the engine and transmission of the American cheap car are rather heavy and large, and it is endeavoured to keep the engine so inefficient that it can never overload either itself or any portions of the vehicle. This produces a very smooth-running car and does not call for either the same quality of material or the same high accuracy of workmanship as the little car which is capable of the same speed, and is all the time made to work at its maximum efficiency in order to obtain it. I can see no reason why with our cheap labour and cheap material it should not be possible to make a car with a four-cylinder 4 in. engine, three-speed sliding transmission and a four-seated body, complete with everything for £250. There would have to be no finish, in fact the car would have to be no better in any way than the American prototype, but it could easily be just as good and just as profitable to handle.

Personally, however, I think that the lack of development of the Colonial market is far more serious than any loss of trade likely to be caused by American competition in England. The motor car trade of the Empire has gone to America in the main through the greater commercial aptitude of the American and through the, in some respects, greater suitability of the American car for rough country—it is the good American cars that go to our colonies. It has so often been repeated that high clearance, standard track, great axle strength, exceptional strength of steering connections, great cooling capacity, and great flexibility of springs are essential features for a successful Colonial car that it is hard to believe any of my readers are not already aware of these facts. On the other hand, never yet have I seen a Colonial model from a British factory which will fulfil these requirements in all respects, though one or two have come very near to it. I believe that few British makers or designers realise what a car actually has to do, day in and day out, once it is away from Western Europe, but I also believe that any good designer is perfectly capable of devolving a Colonial car if he will *really* consider the few points mentioned just above. After that the matter becomes a commercial one solely. Canada, probably, is the most difficult market for any British goods, on account of her proximity to the United States. In South Africa and in Australia and in New Zealand I am certain that British cars would find a ready welcome, if they came equally well equipped for service as their American rivals. When a man is offered two articles, one of foreign and one of Home manufacture, if they are equally serviceable, his patriotism may even go so far as to encourage him to pay a slightly higher price for the Home goods; he will never, however, content himself to put up with less convenience and, whereas we here in this country have everything our own way on the score of comfort and luxury, the best body and the sweetest control in the world are useless if the bottom of the crankcase is a foot deep in mud, and the necessities are, in Colonial work, even greater conveniences than the luxuries.

MECHANISMS FOR BRAKE ADJUSTMENT AND BRAKE LEVER SETTING.

One of the most noticeable features of the Show previous to the one which has just concluded was attention to detail which showed itself best in the form of neat and rapid adjustment for wear on the brake shoes. This year further improvements have been apparent in the details of the chassis, and once more we find that great attention has been given to the brake details as a whole and that certain improvements have now become almost universal. For instance, a great number of cars this year are not only provided with rapid and accurate means for adjusting wear on the shoes, but they also have various devices which allow adjustment to take place in every part of the brake actuating mechanism itself. Not only can the ten-

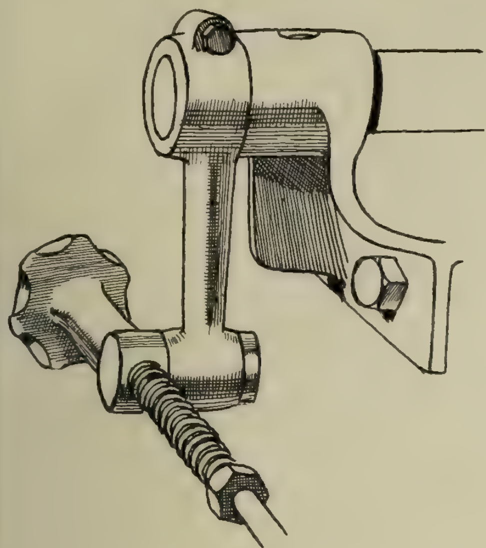


Fig. I.

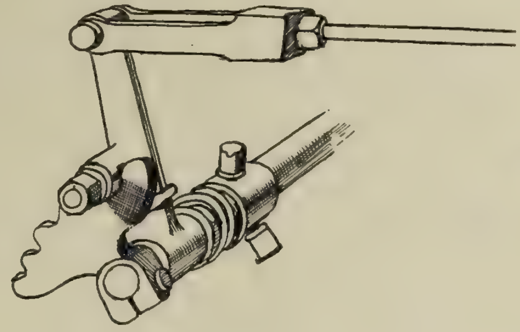


Fig. II.

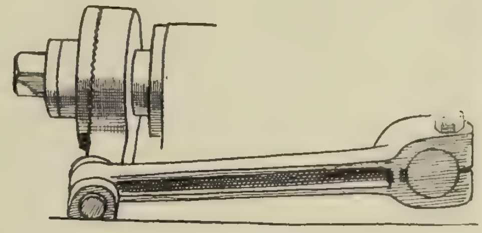


Fig. III.

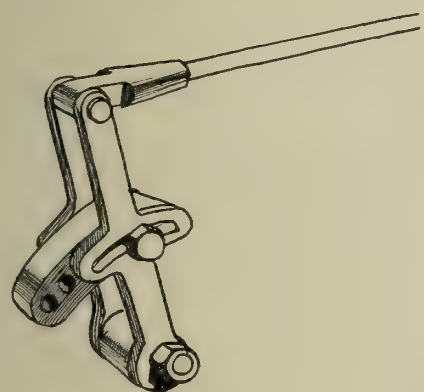


Fig. IV.

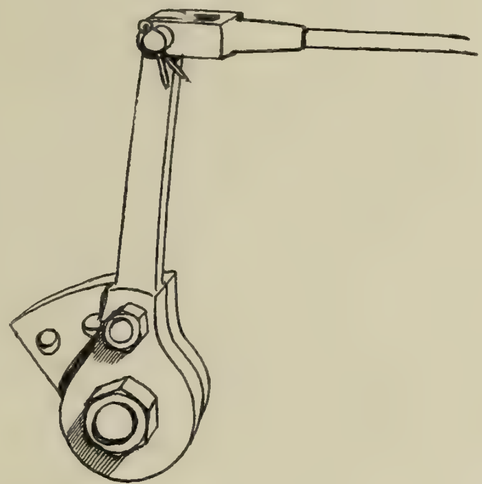


Fig. V.

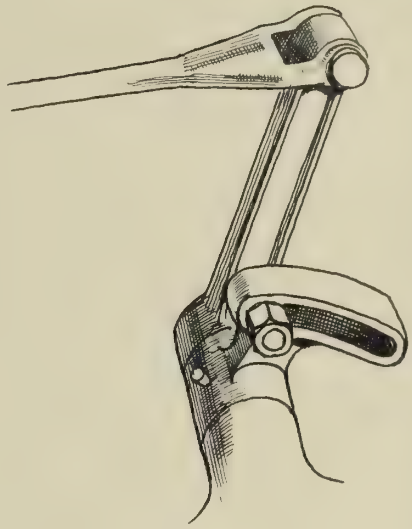


Fig. VI.

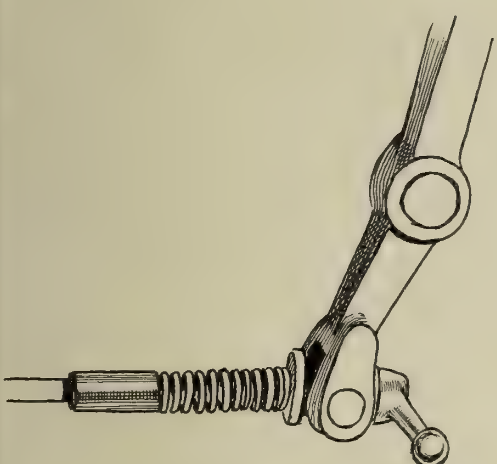


Fig. VII.

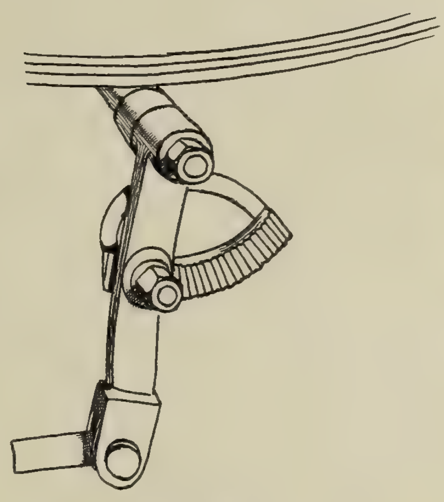


Fig. VIII.

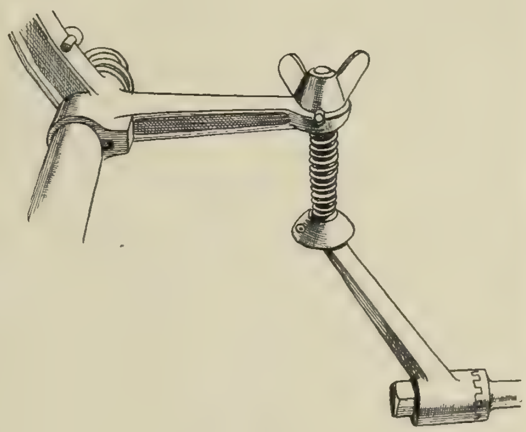


Fig. IX.

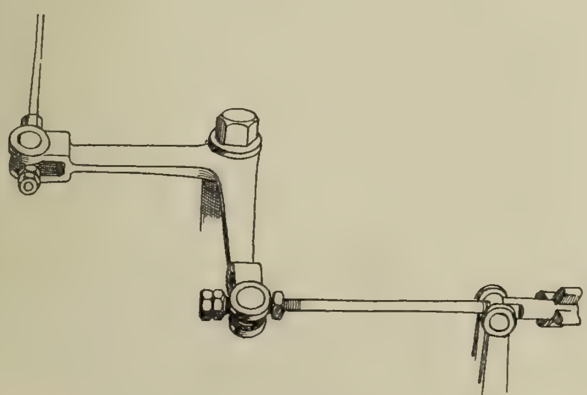


Fig. X.

DIFFERENT METHODS OF ADJUSTING AND SETTING HAND LEVER AND PEDAL ACTUATED BRAKES

sion rods be adjusted at both ends and locked when so adjusted, but the position of the brake pedal can now be set independent of the amount of wear which has taken place in the brake shoes. With cars of some three years back the adjustment of the footbrake for wear might in itself be excellent, but in almost every case when this adjustment had been used the brake pedal took a position in which its maximum leverage could not be applied, and generally was not so accessible or so easy to operate as at the time when the brake was new. Designers have now realised the extreme importance of brake gear adjustment in every direction, and consequently it is now possible to set the pedal to the position of its maximum leverage whatever the adjustment of the brake shoes may be. Again, in the old car the tension rods being threaded at either end, could be adjusted to a certain degree by removing the jaw pin and turning the brake rod, locking it when in position by the lock-nut provided on the thread. In rare cases a nut had been brazed on the centre of the tension rod to enable a spanner to be used in adjustment, and the threads were right and left hand, so that it was unnecessary to uncouple the jaw. But in the main the tension rods themselves were not adjustable, although the actual adjustment of the footbrake shoes was frequently placed upon them. Designers cannot be urged too much to give their best attention to details such as the above. The average private owner will be anxious to adjust a brake which is provided with some ingenious form of fitting making adjustment at once easy, accessible and cleanly, and he is far more likely to give the vital and necessary attention to his brakes if his adjustment is of the sort that he can show to his friends with a certain degree of pride.

The average driver is probably as bad as regards attention to minor details unless some fitting is placed on the chassis which will readily draw his attention to the part requiring attention. In these days when the balance beam type of brake compensation is so much used, the same fine type of adjustment is not needed as was the case when each side brake had to be adjusted separately, and most accurately, in order to obtain any description of brake effect. In one particular chassis every joint on the whole car was provided with a quadruple lock-nutted tension rod, so that the whole could be adjusted from beginning to end. The chassis in question was one which had been evolved by a firm largely interested in taxi-cab designs, and it is probable that the abnormal wear subsequent upon the savage usage of the ordinary London taxi-cab driver was responsible for such a complete and effective arrangement.

Fig. I. is a sketch of the brake adjustment on the Fafnir car and is an example of the neat, accessible manner in which such adjustments are invariably designed. The brake actuating arm is held to the brake rod by a split lug and bolt. At the lower end there is a swivel pin through which the brake rod is passed, and at the further side a small hand wheel can be turned against the action of the spring, observable on the tension rod, which will lock the hand wheel automatically after a certain amount of movement.

Fig. II. is an example of the brake pedal setting used in conjunction with adjustment, and is one fitted to the Delahaye chassis at Olympia. In this there is a large notched quadrant bolted on the brake actuating rod, whilst the lever takes a bearing on a long boss, quite free on the actuating rod, which passes through it. On the side of the lever there is a boss which is slotted to receive the notched quadrant, itself held to the boss by a nut and washer. In this design it is only necessary to slack back the nut and free the segment on its shaft, in order that it may be pulled outwards sufficiently to enable it to move forward to the next notch, at which point it can be pressed back and the whole bolted up again, altering the lever position relative to the brake pedal. A variation of this type, which can be handled in a quicker and cleaner manner, is the adjustment used for the footbrake rod on the Niclausse cars, Fig. III. In this case the car has an expanding brake with an actuating rod carried in the modern fashion, entirely inside the gearbox, and in order that the setting shall be as simple as possible, the surfaces of the disc on the actuating rod and that on the lever which operates it are provided with small teeth engaging with one another, thus locking the whole device as though solid when the nut is tightened. It will be seen that it is merely necessary to slack the nut, disengage the toothed disc and move it on as many teeth as are necessary to restore the pedal to its corret position.

On the Arrol-Johnson cars a totally different form of adjustment is provided; in this case for both brakes, which are expanding and on the rear wheels. Fig. IV. shows the general arrangement of the setting levers which consist of twin levers either side of a solid segment with a number of horizontal holes. In each of the twin levers there is a long slot and a hexagon bolt grips both of them to the central segment. Thus there is a wide variety of adjustment, rather more in fact than one would think necessary for the type of brake employed, consisting firstly of the adjustment due to movement in the slot, and secondly of the bolt when moved from one hole in the segment to any of the others seen on its surface.

Something of the same description has been applied to the Martini rear wheel brakes, Fig. V., in which the actuating lever is coupled to a segment having a number of holes drilled along the line coincident with the centre of a bolt hole in the lever itself. The segment is connected to the cam operating mechanism of the brake, while the arm, although taking a bearing on the operating rod, is free but for the bolt in the hole already mentioned. To set this type of lever it is necessary to withdraw the bolt entirely and to place it in one of the other holes. Consequently the adjustment is not as fine as that which can be obtained by the use of some of the other devices, while the necessary withdrawal of the bolt, which may be caked with mud, or rusted by water, may make it more troublesome than other adjustments.

In Fig. VI. another type is shown with better possibilities of adjustment. It consists simply of a long slot pinned to the operating rod, while the brake lever swings loose on the end of the same rod and is held to it by a nut and bolt working in the slot. In this case all that has to be done is to slack the nut and move the lever to any desired position, afterwards tightening up the nut, and there is not much possibility of its unpremeditated movement, although security depends on friction alone. The example in question was taken from the Overland car, the same arrangement being applied to every brake lever on the entire chassis. Another neat form of adjustment which has come into great prominence during the last year, is the adaptation of the wing nut with spring locking gear for the external adjustment of the hand brake. Fig. I. was an example of the standard form of foot brake adjustment, whilst Fig. VII. shows the method now used for adjusting the hand brake tension rod by rotating a hand nut with a face provided with "V" notches, locking to a suitable surface by a spring seen on the tension rod. This arrangement is in use on the Armstrong-Whitworth.

Fig. VIII. is the brake lever setting on the Peugeot, consisting of a toothed quadrant holding the brake actuating mechanism, which is gripped to the brake lever by tightening the nut seen on the latter. In this case quite a fine adjustment is obtained in quite a simple manner, and there is no possibility whatever of such a device slipping, however violently the brake may be applied. On the Metallurgique (Fig. IX.) the setting is of an entirely different type to those hitherto described, consisting of a multi-toothed dog clutch similar to the older type of direct drive, which can be brought out of mesh by slackening a nut on the brake lever boss and meshing again in a different position, when the nut is returned. It is probable that this device has more good points than any of the various devices already described, since it is at one time easy to adjust and absolutely rigid when it has been adjusted, while its contact area is considerable. The same sketch shows the brake shoe adjustment, consisting of a butterfly nut locked by a spring.

The sketch, Fig. X., is an excellent example of the way in which every tension rod on the brake gear can be made separately adjustable. It will be observed that each rod has four separate locking nuts, and that a notched form of locking gear is incorporated in the hand wheel by which the ordinary adjustment is effected. The joints themselves are of the roller type, through which the threaded rod passes, and the nuts at either end can move this rod through the joint according to the direction in which adjustment is required. Throughout the whole brake gear extreme care has been taken of adjustment, and it is extremely unlikely that a device of the above description should be incorporated in a chassis mainly used for taxi-cab work, unless it was free from the majority of ills to which a brake gear locking device is liable.

A comparison of the various designs herein described is extremely interesting, because it shows in a vivid manner the trouble which has been taken to eliminate any possible source of trouble and to render every adjustment on the chassis perfectly easy to handle. It is an open question whether so much adjustment can actually be required during running, but it is absolutely necessary that a setting device be used in connection with the brake pedal so that at all times an equal leverage can be obtained, and designers ought to provide this adjustment.

THEORY AND PRACTICE.

Some Notes on their Relative Values.

By Bertram C. Joy.

Though I have headed these few remarks "Theory and Practice," it would perhaps have been more correct to have reversed the order of the words and to have written "Practice and Theory," for it is in this latter order that these two usually occur. I do not think that ever before in the history of engineering has theory been so completely thrown to the winds as was the case in the early days of automobile design in this country. Either the designers of that time were unable to adapt the theory used in other branches of engineering to their own particular requirements or—and I believe this is a great deal nearer the truth—much of the theory, formulæ, etc., available were quite unsuited to design of such a totally different character as that of the motor car. To take just one example of the latter description. Suppose it were desired to discover the thickness of metal suitable for an engine cylinder, for instance, a table such as I have by me at the moment of writing might be referred to; this table has for its heading, "Thickness of Cast Iron Cylinders for pressures of 500 lbs. per square inch and under," and against the inside dimension of 4 ins. appears the requisite thickness, viz., ½ in. This dimension, as will be at once recognised, provides about twice as much metal as is actually necessary, and hence the table referred to would have been quite useless to the automobile designer. How the extremely small dimensions now in use in cases such as cylinder wall thickness, to mention just one, were arrived at is difficult to say, but I think that the courage of a few designers is chiefly responsible. In a work on the subject of motor design, and under the heading of carriage springs, appears the following remark:—"It is recommended that copious notes be taken, whenever possible, of springs in actual use, *and these, when tabulated, will be of more practical value than any formulæ*" These last words which I have italicised appear to constitute a very sound and practical piece of advice, and are applicable, I am confident, to a great number of parts other than the axle springs.

There are very few designers, I think, who do not keep always at their elbow a book in which are entered all sorts and conditions of notes referring to matters of interest which may from time to time pass through their hands in the drawing office. Such notes and particulars as these when present in sufficient quantity may be made to form the basis of a formula, or perhaps better, they may be set down graphically for future reference. Thus, out of many odd scraps of information, eventually is evolved a highly useful and probably quite thoroughly reliable diagram or table.

By the foregoing remarks I merely intend to suggest that a formula with a foundation of results with which one has personal acquaintance is generally more to be depended upon and provokes a greater degree of confidence than does one the origin of which is but imperfectly known. I came across a somewhat forcible example of a formula of the latter type a few years back. It referred to gudgeon pin diameters of no less than 2½ inches for the gudgeon pins of a four-cylinder 25 b.h.p. engine. In this case the formula was so immensely and obviously incorrect that it could scarcely be misleading.

There have been a few instances, doubtless, in which practical inventions have resulted from what almost appeared to be the dreams of theorists. The Diesel engine, I understand, came about in this way; thus the most theoretically perfect working cycle was laid down on paper and a practical engine was evolved—without, I believe, excessive experiment. Theory assuredly led the way here. I can call to mind other cases, however, in which practical results have led to a discovery of correct theories.

Whether any particular method of effecting an object is theoretically correct or otherwise, does not necessarily affect its practicability. The fact that a device has been in use many times stamps it beyond doubt as a *practical* one, and if it were deemed essential that every part of a modern machine must be theoretically perfect to be included, a great number of machines that are in existence to-day would not be possible. Indeed, to sum up the question, I think one may state a formula to the effect that *any piece of design need be as much in accord with correct theory as practical requirements permit, and no more.*

THE SOCIETY OF AUTOMOBILE ENGINEERS.

A brief account of their first visit to Europe and of their first joint meeting with the I.A.E.

In our last issue we announced the fact that a considerable party of members of the Society of Automobile Engineers (of America) were about to visit this country and France, and we gave then a provisional programme of arrangements. The number who actually landed on November 6th totalled 43, and one or two other members joined the visitors subsequently. It is not proposed to give any detailed accounts of the visits to the various works which were undertaken, although it may be mentioned that these included the London General Omnibus Co., W. and G. De Cros, Limited, The Humber Motor Co., Ltd., Daimler Motor Co., Ltd., The Wolseley Motor Co., Ltd., Rudge-Whitworth, Ltd., The Coventry Chain Co., Sunbeam Motor Co., Ltd., Hans Renold, Ltd., Crossley Bros., Ltd., and the Crossley Motor Co., Ltd., in this country, and the works of Messrs. Panhard and Levassor, Lemoine and Delage, in France, together with the aeroplane factory belonging to M. Robert Esnault, Pelterie. Small parties also visited several other French works individually, and the official programme concluded by a banquet, which was given by the Michlin Tyre Co., at the Automobile Club de France on the evening of Wednesday, November 22nd. The visitors also had a joint meeting with the I.A.E., which is reported on page 558, and the Institution entertained them to dinner on the following Saturday evening.

Everywhere the visitors were welcomed accordingly and almost everywhere they were entertained, while taken as a body they, undoubtedly, were surprised greatly at the magnitude and extremely flourishing condition of the British industry. Several of the American visitors had made many trips to Europe and were well acquainted with the situation here, but the majority had never before crossed the Atlantic, and their expressions concerning motoring and the motor industry in England and France are distinctly interesting. Firstly, they were almost unanimous in expressing surprise at the magnitude of the British industry, the high average quality of the design and the extreme accuracy of workmanship, which appeared to be usual.

From the point of view of organization and equipment, several stated very definitely that such works as the Daimler and Wolseley were equal to anything that could be shown in America. On the other hand, as was to be anticipated, we have been criticised owing to the amount of hand work which still goes to the construction of a chassis, but we believe none the less, that the careful hand fitting of every part is in a large measure responsible for the splendid service which the modern car can give. However, no useful purpose is to be served by going into small detail concerning matters which are more or less general in application.

This European excursion of the American Society is to be regarded as extremely important, because it marks a definite point in the development of automobile engineering as an important profession. The meeting, which took place at Storey's Gate on November 8th, 1911, cannot fail to have marked the commencement of a period of closer communion between the two nationalities of automobile builders; the two who stand together for quality and quantity against the whole world.

There is no doubt that many of the visitors will go back with much fresh knowledge which will be of use to them, but it is equally sure that they have left behind them with us an at least equal amount of information. It is to be hoped that the invitation extended to the I.A.E. to pay a return visit to the United States will be accepted at no distant date by a representative body of men, for there can be no doubt in the mind of a man who has made such a trip as to its practical business value.

PRESSURE AND TEMPERATURE IN THE INTERNAL COMBUSTION ENGINE.

By James Langmuir Napier.

SOME apology seems desirable in venturing upon a subject which may be considered to be already threadbare, and the excuse offered is that this is an attempt to diminish to some extent the gap existing between theory and practice in the matter of cylinder pressures and temperatures, and particularly to show the result of extending the basis of theoretical reasoning so as to conform more nearly with practical conditions.

The nature of the subject renders the employment of general symbols unavoidable, but definite numerical computations are given throughout, and readers who find mathematics indigestible may skip. Much old ground has been covered, of which the treatment is partly new, and I am not certain that the newness is an improvement. Whatever apparent complication has been added in this way has been due to my desire to express temperatures always in terms of an explosive mixture of definite heating value.

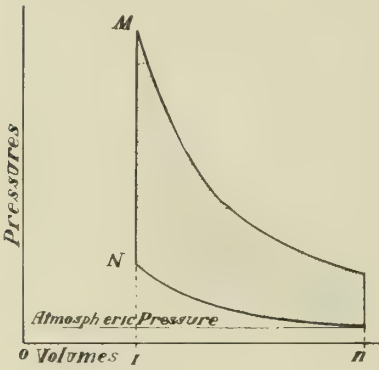


Fig. I.

Fig. I. represents a theoretical pressure diagram for an internal combustion engine, acting on the ordinary Otto cycle. The compression volume is 1, and the total volume is n. The number of expansions is therefore n. The expansion curve is such that $P=MV^{-1\cdot4}$, where P is the pressure at any volume V. Similarly the equation to the compression curve is $P=NV^{-1\cdot4}$. The explosion pressure is M, the compression pressure is N, and the atmospheric pressure is p, all measured from absolute zero. The compression curve cuts the atmospheric line at volume n, and the value of N is therefore $pn^{1.4}$. The value of M depends on the heating quality of the explosive mixture. R is the ratio between M and N.

We have, then, the following abbreviations :—

n=Number of expansions (or compression ratio).

$R=\frac{M}{N}$,

p=Atmospheric or initial pressure=14.7.

pc=Compression pressure.

pe=Explosion pressure.

pt=Terminal pressure.

The net pressure shown by the shaded portion of the diagram, Fig. I., at any volume V is $(M-N)\ V^{-1.4}$. Integrating between the limits $V=1$ and $V=n$ we find :

$$\text{Area} = 2\cdot5\ (M-N)\left(1-\frac{1}{n^{\cdot4}}\right)$$

Since $R=\frac{M}{N}$, $(M-N)=N\ (R-1)=pn^{1\cdot4}\ (R-1)$ and we may therefore write

$$\text{Area} = 2\cdot5\ (R-1)\ pn^{1\cdot4}\left(1-\frac{1}{n^{\cdot4}}\right),\ \text{and}$$

$$\text{Mean pressure} = \frac{2.5\ (R-1)\ pn^{1\cdot4}\left(1-\frac{1}{n^{\cdot4}}\right)}{n-1}$$

$$=2\cdot5\ p\ (R-1)\frac{n^{1\cdot4}-n}{n-1} \quad \ldots \quad \ldots \quad (1)$$

The theoretical efficiency is $\frac{\text{Area between 1 and } n}{\text{Area between 1 and infinity.}}$

$$\text{which is } \frac{2.5\ (M-N)\left(1-\frac{1}{n^{\cdot4}}\right)}{2\cdot5\ (M-N)}$$

$$=1-\frac{1}{n^{\cdot4}} \quad \ldots \quad \ldots \quad \ldots \quad (2)$$

Numerical values of this expression are :—

For 2 expansions	efficiency	= 24.22	per cent.
,, 3	,,	= 35.56	,,
,, 4	,,	= 42.57	,,
,, 5	,,	= 47.47	,,
,, 6	,,	= 51.17	,,
,, 7	,,	= 54.08	,,
,, 8	,,	= 56.47	,,
,, 9	,,	= 58.47	,,
,, 10	,,	= 60.19	,,

It may be remarked that this formula for efficiency excludes all considerations except the number of expansions. Whatever may be the temperature of the atmosphere or the temperature added to the mixture by explosion, makes no difference to the theoretical efficiency.

Assuming that the piston is at the beginning of the induction stroke, and that the compression space is filled with air at atmospheric temperature and pressure, I use the following abbreviations for the successive temperatures reached, all being expressed in absolute centrigrade units :—

T =Temperature of atmosphere=290° C. absolute.
T_c =Temperature at end of first compression stroke.
T_e =Temperature after first explosion.
T_t =Temperature at end of first expansion stroke.
QT =Initial temperature during continuous running.
QT_c =Temperature of compression during continuous running.
QT_e =Temperature of explosion during continuous running.
QT_t =Terminal temperature during continuous running.
T_a =Temperature added to explosion mixture on combustion at constant volume, assuming no dilution by previous products of combustion (or, in other words, the theoretical heating capacity of the mixture).

When a volume of gas v_1, at absolute temperature t_1, and absolute pressure p_1, changes these quantities to v_2, t_2, p_2 the result is expressed by the equations :—

$$\frac{v_1}{v_2}=\frac{p_2 t_1}{p_1 t_2}\ ;\ \frac{p_1}{p_2}=\frac{t_1 v_2}{t_2 v_1};\ \frac{t_1}{t_2}=\frac{v_1 p_1}{v_2 p_2}$$

From these equations and from the assumed relations we obtain by simple algebraical processes a number of results connecting the various quantities, which are here summarised :—

$$p_c = N = pn^{1\cdot4}$$
$$p_e = M = Rpn^{1\cdot4}$$
$$p_t = Rn$$
$$R = \left(1+\frac{T_a\ (n-1)}{T\ n^{1\cdot4}}\right)$$
$$T_c = n^{\cdot4}T$$
$$T_e = \left(n^{\cdot4}\ T+\frac{T_a\ (n-1)}{n}\right) = R\ n^{\cdot4}\ T$$
$$T_t = RT$$

R has a maximum value when $\frac{dR}{dn}=O$, that is when $\frac{d}{dn}\left(\frac{n-1}{n^{1\cdot4}}\right)$ $=O$. This occurs when $\frac{n-1}{n^{1\cdot4}}=\frac{1}{1\cdot4\ n^{\cdot4}}$, and therefore when $n=3.5$. This result is independent of the value of T_a; that is, it is independent of the heating quality of the explosive mixture. Whatever value be given to Ta, R is unity when n is unity and when n is infinite.

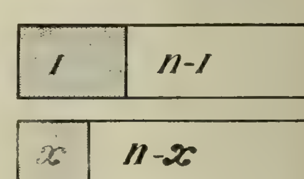


Fig. II.

So far I have assumed the compression space at the beginning of the induction stroke to be filled with air at the temperature and pressure of the atmosphere. This condition is altered after the first explosion, and at the end of the first exhaust stroke the compression space (1, Fig. 2) is full of gas at atmospheric pressure and temperature RT.

A new charge of explosive mixture at temperature T, and of a volume which perhaps may not be precisely obvious, is then drawn in. Neglecting slight differences in the specific heats

of the gases involved, and assuming that transfer of heat takes place without mixture, the state of affairs at the end of the induction stroke would now be represented by the lower diagram, Fig. II.

The exhaust gases have lost temperature and shrunk to volume x. The charge has gained temperature, and now fills the volume $(n-x)$. The temperature of the whole mass is greater than T, and after a sufficient number of explosions will settle down at some value QT. The corresponding terminal temperature will then be RQT.

Since the masses x and $(n-x)$ are now at the same temperature and pressure, their magnitudes are proportional to the masses of the residual products of combustion and of the fresh charge introduced. Therefore the temperature of the mixture will be :—

$$\frac{x\,QRT + (n-x)\,T}{n} = QT \quad \ldots \quad \ldots \quad (3)$$

$$\text{but } \frac{x}{1} = \frac{QT}{QRT} = \frac{1}{R} \quad \ldots \quad \ldots \quad (4)$$

Substituting this value of x for the co-efficient of QRT in equation (3), we have :—

$$\frac{QT + (n-x)\,T}{n} = QT$$

$$(n-x)\,T = QT\,(n-1)$$

$$\frac{n-x}{n-1} = \frac{QT}{T} \quad \ldots \quad \ldots \quad \ldots \quad (5)$$

That is to say that the amount of explosive mixture introduced is unaltered under the new conditions of temperature, and remains $(n-1)$ volumes at atmospheric temperature and pressure.

Since from equations (5) and (4) :—

$$\frac{n - \frac{1}{R}}{n-1} = Q,$$

$$\left.\begin{aligned} Q &= \frac{Rn-1}{R\,(n-1)} \\ \text{and } QR &= \frac{Rn-1}{n-1} \end{aligned}\right\} \quad \ldots \quad \ldots \quad \ldots \quad (6)$$

The efficiency stated in terms of temperature is :—

$$\frac{\text{Temperature added—Temperature rejected.}}{\text{Temperature added.}}$$

Q cancels out in this formula, which becomes :—

$$\frac{(T_e - T_c) - (T_t - T)}{T_e - T_c}$$

$$= \frac{n^{\cdot 4}T\,(R-1) - T\,(R-1)}{n^{\cdot 4}T\,(R-1)}$$

$$= \frac{n^{\cdot 4} - 1}{n^{\cdot 4}} = 1 - \frac{1}{n^{\cdot 4}}$$

the amount previously arrived at.

We have therefore during continuous running :—

$$\text{Initial temperature} = QT = \frac{T\,(Rn-1)}{R\,(n-1)}$$

$$\text{Temperature of compression} = QT_c = \frac{n^{\cdot 4}T\,(Rn-1)}{R\,(n-1)}$$

$$\text{Temperature of explosion} = QT_e = \frac{n^{\cdot 4}\,T\,(R\,n-1)}{n-1}$$

$$\text{Terminal temperature} = QT_t = \frac{T\,(Rn-1)}{n-1}$$

The temperature of compression has a minimum value when $\frac{d\,Q\,T_c}{dn} = O$. The expression for QT_c given above includes R, which may be written as an explicit function of n, and the complete differentiation of the expression would yield a very elaborate and unconvincing result. It is better to treat R as a constant, and on this condition we find that the temperature of compression has a minimum value when

$$n = \frac{1{\cdot}4\,R - {\cdot}6 \pm \sqrt{(1{\cdot}4\,R - {\cdot}6)^2 - {\cdot}64\,R}}{{\cdot}8\,R} \quad \ldots \quad (7)$$

For any value of R this equation gives two values of n, one of which, being less than unity, need not be considered. The other value of n has two obvious limits, viz. :—

When R is unity, n is unity.

When R is infinite, n is 3.5,

which is only another way of putting the results already arrived at with regard to the maximum value of R.

Calculating n from equation (7), it will be found that at ordinary values of R the value of n, giving minimum temperature of compression, approaches its higher limit. Thus :—

When $R=3$ min. temp. of comp. occurs at $n=2.82$
,, $R=4$,, ,, ,, ,, $n=3.04$
,, $R=5$,, ,, ,, ,, $n=3.14$
,, $R=6$,, ,, ,, ,, $n=3.20$

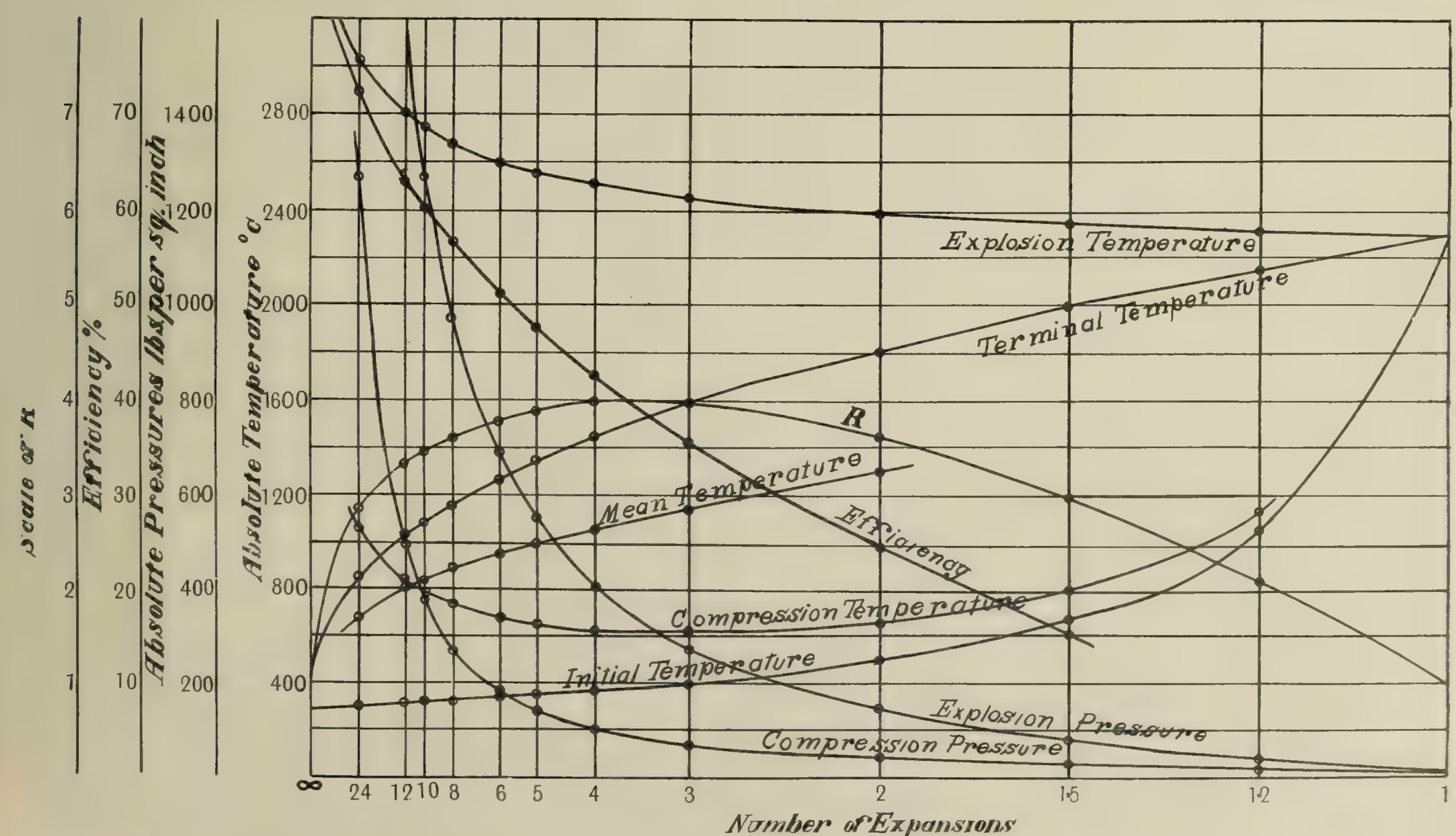


Fig. III.

The temperature of explosion increases as T_a does, and by the same amount, for

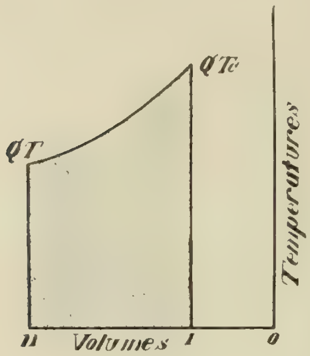


Fig. IV.

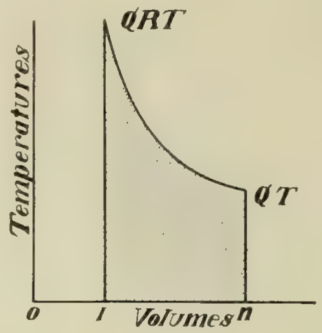


Fig. V.

$$QT_e = \frac{n^{\cdot 4}\, T\ (Rn -)}{n - 1}$$

$$= \frac{n^{1\cdot 4}\, T \left(1 + \frac{Ta\ (n-1)}{n^{1\cdot 4}\, T}\right) - n^{\cdot 4} T}{n - 1}$$

$$= \frac{n^{1\cdot 4}\, T + T_a\ (n-1)\ -\ n^{\cdot 4} T}{n - 1}$$

$$= n^{\cdot 4} T + T_a$$

Similarly, the terminal temperature may be conveniently expressed in terms of T and T_a, for

$$QT_t = \frac{QT_e}{n^{\cdot 4}}$$

$$= \frac{n^{\cdot 4}\ T\ +\ T_a}{n^{\cdot 4}}$$

$$= T + \frac{T_a}{n^{\cdot 4}}$$

Curves of temperatures and pressures calculated from the foregoing equations are shown in Fig. III., which is extended to include all values of n between 1 and infinity. When n is unity, all curves indicating temperature under the condition of continuous running, converge to the temperature $(T + T_a)$ When n is infinite the curves of initial and terminal temperatures converge to T, and the curves of compression and explosion converge to infinity. Curves of pressure converge to infinity, when n is infinite, and to atmospheric pressure when n is unity. The curves are calculated on the basis of $T_a = 2{,}000$,

Number of Expansions

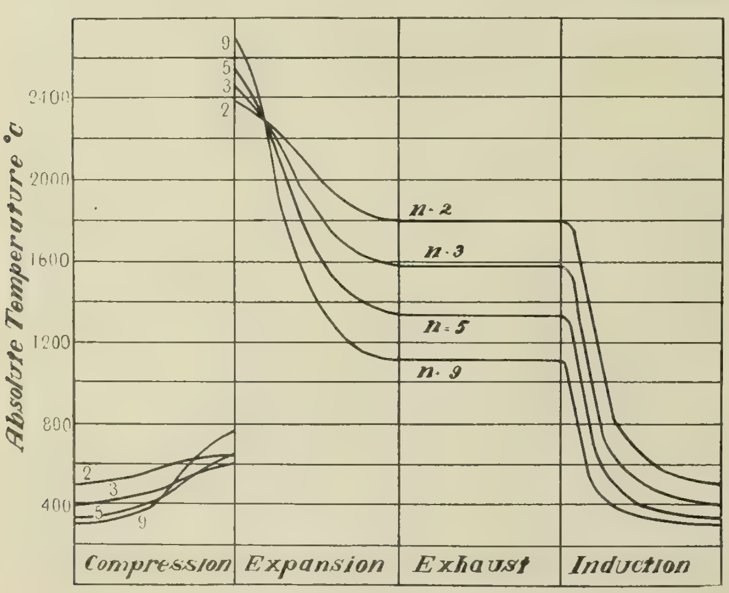


Fig. VI.

which represents a fairly weak explosive mixture. The curve representing mean temperature is plotted from the following considerations :—

If Fig. IV. represent a curve of temperatures during the compression stroke, the temperature at any volume, V, will be :

$$\text{Temp.} = \frac{QT_c}{V^{\cdot 4}}$$

and the mean temperature between $(V=1)$ and $(V=n)$ will be :—

$$\text{Mean Comp. Temp.} = QT\ \frac{(n - n^{\cdot 4})}{\cdot 6\ (n - 1)}$$

$$= T\left(\frac{R\,n - 1}{R\ (n-1)}\right)\ \left(\frac{n - n^{\cdot 4}}{\cdot 6\ (n-1)}\right)$$

Similarly the mean temperature during the expansion stroke will be :—

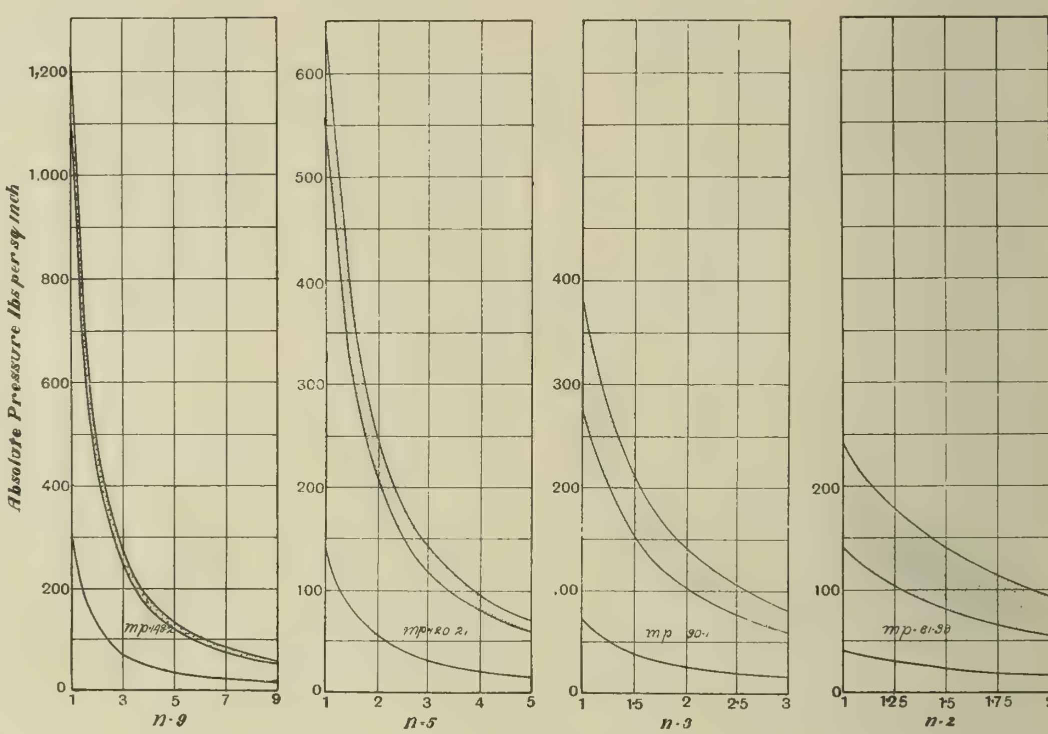


Fig. VII.

$$\text{Mean Exp. Temp.} = RQT\ \frac{(n - n^{\cdot 4})}{\cdot 6\ (n-1)}$$

$$= \frac{T(Rn\ -\ 1)}{(n-1)}\ \ \frac{(n - n^{\cdot 4})}{\cdot 6\ (n-1)}$$

The temperature during the exhaust stroke is :—

$$QT_t = RQT\ =\ \frac{T\ (Rn - 1)}{n - 1}$$

If Fig. V. represent a curve of temperatures during the induction stroke, the temperature at any volume, V, will be :—

$$\text{Temp.} = T \frac{RQV}{RQ\ (V - 1) + 1}$$

and the mean temperature during the induction stroke will be :

$$\text{Mean Ind. Temp.} = T\left\{1 + \frac{(RQ\ -\ 1)\ \log_e (Rn)}{(RQ)\ \ (n-1)}\right\}$$

From these equations I have also plotted Fig. VI., which shows graphically the temperature during a full cycle of two revolutions for four rates of expansion, viz. :—

$n=2$, being a low rate of expansion,
$n=3$, being a medium rate of expansion.
$n=5$, being a high rate of expansion.
$n=9$, being a very high rate of expansion.

The scale of abscissæ is proportional to time, and the curves are therefore shown with the distortion due to the circular path of the crankpin, but neglecting the obliquity of the connecting rod. The following table gives the principal figures :—

	$n = 2.$	$n = 3.$	$n = 5.$	$n = 9.$
T_a	2000° C.	2000° C.	2000° C.	2000° C.
T (abs)	290° C.	290° C.	290° C.	290° C.
R	3.613	3.963	3.897	3.546
Q	1.723	1.374	1.186	1.090
RQ	6.226	5.446	4.622	3.864
Abs. Initial Temp. = QT	500°	399°	344°	316°
Abs. Comp. Temp. = $n^{.4}QT$	659°	618°	655°	761°
Abs. Expl. Temp. = $Rn^{.4}QT$	2382°	2451°	2552°	2699°
Abs. Terminal Temp. = RQT	1805°	1580°	1341°	1120°
Abs. Mean Temp.—Comp. Stroke ..	567°	481°	444°	434°
Abs. Mean Temp. —Expansion Stroke ..	2048°	1905°	1729°	1539°
Abs. Mean Temp.—Exhaust Stroke ..	1805°	1580°	1341°	1120°
Abs. Mean Temp.—Induction Stroke ..	771°	583°	459°	383°
Abs. Mean Temp. of Cycle	1298°	1137°	993°	869°
Heat Efficiency = $\left(1 - \frac{1}{n^{.4}}\right)$	24.22%	35.56%	47.47%	58.47%

Some instruction may be derived from Fig. VII., which shows graphically the pressures due to the temperatures given in the table. The diagrams are supposed to be from cylinders having equal lengths of stroke, but different compression ratios. The shaded portion of each diagram represents the difference between what may at this stage be called the "true" theoretical pressure and the "ideal" theoretical pressure arrived at by neglecting the existence of the residual products of combustion in the compression space, the "ideal" being always Q times the "true." This graphic representation of Q is the chief point of interest in the diagram.

Since we assume the same quantity of mixture to be taken into each cylinder it may be anticipated that the mean pressures will have some simple relation to the heat efficiencies. We have seen (equation 1), that the mean pressure is $2.5\,p\,(R-1)\frac{n^{1.4}-n}{n-1}$, and since $R = 1 + \frac{T_a\,(n-1)}{T\,n^{1.4}}$ the expression for the mean pressure may be written $2.5\,p\frac{T_a}{T}\left(1-\frac{1}{n^{.4}}\right)$, that is to say, that for the same value of T_a the mean pressure in each cylinder should be directly proportional to the efficiency due to its degree of compression. There may be reasons for modifying this statement later, but it is true at the present stage of the argument. Note in Fig. VII. that the scale of pressures for the left-hand diagram has been made for convenience one-half of the scale for the others. Note also that the limit of mean pressure when n is infinite is $2.5\,p\,\frac{T_a}{T}$, which with the present values attached to the symbols amounts to 253.4 lbs. per square inch.

The figures in the preceding table assume a low heating value for the explosive mixture, and yet with the high rate of nine expansions it is apparent that the mean temperature of the cycle is such as would necessitate external cooling. The general indication is, however, that by using a sufficiently high compression ratio the mean temperature of the cycle might be reduced to a figure at which an engine could work continuously. The theoretical limit of temperature in this direction, that is when n is infinite, is in fact $\frac{4\,T}{3} = 387°$ C., abs., a temperature not greatly in excess of the boiling point of water. It is recorded that it was anticipated that the Diesel engine would be able to dispense with external cooling on account, among other things, of the high compression employed, but the anticipation was not practically verified.

The theory of the internal combustion engine, with which we have been dealing, ignores the envelope of the charge, and although this proceeding constitutes a perfectly legitimate and useful preliminary, it can only be accepted as final on the clearly fallacious assumption that the envelope has no physical properties in relation to heat. The complete elucidation of the action of heat in a cylinder is a problem of staggering complexity, but some advance may be made by adding to the purely mathematical theory as already set forth, the effect of attributes of the envelope which are fairly deducible from experiment. To arrive at definite quantitative results it will be necessary to put a definite value on the attributes, but any error of estimate, while it may misrepresent results in the matter of magnitude, will not destroy their apparent tendency.

(To be continued.)

A YEAR'S DEVELOPMENT IN AERONAUTICS.

The progress made in construction during 1911.

By W. G. Aston.

THE aeroplane developed into a practical vehicle with such unprecedented rapidity that little more than a year ago it reached a stage at which its development could hardly continue to be so sensationally marked, and from which further steps towards perfection could only be made, as it were, in short strides compared with that which had taken place previously. The same state of affairs has been undergone in the development of the motor car, though the slow-progress stage of aeroplane design has naturally been reached very much earlier than in the case of the road automobile, since the former has been, to a very large extent, dependent upon the development of the latter. Even when one realises this, one is fairly justified in complaining that progress in aeronautics has not proceeded at the pace it might have done, and it is therefore perhaps well—before reviewing what has been done—to consider the factors which have combined to provide the retardation. It may, however, at once be said that indications are not wanting that the prime cause of this retardation is being realised, and one may safely look to the very near future to see it, at all events partially, removed.

As is so frequently the case, the first practical production of the aeroplane was made by men whose theoretical knowledge of the subject was, to say the least of it, comparatively slight. Their work has since then been discussed and expanded to an enormous extent by mathematicians. Yet, whilst aeronautical knowledge has grown with an almost incredible speed, practical mechanical progress in aeronautics has lagged considerably behind, for the reason that there is a very marked difference between the constructor of aeroplanes and the man who has made the study of dynamics of flight his immediate interest. This circumstance, which however has probably reached and passed its acutest form during the last two years, has been a very unfortunate one, since the aeroplane and its development have already, to a large extent, become a commercial matter, in consequence of which it is as a rule only the men who have made a name by the production of some sort of successful aeroplane at an early date, who have been able to obtain patronage and the all-important financial backing. And to make matters even worse, it is this type of man who is unhappily all too prone to regard the theoretical standpoint as a quite unnecessary adjunct to practical achievement. One is therefore not surprised to find amongst the biggest firms in the industry that there have appeared at times unmistakable signs of positive retrogressiveness. On the other hand, the present year has seen the springing up of several smaller firms whose policy has been founded upon a more stable basis, and it is therefore clear that it cannot be long before those whose names are the first in the industry will realise that further progress will be more economically obtained by theoretical research rather than by experimental trial and error.

Another factor which has helped considerably towards impeding progress has been the beatification (one had almost said, the canonization) of the aviator who pilots the machine, rather than the constructor or designer who has made his feats possible. To take a horse-racing analogy, the jockey has received all the credit more properly due to the horse. This in itself would be a trifling matter, scarcely worth mentioning, were it not for the fact that several daring pilots who have contrived to win lavish prizes, have applied the money towards constructing a new make of machine, and in the majority of cases these new constructors have joined the army of so-called "practical men," as opposed to the theoreticians. As has been pointed out, the fact that the art and industry have been thus divided

into two opposing camps is a matter that cannot fail to find its own adjustment in the course of a short time, and although, therefore, of merely transient importance, it is impossible properly to estimate the value of aeronautical progress made during the past year without taking it into consideration; and it is certainly of passing importance, inasmuch as it largely explains why at least as much progress in the design and construction of aeroplanes has been made in England as in France. On the Continent patronage of the aeroplane is readily forthcoming, not only from the pockets of private individuals, but also by the direct and well-considered encouragement of a progressive and far-seeing government. In England both sources of such encouragement have shown themselves deplorably lacking. On

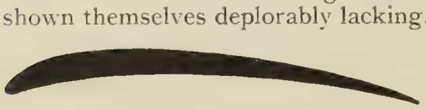


Fig. I.

the other hand, this very fact has proved an incentive for the British constructor to avail himself of every possible means for improving his machine. Whilst, therefore, the French manufacturer is turning out his aeroplanes in dozens (while his British rival is manufacturing in units), the home manufacturer has been practically forced to turn the greater part of his attention towards improvement in design, and that this has actually been achieved is sufficiently proved by the undeniable fact that more than one essentially British design has obtained marked adherence on the Continent. As far as progress towards the perfect aeroplane is concerned, honours between the two countries are easy, though this is very far from being the case when the magnitude of the industry in each country is considered. Both in this country and abroad, the attention of the aeroplane designer has been devoted almost exclusively to an improvement in the speed qualities of the machine, apart from the obvious method of increasing the power. This tendency provides the most notable feature during the past year's progress; a second one, which is to a considerable extent dependent upon and arising out of the first, being the influence which the design of the monoplane has had on that of the biplane. Since higher speed for given engine power can only be obtained by means of a reduction in head resistance, every effort has been made to bring the latter to its lowest possible figure.

This end has been sought in three directions. Firstly, by completely covering in the fuselage, and in some cases by making it a shape approximating to a body of stream line form. Secondly, by employing wing sections by which the component of drift is reduced without diminution of the component of lift. Thirdly, by completely encasing the engine, and as far as possible also the passenger. The covered-in fuselage succeeded instantly in gaining popularity, but although it certainly possesses many valuable advantages, it did not succeed in establishing itself as a step towards progress until its principal disadvantages had been removed. That is to say, a covered-in fuselage, unless of well-considered design, is capable of promoting a *side*-resistance which, in tending to turn the machine about its vertical axis, is bound to make its control difficult when flying across the wind. This trouble has been overcome in three principal ways: In the Depurdussin monoplane by making the fuselage very shallow and furnishing it in front with a deepened cockpit to provide accommodation for the pilot and passengers; in the Bleriot and Handley-Page monoplanes by merging the fixed horizontal tail-plane extension gradually into the sides of the fuselage, so that in cross section it passes progressively from a square to a flattened rectangle; or, as in the Breguet biplane and Paulhan monoplane, making the fuselage of practically circular section from stem to stern.

The second method of reducing head resistance, viz., that of improving the form both of the plan and the section of the wings, is certainly of fundamental importance, though in this connection it is not unfair to say progress has done little more than just commenced. With regard to the section of the plane, this has undergone a considerable amount of alteration and modification from that which previously found almost universal adherence, but in very few cases is the change very apparent to the casual observer. The principal tendency is to adopt the form originally designed by Mr. Horatio Phillips, which has been done notably in the Nieuport monoplane and Astra biplane. The Phillips section, Fig. I., is contrasted with the ordinarily accepted type, Fig. II., Fig. III. representing the section employed by Cody, from which very excellent results have been obtained. As will be seen, it takes the form of two independent stream line bodies placed tandem.

With a view as much to the reduction of head resistance as to providing an aid to the acquisition of natural stability, the general tendency is now to make the depth of the plane, the width of the plane, the angle of the plane, and the camber of the plane diminish progressively from the root of the wing to the tip, the general "washing out" of curvature and angle, resulting in a flat formation at the tip of the

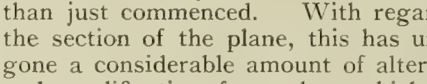


Fig. II.

wing, which to a considerable extent reduces the effect of end losses. That is to say, the disturbance of the air between its uninterrupted state and its interrupted state is effected as far as possible progressively. With the same end in view, attempts have been made to obtain a better plan form, especially with regard to that of the tips of the wings. The worst possible form in this respect is the rectangular, which is, however, adhered to in a number of biplanes in which constructional facilities have been considered to be of more importance than freedom from end losses. The square tip is also retained in the Antoinette monoplane, but in all other machines of this type the tip is either rounded or else either the front or rear corners of the wings are clipped off. Although fundamental theoretical considerations would lead one to suppose that the greatest length of the wing should be found along its entering edge, several constructors claim advantages for an exactly opposite system, in which the leading edge is shorter than the trailing edge; that is to say, in which the front corner of the wing is lopped off. In the case of wing tips, however, the shape adopted is largely dependent upon constructural facilities, but on the other hand there are indications that in the near future more attention will be devoted to this important point, as constructors are beginning to realise that for the time being the aero-dynamic value of a design is of more importance than any difficulty which may crop up in putting that design into actual practice.

Enclosed Engines.

The third method in which reduction of head resistance has been sought, is that of hiding the engine as far as possible inside the fuselage. As the prime mover for aeroplanes, the Gnome engine stands in a

Fig. III.

class entirely by itself, but it would be idle to deny it has a certain number of disadvantages. These, when the flying speed of the machine is a comparatively low one, are easily outweighed by the inherent advantages of a type of engine which gives a wonderfully uniform torque, is always in perfect balance, is satisfactorily air cooled, occupies a very small amount of space and requires no fly wheel. It is only when really high speeds are entered upon that the Gnome engine exhibits its weakness: that it of itself offers a very large area of resistance, both to its own rotation and to the forward travel of the machine to which it is fitted. Neither of these troubles are very easily overcome, and hence it is clear that at a certain point such advantages as the Gnome engine has will be outweighed by its disadvantages. That this point has been reached, or will very soon be reached, is evidenced by the fact that out of 29 machines taking part in the recent French Miltary Trials, only 12 were fitted with Gnome engines. The reason for this apparent discarding of a motor which has had more hand in the progress of the aeroplane than all other aviation engines put together, is not far to seek. The disc area of a Gnome engine is, at the very least, 4 sq. ft. (probably considerably more), and if the machine be flying at 60 m.p.h., the pressure upon this area will be somewhere about 43 lbs., the overcoming of this pressure requiring an expenditure of approximately 7 h.p. This, of course, is a very heavy penalty to pay for the advantages of a rotating-cylinder engine, especially when it is remembered that a very considerable amount of power is absorbed by the engine in overcoming the direct rotational resistance of its cylinders. The stationary, water-cooled engine, on the other hand, may readily be boxed in completely at the prow of the fuselage, whilst there is no difficulty in finding a suitable position for the radiator, which need not, however, be made to offer more than a negligible amount of head resistance. The removal of the head resistance of a stationary engine is a perfectly easily accomplished matter, which can bring nothing but good in its train, whereas to gain the same effect in an air-cooled rotating cylinder engine is a decidedly difficult problem. There is, therefore, a decided tendency to

rehabilitate the stationary engine, especially for high speed work, but it is very unlikely that engines of the Gnome type will be superseded entirely for a considerable time. On the other hand, stationary radial engines, both air-cooled and water-cooled, have made some strides in favour, although at best neither of these types can be much more than partially covered in. On the new Paulhan-Tatin monoplane, which has been taking part in the French military trials, an attempt has been made to do away with the resistance of a Gnome engine by placing it completely inside the stream line fuselage, and allowing a certain quantity of air to be thrown on to the cylinder through louvres arranged circumferentially round the fuselage. Whether this device will prove entirely satisfactory remains to be seen, but if properly carried out there is little reason why it should not prove advantageous. Although the draught of air thrown on to the engine will not be great, it is probable that it will prove sufficient, since there seems little doubt that in its usual position, immediately in the wake of the propeller—to the slip-stream of which it is fully open—a Gnome engine succeeds in keeping itself considerably cooler than it should be, and this no doubt accounts for part at least of its comparatively low efficiency.

The Coanda biplane, entered for the French military trials, exhibits an arrangement of the Gnome engine which is decidedly interesting from the fact that although the engine is, as usual, fully exposed, the resistance is very considerably reduced. Two engines are employed, and are fixed one on each side of the fuselage, instead of directly in front. In this position they drive the tractor screw through bevel gearing. This arrangement, if proved to be satisfactory in practice, may easily lead to the adoption of a similar one on a number of machines, for it is obvious that a single Gnome engine could be arranged to provide the same advantage, viz., diminution in head resistance, by being mounted in a horizontal plane, either above or below the fuselage. It is true that the necessary gearing involves a slight loss in efficiency, but as we are to see later there is a strong tendency for the geared-down propeller to come into vogue.

The above are the principal methods by which reduction in head resistance has been achieved, but attention has also been paid in this respect to the landing chassis, notably by employing stream line covers for the running wheels, or making the latter of the disc type. It is highly questionable whether the ultimate gain will be worth the trouble involved, even supposing that no other disadvantages be deduced. This latter hypothesis is not, however, so easily disposed of, since the area of such stream line casings, etc., may, and probably will, be sufficient to cause some effect at least upon the machine's lateral stability when it is flying across the wind. At the same time it must be admitted that the design of landing chassis in general has not reached a very high standard, and that they are not only, with few exceptions, cumbersome, but they also offer a decidedly sensible amount of direct head resistance.

It is in the endeavour to reduce head resistance that the modern biplane has been brought so closely into line with the monoplane, inasmuch that in comparing the latest examples of the two types it may almost be said that they differ only in the number of the planes. The modern biplane is in fact a very considerable advance on its predecessor of twelve months ago, and for this the greatest credit is certainly due to Roe, who was the originator of this particular type, which is well exemplified in the Avro and Breguet machines. Far greater progress has been evinced in biplane design than in that of monoplanes, with the result that whilst the former type is well capable of lifting a considerable weight, its speed capabilities have been improved to such an extent that the best of its kind are comparable with the fastest monoplane of equal power. This is almost entirely due to the employment of a covered-in fuselage, by means of which the head resistance of both passengers and engine has been reduced to a reasonable figure, and also to a slight extent—as in the case of the Bréguet—to a construction which has largely done away with the necessity of furnishing a maze of tie wires and a plantation of struts between the planes. The latest biplane designs would seem to show that the front elevator, either independent or connected with a controllable tail flap, has had its day, the modern biplane being controlled entirely by the organs of the tail. A great improvement has also been manifested in the disposition of the centre of gravity and the centre of thrust, which has in most cases been raised sufficiently high to permit of the advantages of a non-lifting tail. As one result of this, the appearance of the machine has been generally considerably improved, at the same time its structure has become far more rigid and less liable to derangement. With the propeller in front, as has become almost the accepted practice in biplane design, the landing chassis can be considerably neatened and its ugly, leggy appearance largely avoided.

Automatic Stabilising.

During the year past little has been done towards an advance in the introduction of means for providing a modicum of automatic stability, beyond the general employment of non-lifting tails. Nearly all designers are satisfied as to the great merits of this type of stabilising device, though it is worthy of note that Blériot has returned to the lifting type of tail in the 140 h.p. monoplane which he entered for the French military trials. It is certainly difficult to assign an adequate reason for this apparently eminently retrograde step, but as practically all modern Blériots of other types are furnished with non-lifting empennages, it is improbable that the present case is anything but an experiment. The virtue of the non-lifting tail is roughly demonstrated by the following device :—Let lines be drawn perpendicularly to the chords of both the main plane and the tail plane (such lines representing the pressure reactions upon the planes). If the lines intersect above the machine the latter is supported in a state of stable, longitudinal equilibrium, and is therefore analogous to a pendulum suspended with its bob-weight below the point of support. Any oscillations which take place in the vertical plane will accordingly tend to "damp out," the time taken for this being dependent upon the weight of the machine and the height above it at which the perpendicular lines, above mentioned, intersect. It must be remembered in this connection that in the case of a machine with a fixed flat tail, behind which are controllable elevating planes, that the pressure line will not in this case be drawn perpendicular to the controlled plane, but to an imaginary line which represents the *average* angle of incidence of the plane taken as a whole.

The employment of non-lifting tails has rendered the control of the machine, when the power of the engine is cut off for a glide, a much more easy matter than was previously the case with a lifting tail. In the latter circumstances the removal of the slip stream of the propeller, from which the tail plane derived the greater part of its support, caused the rear end of the machine to drop, to correct which such an inclination of the front or rear elevator plane had to be made as might seriously compromise the natural longitudinal stability otherwise possessed by the machine. Even with a non-lifting tail the same drooping occurs to a small extent, and it is therefore quite possible that future machines may be furnished with a tail plane having a slightly negative angle of incidence, the effect of which would be that when the slip stream of the propeller was removed the tail would—without any adjustment on the part of the pilot—tend to lift the rear of the machine and so automatically alter the attitude of the machine in such a way as to make it suitable for a glide with the power cut off. For lateral stability the dihedral angle is generally used, except in such machines as the Farman biplane, in which the centre of gravity is placed slightly below the centre of pressure of the planes in order to gain the same pendulum effect.

In the Dunn, Etrich, Weiss, and Handley-Page monoplanes the main wings are so designed to be self-righting, that is to say, they have, without the aid of any subsidiary tail, a slight degree of natural stability both longitudinal and lateral. In these machines the plan-form of the wings is approximately that of a flattened letter V. The angle of incidence is greatest at the apex, which is in front, and from thence to each tip both angle and camber of the plane grow less and less until a point is reached at which the angle of incidence is neutral. From this point onwards to the tips, the angle is increasingly negative. It will easily be seen that the two negative tips of the wing, considered separately from the rest of the plane, have exactly the same effect as the single non-lifting plane, which is ordinarily supported at the end of the fuselage. A negative angle is used, however, inasmuch as the moment of the plane tips about the centre of pressure is comparatively small. Hence the force required to provide the desired turning couple is proportionately large. No separate tail at all is used in the Dunn machine, but in the Weiss, Handley-Page, and Etrich monoplanes a secondary tail is employed, principally with a view to gaining the same stabilising effect without introducing wing tips turned down to a marked extent.

Another reason for adopting the V plan-form is for the purpose of obtaining a degree of lateral automatic stability without introducing the disadvantages which are inherent to the dihedral angle.

If one of the above mentioned machines cants over on its horizontal axis, the tendency for the whole machine is to slip down in the direction of the lower wing, and in doing so it essentially changes its path of travel from a line parallel with the horizontal axis to a line at an angle to this axis, such angle being dependent upon the angle to which the machine is tilted. When this becomes the case, the arrangement of the plan form of the wings ceases to be symmetrical, and the lower wing becomes effectively of greater value than the upper, since while its aspect ratio is increased, the aspect ratio of the upper wing is decreased, or in other words, the lower wing undergoes an increase in the length of its effective entering edge with the result that a restoring couple is formed, which tends to set the machine on an even keel again.

Machines of the tail-first type show strong signs of gaining in favour, and there are now three members of this class which, so far as full sized machines are concerned, originated with the British Valkyrie. These are the Voisin-canard and the Blériot-canard, the first being a biplane and the second a monoplane. From what has already been said with regard to natural longitudinal stability, it will readily be perceived that if the fixed tail plane be placed in front, it can be set at an angle of incidence considerably greater than that of the main plane, and can thus be made to do a considerable amount of lifting. That is to say, with given planes of equal area, the tail-first machine, at the same speed, will lift more than the ordinary type, besides having two other advantages, viz., that with a plane set out on an outrigger in front of him, the pilot is afforded the greatest ease in controlling the machine, as the effect of his control is at once clearly and unmistakably visible to him; and secondly, that with the propeller behind the main plane, neither of the sustaining surfaces work in the slip stream of the propeller, and hence when the engine is cut off, no sudden controlling movement has to be carried out to counteract the effect arising from the absence of slip stream. This type of tail-first machine, in which the front plane is fixed and the control of the machine carried out by a secondary movable front plane, must not be confused with a machine of the original Wright type. For whereas in the latter the front plane had at times to be set as entirely to destroy the natural longitudinal stability of the machine, in the former the employment of a fixed front plane ensures that the point at which this action takes place is extremely unlikely to be reached.

It is rather singular that whilst most of the other organs of the aeroplane have undergone a certain amount of improvement, very little has been done towards perfecting the landing chassis, although it is true efforts have certainly been made to simplify it, to lighten it, and to reduce its resistance. This, however, is not so much the point; the fact being that there are practically no chassis suitable for anything but prepared aerodromes and large fields. Skids have come into practically universal use, but their full value is not yet taken advantage of, for while they might be of decided value in running upon roughish ground, they are rarely arranged so as to be anything but a hindrance in running the machine over anything but smooth grass.

Generally speaking, the two principal disadvantages of the accepted forms of landing chassis are: firstly, that there is an insufficiency of suspension, and secondly, that satisfactory means are not provided for the landing chassis as a whole to be universal. The value of both these points is too great to be overlooked. As a rule, a pair of wheels are sprung with rubber bands on to skids which are fixed to the fuselage of the machine with rigid struts. The wheels being comparatively small in diameter and the rubber buffers not much stronger than is necessary to carry the weight of the machine, a fairly abrupt landing brings the machine down on to the skids as harshly as if there were no shock absorbing device at all, and the result is that a considerable stress is imposed over the whole of the construction. With a fixed type of chassis again, landing across the wind, which often occurs in cross country flying, either in the case of a forced descent or in landing where there are no very large fields, becomes a matter of considerable risk, and not infrequently results in part of the landing chassis being carried away, or else in one of the wing tips being damaged. It seems fairly obvious that if the aeroplane is to have any future at all, the greatest efforts must be made to make it as far as possible independent of prepared grounds, and it is scarcely too much to say that at the present moment it would satisfy nearly all requirements in this respect were more attention paid to the landing chassis.

During the past year several constructors have realised the disadvantage of employing the landing chassis as a king post for the triangular staying of wings, viz.: that the effects of a rough landing, when this is done, more often than not extend to the wings themselves, instead of being localized on the running gear.

Amongst other tendencies which appear to be leading up to desirable improvements, must be mentioned that of discarding the direct driven propeller in favour of an indirectly driven one running at a lower speed than the engine. It is clear that this must essentially make for higher propelling efficiency, since the power absorbed by a propeller varies as to the cube of its speed, whereas the thrust is proportional only to the square. Further, within reasonable limits, there is no reason why the propeller should not be increased either in diameter, blade-area, pitch, or number of blades, as there are few aeroplanes in which a larger propeller than is used at present cannot be conveniently installed. The use of gearing is, however, open to several serious objections, the principal ones being the considerable increase of weight involved, and the large amount of power losses introduced into the transmission. This being the case, it is clear that there is certainly an opening for a low-speed aeroplane engine giving a reasonable output, at about 700 revolutions per minute, and there is very little doubt indeed that if such an engine were put upon the market it would meet with a considerable demand. It is not surprising to find that in France 4-blade propellers are coming into vogue, and there is every reason to think that they will continue to do so, as on theoretical considerations (when properly designed) this type represents the best possible form. One is also not surprised to find that the metal propeller has practically ceased to exist, wooden ones having been adopted by firms such as Voisin and Antoinette, both of whom had previously been strong supporters of the metal screw.

One of the most important events of the year—at all events in so far as the design of the aeroplane is concerned—was the recent introduction of a double-engined biplane by Messrs. Short Brothers of Eastchurch. This is the first of its kind that has made any public appearance, and that it has proved itself an unqualified success must be a source of gratification to the British industry. Briefly described, the Short double-engined machine is a biplane of the Farman type with a non-lifting tail and a front elevator. A Gnome engine in the usual Farman position drives a direct-coupled propeller immediately behind the middle of the lower plane. Directly in front of the main planes are two tractor screws, each placed at some little distance from the axis of the machine, and driven through chains at half the speed of the second engine, which occupies the position usually taken by the pilot in a Farman machine. Both engines are 50 H.P. Gnomes, and whilst the diameter of the three screws is about the same, the pitch of the propeller screw (which runs at engine speed) is about half of that of the tractor screws. An important point is that the line of thrust of the tractor screws is considerably above that of the propeller, and hence by varying the power of the two engines (between which there is no coupling) the height of the resultant centre of thrust can be altered or lowered at will. The machine, when once in the air, flies easily with the power of one engine, either when applied through the tractor screws or through the single propeller. It will be realised that this machine represents the first case of an aeroplane with any pretensions to having a variable speed, or having a stand-by engine for use in case of emergency. By suitably controlling the two motors, the speed of the Short biplane can be varied over some ten to twelve m.p.h. (maximum and minimum), no little achievement in itself. It need hardly be said that the advantages of a machine which can fly at 60 m.p.h. and land at 48 m.p.h., are very pronounced in comparison with one that must land and fly at the same speed.

With the exception of the Short biplane, no other notable attempts appear to have been made to gain this highly desirable end. Important experiments have been carried out in England, France, and America with hydro-aeroplanes, and in each case considerable success has been obtained, though only over very smooth water. The inherent difficulties in making a machine of this type negotiate rough water seem to make it open to doubt whether, so far as the aeroplane may become the scout for a warship, the experiments are being carried out along right lines.

Developments in the dirigible balloon have served only to accentuate the limitations and disadvantages of a "lighter-than-air" system, and there is very little reason to think further experiments will be persisted in. It is worthy of note, however, that during the past summer a very fairly successful airship passenger service has been maintained in Germany.

THE PROGRESS OF MOTOR BICYCLE DESIGN DURING 1911.

AT the recent Olympia Show it was apparent that considerable forward progress has been made with the design and construction of motor cycles. Generally speaking, the details which have been improved or re-designed are those more nearly connected with the comfort and handling of the machine, than any special efforts to obtain greater efficiency or service. In the engine unit magnetos are for the most part placed in a position where they are little likely to obtain a liberal quantity of dirt or dust, designers being greatly aided in this by the enclosed magneto as supplied by several of the magneto firms, although one firm has deliberately placed its magneto lower than usual, and below the silencer. When in position, the magneto machine is nearly always driven by chain, and in one particular case by a silent chain, although there are still a number who use a train of spur wheels as a magneto drive. One is led to wonder what particular advantage can be found from such an arrangement, since it is so very easy to design a magneto base plate that can be adjusted to take up any wear which may occur in the chain. Moreover, spur gears are not easy things to cut even in the haphazard manner in which such things are generally treated in motor cycles. In running, they are not nearly as quiet as the ordinary chain drive, and they entail numerous comparatively large bearings, which it is very hard to lubricate properly. Why a silent chain should have been adopted for the drive of a magneto on a motor cycle it is also rather hard to see, since the ordinary roller chain needs but little attention, is practically quiet, if kept properly adjusted, and, one is glad to remark, perfectly easy to adjust in the majority of modern machines. One point which is still exceedingly striking when considering the magneto in itself, is that no proper fixing seems to be coming into favour, similar to those used in motor car manufacture. In two cases only from the whole Show were motor cycle magnetos attached to their bases by the quick acting clamps so familiar to car designers and so convenient to all private owners, on the rest four hexagon bolts were inevitably used. One would think that such an attachment would be more important on the motor cycle than with a car as the proper adjustment of the chain drive is almost a necessity. As to the engines, the absence of any length of valve guide is extremely noticeable. Most machines still have the stumpy, ill-proportioned guides, which, it is safe to say, will develop as much undue wear as their predecessors. Some firms are wise enough to place a thin insulating washer between the valve spring and the cylinder flanges, a most necessary precaution, the lack of which was responsible for many broken exhaust springs during past years. Many more, however, while omitting this safeguard, have also provided a valve spring with the smallest number of turns possible to place on that particular valve, though one would imagine that Brooklands experience would have taught them the value of a well proportioned spring, even in touring vehicles. As to the valves themselves, a few seem to be about half the cylinder bore in diameter, with well radiused heads and commendably light all over, but there are still great quantities of designers who adhere to a heavy, peculiarly clumsy valve, apparently without thought of the weight or stress that valve may cause. Cotter pins are no longer seemingly composed of portions of iron wire, but for the most part are good solid pieces of metal incapable of shear, and yet not sufficiently large to cause breakage in the valve stem foot. Another excellent point is the immense number of exhibitors who are fitting an adjustable tappet head, mostly consisting of a hexagon-headed set screw with its attendant lock nut. This is infinitely better than the immense circular cap which had to be replaced for adjustment, and which usually raised the valve from the extreme edge of its diameter. No motor cycle engine can stand much play between the valve tappet and the valve stem which it operates, and it is therefore all the better when such play can be taken up by a spanner and not by removing the valve in order to replace the mushroom head of its tappet.

Pistons seem considerably lighter than they were, and many are as light as one could possibly wish for the engine size, far lighter in proportion than a number of car engines. They are also, as a rule, better designed than the motor car piston, although the gudgeon pin fixing is still confined in the majority of cases to a set screw, which one hopes will soon disappear. A tendency certainly exists to pay much greater attention to the lubrication of all bearings. Nearly every stand had a machine fitted with some form of lubricator which could deliver oil without the assistance of the machine's rider. It is true that most of these devices are the suction operated drip feeds with small sight glasses, similar in practically every detail to those which have been discarded from the dashboard of a motor car for a great number of years, mostly on account of the trouble given by such an arrangement during the winter, the delicacy of the needle valve, where grit is likely to interfere with the flow, and the trouble caused by splashed oil obscuring the sight glasses. It would seem that the adopted form of lubricator is open to all these faults and possesses but one advantage over the motor car drip feed in that it is a good deal easier to make a delicate adjustment of the needle valve. Still, the adoption of even such a form of lubrication instead of the old hand pump is a forward step which cannot but have an effect on the efficiency of the machine, while the presence of one machine with a mechanically forced feed pump, and another with the full sump and oil pump, shows that attention is at last being given to one of the most important details in motor cycle construction.

Of carburettors there is little to be said, since the only radical alteration is the lengthening of the inlet pipe to accommodate the new position of the magneto. Control is still operated by friction levers, which move air and throttle slides exactly as they have for the past few years, and there seems little tendency to do away with the hand air lever in favour of a type of carburettor similar to that used on motor car engines. Silencing has been given a little more attention since a certain foreign machine demonstrated that an engine could be powerful as well as quiet, and one or two machines have exhaust pipes of the type associated with the tourist trophy races, but having a few minor alterations to accommodate them to the touring engine. It is to be hoped that further steps will be taken in this direction, as it is perfectly possible to design a machine with some degree of quietness and with all the power that it needs during its progress through towns or villages, and it is to be hoped that before long the main attribute of a motor cycle in the eyes of the public will not be an open exhaust of the type which even Brooklands has now objected to. Noise as a problem does not seem to have been considered by motor cycle designers in the way it possibly should have been. Few have made any effort to quieten the rattle of their valve gear, although this may be due in some part to the rapid action cams which are generally adopted. In one case a strange and unusual valve gear has been adopted, which might possibly be quieter than the ordinary type, but it had so little bearing surface and revolved at such a speed that one could hardly imagine it would be satisfactory under usual road conditions for so small an engine. In some cases a curious attempt had been made to enclose the valve stems, after the manner connected with motor car engines, but this had been done in a very half-hearted manner, and there was usually a gap of half an inch or more at the back of each cover through which dust and dirt could enter with the greatest facility. Why this had been done is not very clear; it could hardly have had any effect on the silence of the machine and had no protective qualities whatsoever. Why a ridge had not been left on the cylinder so that a faced joint could be made is equally unapparent.

There is a tendency to lower the compression even further than was done last year, possibly because there is still considerable trouble from pre-ignition on long runs. With the engine control much has been altered. In the first place a considerable number of machines had no less than six levers on the handlebars. In many cases this seemed to have been done in order to get rid of an extra lever which went to make up a set from the manufacturer. Throttle and air with the forward brake would be on the right-hand side, while a similar pair of levers had been mounted on the left. One of these was wanted, and rightly so, for the ignition advance, while a separate one on the same handle bar is nearly always used as a valve lifter. The trouble then seems to have been determining what could be done with the sixth lever.

Some designers had attached it to the cut-out, some to the exhaust valve lifter; nobody seemed to have any clear idea of what to do with it, and consequently attached its wire to anything which caught their eye first. It cannot be said that all these levers, on top of the clutch pedal, the brake pedal and the foot cut-out, are things which add to the comfort of riding, or make it any easier for a beginner, and it is believable that a radical change will come upon these levers, which will leave three on the handlebar at the most, and the brake and clutch pedals. At last it has been thought necessary to tackle the starting problem and now practically every make has its free engine with hand of foot starting gear, and consequently one of the bad disadvantages of a motor cycle have ceased to exist. As a result of the tourist trophy change speed gears have come into great prominence, and owners of side car machines or lightweights will have less physical labour in future than has been their unfortunate lot in the past. That all these gears are not of the type which would be termed efficient, or even accessible, goes without saying, but it is so pleasing to notice the adoption of any form of variable gear that it is only to be hoped that each will prove perfectly satisfactory. There still exists a considerable amount of confusion in the minds of designers as to what exactly happens when a motor cycle is travelling over a rough surface, and their subsequent solutions of this difficulty are only indicated in the somewhat diverse arrangement of their spring forks, which act in practically every direction it is possible to allow.

Presently somebody will be induced to carry out a series of experiments and determine the amount and direction of the principal forces in a front wheel when subjected to severe shock, and if this is carefully done the result will probably entail complete alteration of many of the spring fork designs which are on the market.

With a very few exceptions mud guards remain of exactly the same dimensions and general design as they were last season, and the only machine which appears to have made any progress in this direction is one of the two-strokes. One of the great drawbacks of motor cycling is the extreme quantity of mud and dirt which is collected by the rider if he attempts to use his machine during the winter. It would, therefore, seem an excellent policy for designers to study the question most thoroughly to see whether mud guards could not be greatly improved and the machine rendered more pleasurable in consequence. So far no real thought has been given to the cross section of the guards, which readily allow mud to creep round their edges and be carried back to the machine or its rider. Of course, mud guards have always been one of the largest and most difficult problems which have ever confronted a designer, because it is not atall easy to design a guard which will protect the machine from mud and still allow the amount of accessibility which is absolutely necessary when dealing with tyre repairs. Another matter which one is pleased to see is receiving attention is the isolation of the rear wheel by means of springs. There have been in the past a certain number of attempts to design a machine whose frame shall be sprung from both wheels, but of late such movement seemed to have ceased until the Show. There, however, there were quite a number of different designs, which provided an increased comfort for the rider and practically a complete isolation from shocks to the rear wheel. It is a point worthy of much attention, because at the present moment a motor bicycle is not at all pleasant on a very bad road, and it is practically impossible to tour over any length of road without running over parts where the surface is exceedingly tiring.

Amongst the various motor cycle stands there were some which exhibited a three-wheel vehicle, in many cases termed a "runabout." Whether the coming year will see the revival of a three-wheel vehicle is an open question; it is very hard to believe that any satisfactory service can be obtained from such a construction, because, unless great care is used in disposition of weights, a three-wheel vehicle is peculiarly liable to sideslip in greasy weather, and the writer has known one machine, which was probably the most successful of its kind, but whose side-slipping possibilities were diabolical. What advantages three-wheel constructions have over four it is very hard to see, because although a differential may be omitted from the design, yet this is not such a great advantage as it seems, because the clutch type of differential seems to be perfectly satisfactory and is quite cheap to manufacture. In the vehicles exhibited there were very few which showed any sign whatever of having been designed. Most of them seemed to be put together with a view of evolving the strangest and most unmechanical creations which could be possible in modern times. In one case a machine was driven through a double jointed propeller shaft and great care had been taken to erect the gearbox so that the driven shaft was exactly in the centre of the rear wheel, consequently necessitating a shaft which was perpetually working at a great angle on both the inadequate universal joints. In itself the gearbox was not a satisfactory form of design, but any of its faults were nothing as compared with the final drive. In one other case a peculiar front axle had been adopted which, had it been properly designed, would have been perfectly free and perfectly easy to steer. In the example shown it was apparently necessary to use the full force of one hand to turn the wheel at all and, as a minor point, the driver's seat was in such a position that he could not possibly obtain a good grip of the wheel. In nearly every case the ultimate object of the designer seemed to have been evolution of some extraordinary outline.

As far as final drive was concerned, there was not the increase in chain drive which might possibly have been expected from the tourist trophy result, and in nearly every case where chain drive had been adopted there was little or no attempt to protect those chains from either mud or dust. Chain driving on motor cars failed as a result, not of chain inefficiency so much as the total lack of any protection, from which came abnormal wear in the links, great noise and general transmission troubles. It is curious that nobody has attempted to make a really satisfactory casing for a chain driven motor cycle, because such a machine probably retains more mud than any other road vehicle, and consequently it is not at all likely that a chain drive can be kept free from the wear which proved so fatal in the motor car. It must be remembered that the one real advantage chain drive has over the belt must be the absence of small adjustments and general troubles of an irritating nature, and that no chain can possibly be free from these troubles for any length of time if it is freely lubricated with road grit or mud.

Unfortunately it is still as true as ever that the majority of machines are more copied with variations than designed, and there is very little reason to suppose that any lessons whatsoever are being taken from those parts of a motor car design which might have any bearing on motor cycles. One after another the old attendant difficulties of some form of fitting crop up in motor cycle design. Exactly the same arrangement has given exactly the same troubles with the motor car, and these have been overcome and a comparatively effective design arrived at. Much useless labour and expense would be spared if some notice were taken of the way in which these difficulties had been overcome, instead of attempting new solutions of one's own, or adopting fitments which have already been discarded by some other form of vehicle. Motor cycle manufacture must be considered a branch of engineering in a far greater sense than was ever attached to the building of the ordinary bicycle, as motor cycle designers are far too inclined to adopt bicycle tactics on many occasions and do not realise the true seriousness of designing a mechanically propelled vehicle.

THE INSTITUTION OF AUTOMOBILE ENGINEERS.

The first informal meeting of the Institution of Automobile Engineers was held in London on November 22nd, when a discussion took place on "The Trend of Engine Design at Olympia."

Mr. E. J. Burt, in describing the new Argyll sleeve valve engine, produced curves showing how the power was maintained at a very much higher number of revolutions than with the poppet valve engine made by the same company. He stated that an engine of 101 mm. x 130 mm. gave 52 h.p. at 2,000 revs., and averaged about 3 h.p. per 100 revs. up to 1,500 revs. per minute.

Mr. Cook then described the Lamplough two-cycle engine with the double syphon cylinder arrangement, and Mr. L. A. Legros briefly described the Lucas two-cycle engine as well as the rotating D-valve Darracq motor.

Mr. A. E. Berriman described the four-cylinder rotary valve Itala engine, in which each pair of cylinders has one valve containing four ports and controlling the inlet and exhaust, the valves being driven at a quarter the speed of the crankshaft.

Mr. A. E. Crowdy then described the Hewitt piston valve engine, claiming that the design followed steam engine practice.

An interesting discussion followed, the principal speakers criticising the various designs, which had been described, but no very definite conclusion was arrived at. It was decided by vote to hold further informal meetings of similar kind.

THE 15 H.P. CALTHORPE CHASSIS.

An endeavour to combine high power with moderate manufacturing cost.

THE makers of the Calthorpe cars have arrived at cheapness of production on a comparatively small output, principally by means of reducing everything to its simplest possible expression. Here and there some small amount of efficiency or convenience must necessarily be sacrificed in the pursuance of such a policy, but much less than would be the case if a more complex design were attempted to be cheapened by scamping quality of material or accuracy of workmanship.

The 80 mm. bore engine, which is shown in Fig. 1 on this page, has a stroke of 150 mm. and has large valves of good shape, their diameter being 42 mm.

Contrary to usual European practice, the cast iron piston carries a pair of bushes in which the gudgeon pin bears, itself being fixed by pinning in the small end of the connecting rod. On the score of bearing surface this arrangement probably has but small advantage or disadvantage as compared with normal design, though it to an extent simplifies the fixing of the gudgeon pin, and a little weight might also be saved perhaps. Otherwise the piston is of light section, but its length would seem abnormal for the bore, if not for the stroke.

It may be noticed that the cylinder casting is open-ended, making core withdrawal easy, and that aluminium plates are used as covers, that at the front end also bearing the water intake pipe—cooling being by internal connection circulation.

Two inlet passages pass between the cylinders, and the carburettor has therefore a short branched pipe attachment, but the exhaust arrangement is quite peculiar. In order to follow the reason for the peculiar design of pipe it is necessary to remember that the order of firing is 1, 3, 4, 2, and it may then be observed that cylinders discharge alternately into the upper and lower branches of the outlet pipe respectively. This is claimed to prevent or diminish exhaust interference, and to give a steady flow through the silencer.

It is to be regretted that no adjustment is provided for the camshaft driving chain, but this must perhaps be counted as one of the necessary sacrifices mentioned in the first paragraph. On the other hand, both the crankshaft and camshaft are well proportioned and supported in bearings of fair area, the only criticism possible being that there is considerable overhang between the front crankshaft bearing and the sprocket. It would be easy, too, to add a small bearing at the extreme end in the place now occupied by the starting handle spring.

Lubrication is performed by a plunger pump (the plunger being of the same diameter as a tappet and ball operated in the same way), sucking from the sump through a large filter and delivering to troughs of a wide and shallow section. It is possible to remove the pump and filter complete without the loss of any oil, and without obtaining access to any awkwardly placed nuts. The high position of the pump is no disadvantage, considering the powerful suction of a plunger type, and the accessibility is extremely commendable.

The plate clutch follows usual practice and needs no detailed description, though the striking gear is neat and very easy to remove, as may be judged from the plan view of the chassis in Fig. VI. Thence the drive passes to the gearbox through a coupling which allows a small amount of disalignment. Gears are made from Poldi 10% nickel steel, and are 6 diametral pitch, while the change is controlled by an ordinary sliding gate lever. Although most of the bearings rest in aluminium housings, it is noteworthy that there is a cast iron piece carrying the bearing inside the foot brake drum, where the wrenching stresses are greatest.

For the securing of the sliding gear sleeves, two feathers are used, both the shaft and the tops of the feathers being ground to size.

Concerning the rear axle, the details shown in Fig. III. are sufficient completely to explain it. Thrust from the hubs is taken on the driving shafts

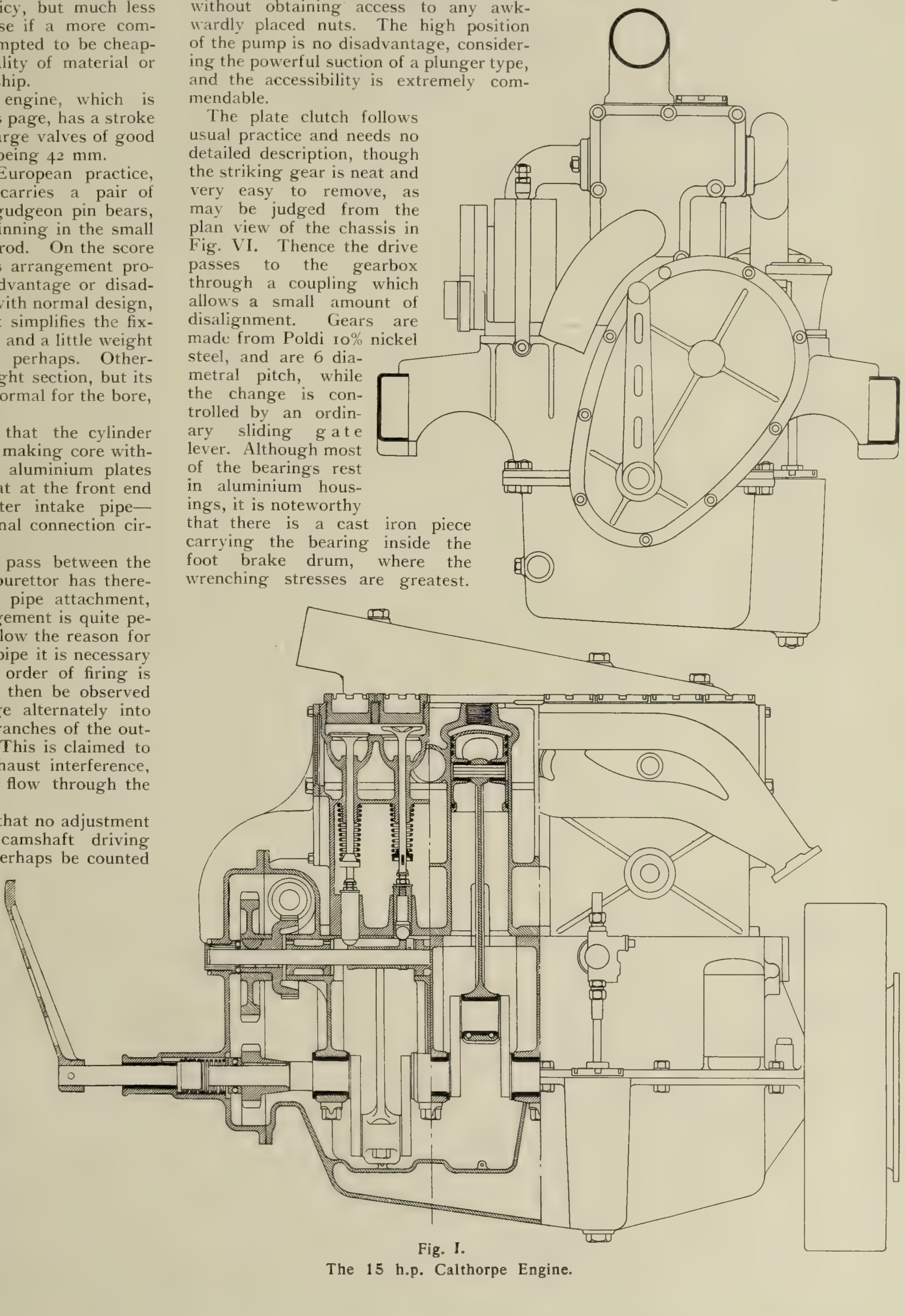


Fig. I.
The 15 h.p. Calthorpe Engine.

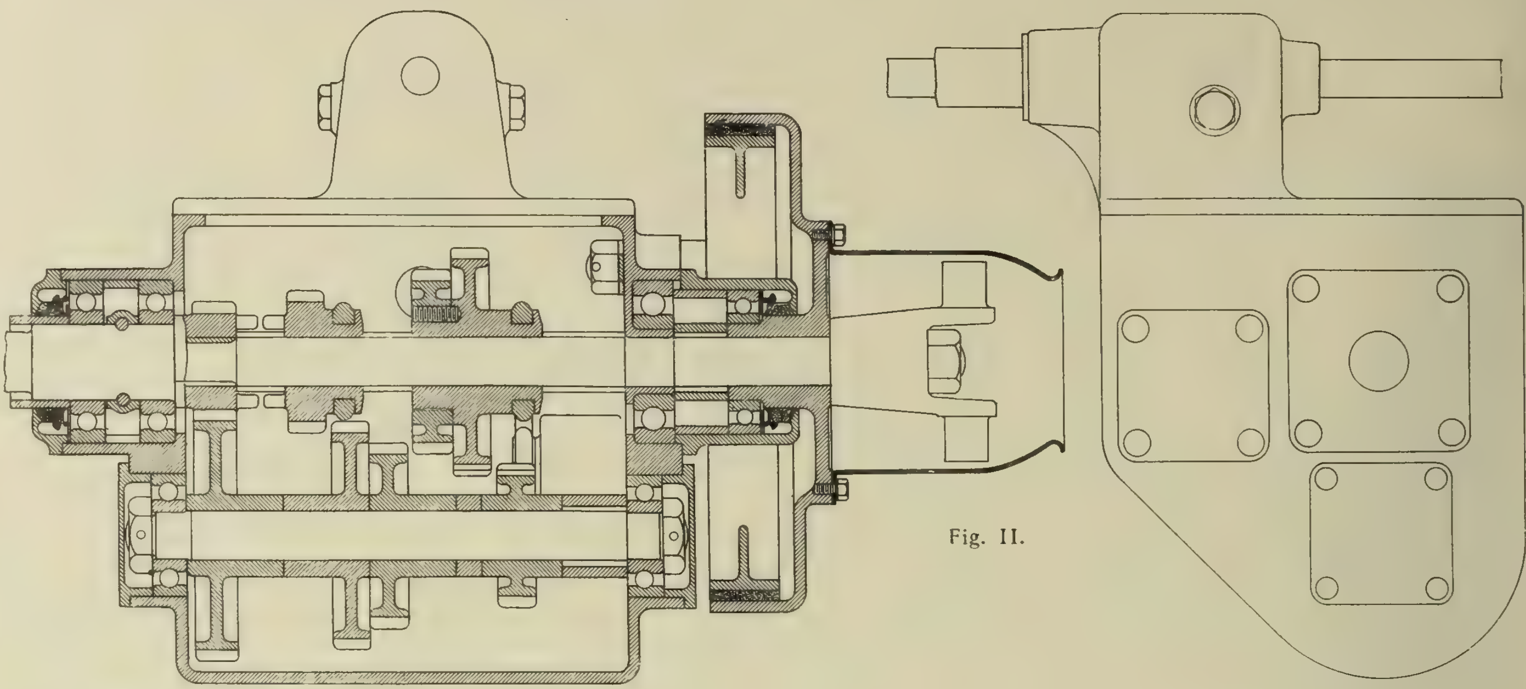


Fig. II.

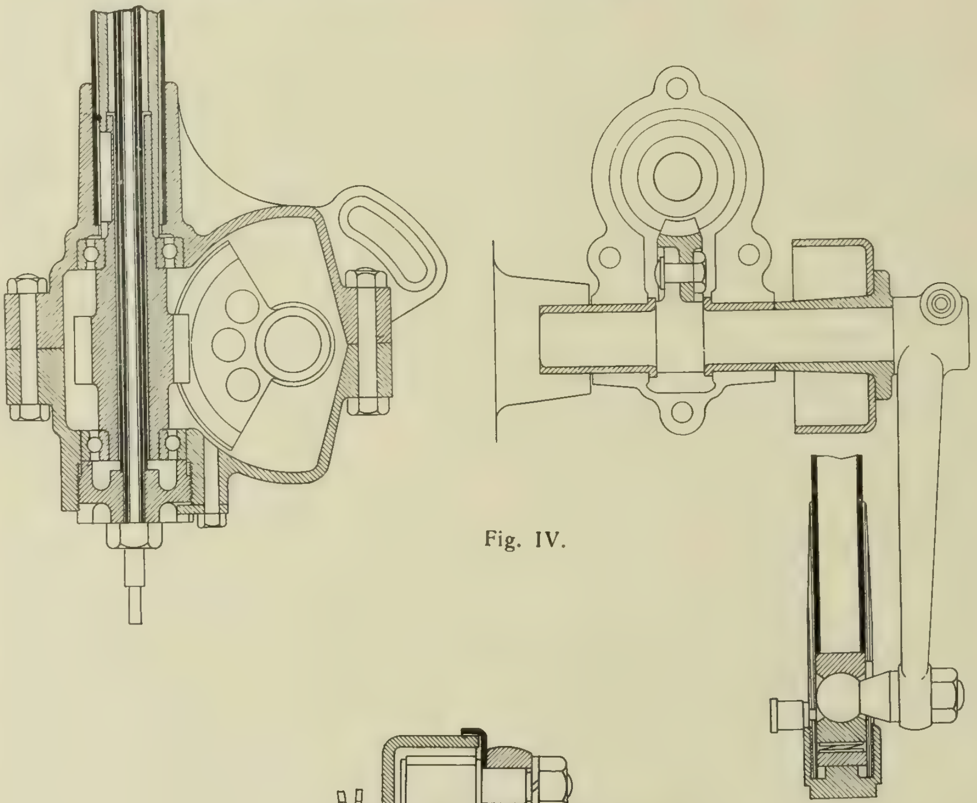
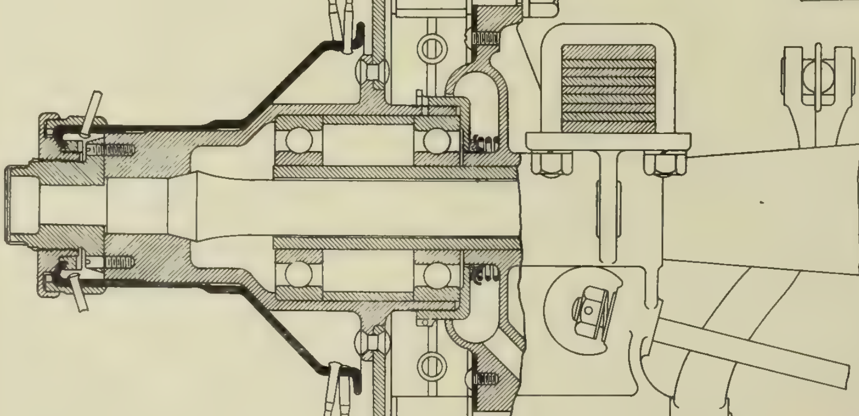


Fig. IV.



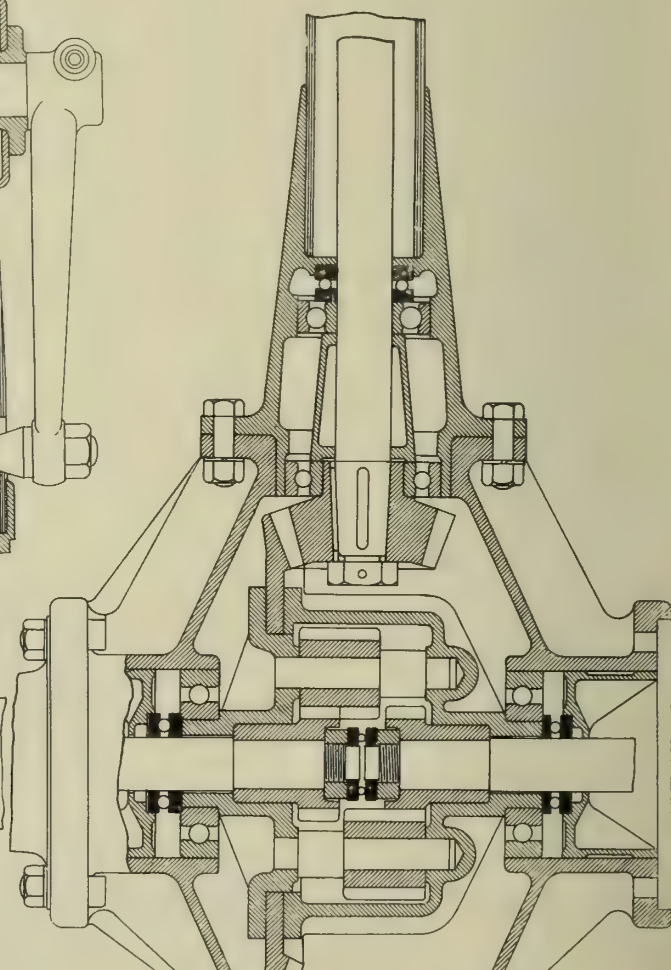


Fig. III.

TRANSMISSION AND STEERING DETAILS OF THE 15 H.P. CALTHORPE CHASSIS.

and passed through the small ball washer which separates them, to the thrust bearings which position the differential. It is also worthy of note that any oil escaping from the axle ends is caught in a chamber formed in the brake cover and fulcrum plate, whence it runs to the actuating shaft bearing.

There is an adjustable angle to the steering column, the method of obtaining which can be seen in Figs. IV. and VI., while Fig. V. shows the decidedly neat front axle swivel and the correct position of the ball joint lubricators. Most other details are shown in the chassis views, Fig. VI., save the fact that the front cross member is dropped deeply, and the gearbox attachment. Being pivoted at the centre of the cross member just over the clutch shaft, the gear is brought into alignment by manipulating set screws in the two aluminium arms at the rear end, bearing on the middle cross member. When the right position has been found, holes are drilled and horizontal bolts put through the arms and the cross members.

Ample rigidity has been given to the frame by the width of the channel at the point where it is inswept, and it is noticeable that the depth of the pressed side members is considerably above the normal towards the front end, where the section changes to form the front dumb irons. The middle cross member, too, which carries the gearbox and the pivots for the propeller shaft casing, is well proportioned for the stress it is called upon to resist. Care has been taken that the centres of these pivots shall lie on the horizontal axis of the universal joint and so reduce telescopic movement of the shaft to the minimum called for by the method of rear springing. The fork has a large bearing on the front end of the propeller shaft tube, and a grease cap is fitted at this point—the need for this is by no means always remembered.

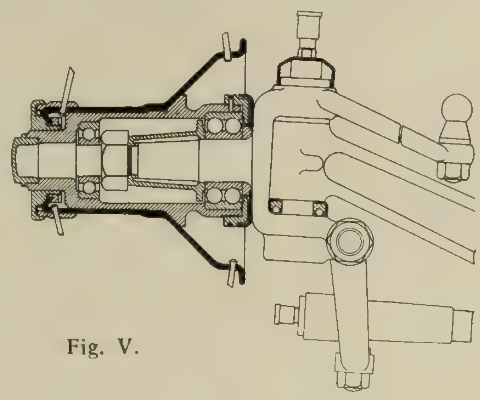


Fig. V.

To a certain extent the triangular stays from the shaft casing to the ends of the rear axle can act as radius rods and so tend to maintain parallelism between the axles.

Concerning the brakes, these follow standard practice for the internal type, with the exception of the ingenious method of lubrication for the hub brakes, actuating shafts which has already been mentioned. The operating shaft for the foot brake is carried through the gearbox, so that all connections are both short and direct, while there are easy adjustments for both pedal and hand lever.

In matters of detail the car is well cared for. Springing and brakework are perhaps especially commendable in action, while the engine is not unduly hard running considering its rather high compression and considerable speed of revolution.

A boxed-in gate is employed, the actual striking mechanism being normal, but all the bearings in it are of sufficient size to make the handling quite free, which may be quoted as an instance where the use of the simplest form can by careful design give almost as good an effect as a more nearly ideal, but more complicated arrangement which would, of course, be considerably more costly to manufacture.

The cooling being by natural or convection circulation, there has been some difficulty in ensuring proper circulation round the carburettor jacket, to overcome which a chamber has been formed in one piece with the exhaust pipe, lying between the front and second cylinders' outlets. Water from the carburettor is taken through this chamber on its way to the radiator, the heating it receives therein being sufficient to ensure rapid circulation. While on the subject of cooling it may, too, be remarked that while the water is taken in at the front end of the cylinder casting, there is a baffle plate in the cast aluminium outlet pipe which compels the exit to be made above the rearmost pair of cylinders, this method being found as effectual as the introduction of baffles or passages inside the casting, while being much more simple. Taken altogether the chassis has obviously been thought out with unusual care and the makers' racing experience shows up in many details.

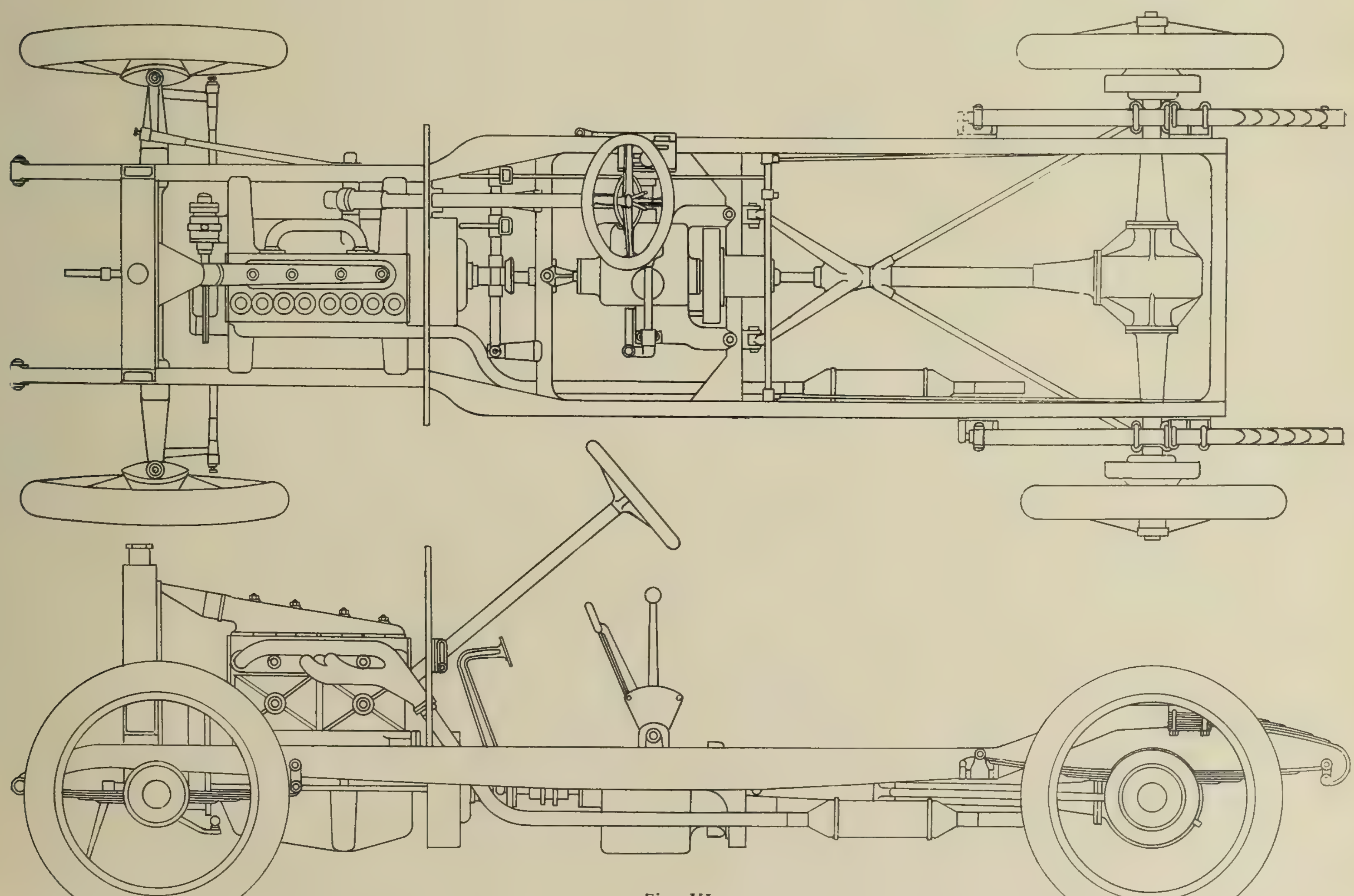


Fig. VI.

The 15 h.p. Calthorpe Chassis

CHASSIS DESIGN.

Some Comparisons between English and American Designs, being a paper read before the Institution of Automobile Engineers on November 8th, 1911.

By Howard E. Coffin, M.S.A.E.

This being the first paper on automobile engineering generally read by an American engineer before a British engineering body, it has been deemed fitting to print not only the paper itself in full, but the speaker's introductory remarks. Mr. Coffin refers to this meeting as being the "first annual joint meeting of the Institution and the Society; whether such meetings will be able to be held every year from now onward is as yet not quite decided, but it is certain that many joint meetings will be held from time to time, and it is hoped that the next will take place in the United States in the summer of 1913, if not next year.

Mr. Chairman and Gentlemen,—

Let me thank you briefly for the kind words of welcome of your honourable Chairman. As merely a speaker of the evening upon matters more or less technical, I believe it is not on the cards that I should in any way make formal reply. All this may come, I trust, later in the week, when some one of our real orators may be given an opportunity. Let me say simply that we are glad to be with you.

As for myself, I feel in addressing you to-night that a great honour rests upon me. I feel that it is an honour, not only because at this particular time, upon this particular occasion, I am upon my feet among you as the representative of your co-workers across the water; but because by the very nature of things this joint meeting is the first of many future meetings of this kind. I feel a responsibility, not only as an engineer addressing himself to the members of an honoured engineering body, but I feel a responsibility doubly great because, by your invitation, it is my privilege to address this, the first annual joint meeting of the two newest and most progressive engineering organisations of the two greatest nations on earth.

Whether these joint meetings shall be held in England or in America, or whether we shall alternate between these two countries, it is not now a time to discuss; but joint sessions of this kind have come to mean much in other, older, and slower going industries than ours, and there is not one of us upon either side of the water but would be broadened and benefitted by some such programme.

Twelve short years, at the most, may be said to have spanned the real life of the horseless vehicle—its rise from a curiosity and a joke to its position as a necessity and as an economic factor in the progress of our modern civilisation. This same time has seen an infant and experimental enterprise claim for itself the place, perhaps of third, and certainly of fourth, in the list of the world's greatest manufacturing industries. Within this same period transportation has been revolutionised, both as to its commercial and social aspects. And within even a shorter period it has been required of the motor car engineer that he do more than has been accomplished in any other line in half a century. The real engineering history of the horseless vehicle has yet to be written—for so rapid has been the growth of this industry and so strenuous has been the life of its engineers, that little time has been permitted for the verification and accurate record of that mass of practical and theoretical engineering data which must sooner or later be collected together as the very Bible of the motor car designer. All of us must contribute to this work—all of us must sooner or later meet together upon the questions of international standards, upon uniform practices in specification and legislation. And it would seem that this present joint meeting may well be expected to break the ice for a shoulder to shoulder co-operative work which must reflect much of credit and of benefit upon all of us.

I believe that it was suggested by your secretary that the American speaker of this evening deal with chassis design. Therefore, a word upon this subject, lest I be accused of wandering too quickly into matters general.

An opinion has been voiced in the English technical Press and by various English visitors to the States, that while Europe, and particularly England, has excelled in high speed motor construction, America has excelled in the chassis work. I can readily understand the reasons for such an expression of opinion, but I cannot so readily bring myself to believe that there is much of real fact upon which to base such an impression.

Blessed as you are with your wonderful roads, it seems impossible to me that you should entertain a proper respect for your own handiwork. To put the matter squarely, I think it is only when as a visitor to America, you see a car negotiating the seemingly impassable, broken, and sometimes almost bottomless country roads, which abound in many of the States of the Union at certain seasons of the year, that your respect for running gear construction ever reaches the proper high water mark.

I must confess to a feeling little short of awe at the amount of use, misuse and abuse which can be withstood by a motor car in the hands of the average American user. Except for certain structural features affecting road or ground clearance, I do not see why the American chassis should perform better than the English—due consideration being had, of course, for the relations of power, weight and road condition.

Good materials and good workmanship are certainly much the same the world over, and while in England a higher average of attention is certainly given to fineness of finish, there can be but little difference in workmanship upon the vital parts. I am referring, of course, to the reasonably well-built cars upon both sides of the water.

A comparison of the engineering practice entering into the better class of American cars does not show any greater degree of uniformity than is to be found in the similar class of English cars. Various spring suspensions are to be found; single and double-jointed cardan shafts are in evidence; some rear constructions are fitted with a torque absorbing member and some are not; the drive of the road wheels is transmitted to the main frame through the springs, through radius rods, or through a tubular housing for the propeller or cardan shaft, etc., as may suit the whim, the pet theory or the necessities of the individual designer. Whatever the dictates of theory, these varied constructions seem, when well made, to perform in an equally satisfactory way, or at least in a commercially satisfactory way. There are some peculiarities in American chassis design. Some serve an engineering purpose; some meet a commercial demand; others would seem to be without excuse.

High powered motors are a necessity; firstly, because of the excess of brute power required to negotiate the poorer class of roads at speed; and secondly because of the antipathy of the average American towards the use of the gear shift lever. He may demand that his car be equipped with a four-speed gearbox, but is disgusted when he fails to take everything on high. Some concerns regularly fitting four-speed and reverse boxes do it perhaps rather for a selling argument than that they expect the low gear to be used.

As nearly as can be judged at the moment, the coming of the long stroke will mean no lessening of bores—merely an increase in powers. A six-cylindered car, for instance, formerly 5 x 5½ inches in bore will for this year be 5 x 7. Four-inch engines are moving up from 4 x 4½ to 4 x 5, 4 x 5½ and even 4 x 6, with no very great changes otherwise in the chassis or weight of the finished car. The low price of fuel, and the low rate of, or total absence of h.p. taxation, tend to encourage rather than discourage this advance in power. A broad statement may almost be made that no American car above the smallest two-seater will carry a four-cylinder motor of less than four inch bore. One or two instances such as the Humpmobile may be said to be the exceptions which prove the rule. Another peculiarity often remarked by those from this side of the water is the brake location. Double brakes upon the rear wheels have become an almost universal practice. Almost every argument in the category can be cited in favour of the cardan shaft service brake. European precedent is a unit for it. Considerations of weight, cleanliness, freedom from grit and wear, ease of adjustment, ease of operation, simplicity of operating mechanism, the equality of the retarding action upon the rear wheels—all favour it.

But early in the making of American motor car history the engineers of two or three of our leading makers found it impossible or inadvisable for structural reasons to fit cardan shaft brakes. Thus early began a campaign of publicity and of education for the double rear wheel

brake. Many engineers favour the cardan, but the buyer wants the rear wheel type, and the manufacturer builds the car to suit the buyer. It is really as simple as A B C.

The underslung frame is another case in point. Fathered for several years by one concern whose management believe in it, lately taken up by one or two concerns who see in it the possibilities for publicity and for an appeal to the human desire for something new and different, it is possible that this construction may become a minor factor. It is a significant fact, however, and I have it upon good authority, that electric railway locomotives built upon this principle, with the centre of gravity well down in the plane of the axles, have been found to possess a lateral rigidity so great as to render them practically useless. All cushioning effect due to the sway so noticeable in the locomotive at speed seemed lost, with the result that the wheel flanges refused to hold the rails. Just how far this experience may hold as indicating the influence that an absence of body sway may have upon tyre wear or upon the road behaviour of a car at high speed, remains to be seen.

The character of American roads and the conditions under which American cars are forced to perform have had necessarily a great influence upon the direction of detail development. Road clearances, spring lengths and spring clearances, the avoidance of extreme lengths in wheel base, precautionary measures against squeaks and rattles through leather and rubber liners, bronze bushings, numerous grease cups, etc., the deep ribbing or beading of all sheet metal surfaces for the prevention of vibration, the secure locking of every bolt and nut in its place, these and a hundred and one other things are being given a particular attention because of the nature of the service to be encountered at the hands of the American user.

It is one of the peculiarities of the American industry that the engine there is, in many things, forced to strike a kind of commercial average. I believe that in no other car-producing country in the world will there be found such extremes in the operating conditions under which a car must perform as in the States. Every day in the year cars are expected to do service with a minimum of adjustment or change upon the part of the operator to meet climatic or road conditions. Now, the laws of that same quantity production which permits of low-priced production, require that duplicate cars be shipped East, West, North and South. Duplicate cars are therefore put into service upon the century-old and perfect roadways of New England, and are with equal confidence expected to negotiate the dirt roads and trails of the newer Western States. The same cars in every detail are asked to perform at the sea level in New Jersey, and among the clouds in Colorado. They must behave equally well in the scorching sand deserts of the South-West, where rain never falls, and in the humid air of the North-East.

While I am speaking to you to-night, duplicate cars are being shipped from a dozen American factories—some to the North, into a winter temperature of from ten to forty degrees below zero Fahrenheit—some to the South, to Florida and to Texas, with the thermometer at above one hundred in the shade, and the roads hub deep in sand. One car may be used upon the beautiful roads of the East, where speed limits are unknown, another may encounter the daily gear work among the Alleghenies or the Rockies.

The engineer's complaint sheet is a curious medley. Cylinder compressions are too high, cylinder compressions are too low; cooling systems are too efficient, and cooling systems overheat; second and third speeds are too low, second and third speeds are too high; carburettor float levels and a dozen other things do not quite suit the varied conditions of temperature, altitude and road. So you see that you do not have all the trouble in merry old England.

American cars, and hence American standards of automobile engineering, were for years regarded as somewhat of a joke upon this side of the water. But because American design differs from those principles which have been found to succeed in Europe, it must not be assumed that the American designer is ignorant as to the trend of practice upon this side of the water. The light, high speed, small bore motor of twelve to fifteen h.p. would perform admirably upon the English-like roads of New England, but would fail miserably in the face of the hub-deep sticky "Gumbo" of some sections of the middle West.

Ample road clearance is an absolute necessity upon any car which is built in quantity to meet a nation-wide market in the States. Among the visiting contingent in this audience I can note the faces of some of those who with me years ago joined the "Sadder but wiser club" as regards this subject. Hence larger wheels, greater axle and fly-wheel ground clearance, and a generally somewhat higher car appearance than is usual upon this side.

Now let me turn to some of the matters which I believe may be of more interest to you as indicative of new departures, or new trends in design and construction. Some of these tendencies are coming to us, of course, from this side—others are home grown—the result of engineering advancement and manufacturing refinement in some instances forced by that hardest of all masters, commercial competition.

Few cars of any power or price will go into the next season without self-starting motors. Several are already fitting starters as regular equipment for this season, and many others are following suit. A dissertation as to the merits and demerits of the several systems in use would consume the entire evening. Electric, acetylene, and compressed air starters are now actually being supplied as stock equipment by several big makers. Whether one of these types will ultimately supplant all others is an open question. Certain it is that the acetylene starter will be used in by far the greater quantities. Moreover, by reason of its extreme simplicity in construction and operation, and by reason of its supply of a graduated combustible mixture to each cylinder under any weather or temperature conditions, it would seem to have rather the better of the argument.

Any way round, the average American buyer has become convinced of the necessity for a self-starter, and a self-starter he must have.

Next the lighting problem. Six months ago it looked like a walk-away for the electric generating outfit. The ability to turn lights on and off at will—the ability to relight the lamps with the snap of one's fingers at the wind, and at that last remaining match in one's pocket—all these things appealed wonderfully.

But the fears of electric competition have brought gas lamp improvements, whereby one may, with a single switch turning movement from the seat, turn on and light the acetylene gas lamps. The common use of the acetylene self-starter would also seem to argue well for the retention of the gas light in combination.

I am not sure that you are all familiar with the gas tank situation. Small gas tanks holding sufficient acetylene gas in storage for from one to two month's lighting or for three thousand engine starts can be had in every village and hamlet in the entire country. Upon the payment of a nominal fee and the return of the empty tank, a fresh tank will be attached in brackets universally supplied by car makers for the purpose. The car owner has no worry as to the empty tank—it goes back by a regular routine through the village dealer to the parent company for refilling and reshipment. Altogether this system provides a very cleanly, light weight, and simple solution of the car lighting problem.

The long stroke motor I have already touched upon. As to *en bloc* castings and the protection of working parts, I do not believe that the newer American cars will be found to differ greatly from European practice.

I have already mentioned one or two chassis peculiarities, double rear wheel brakes, etc. To these I might add the frequent positioning of the gearbox upon the rear axle member. I presume that the arguments pro and con upon this subject are much the same with you as with us—weight and tyre wear upon one side and gear quietness with manufacturing and repair advantages upon the other.

Of course, no engineering discussion would just now be complete without some mention of special motor valves. The Knight motor you know. It is now being adopted by four makers in the States, and apparently with success. There are several American valve mechanisms which seem to promise well. Every one is of the rotary disc or rotary valve type. It seems to be the American creed that if the poppet valve is to be dropped it will be a mistake to replace it with another reciprocating mechanism—sleeve or otherwise. Hence, a concentration upon that type of valve action which may be accomplished by a mechanism free from reciprocating parts. Some of these constructions have shown a fine performance on the road, as the difficulties would seem minor when compared with those overcome by Mr. Knight earlier in the art.

Clutches.

It is not to be expected that the coming annual motor car show in January in New York will bring out any marked change in clutch construction. The cone will probably show a percentage of increase, not because the disc and other types are being abandoned, but because a larger number of the newer and cheaper models will

carry the cone as the cheapest form of the reasonably satisfactory clutch. There is one form of clutch coming into use in the States which is, I believe, entirely unknown upon this side of the water. I refer to the multiple disc form using cork for the friction surface.

That this style of clutch has not come into general use in America is due to two causes—first on account of the royalty required under the patents covering this construction, and second, because of a lack of engineering data as to the proper use of this very wonderful material. Let me say, while I think of it, that there are no patents upon the use of cork either in clutches, brakes, or factory belt pulleys in this country. Let me say also, that after an experience covering cone, metal disc and plate clutches, as applied to many thousands of cars produced by those companies with which I have been associated for the past years, I will make the broad statement that the properly constructed cork surfaced clutch will distance all others in smoothness of action, length of service, and freedom from trouble both in the factory and upon the road. A flushing out with kerosine, perhaps once a month, followed by the addition to the case of half-a-pint of mixture of kerosine and ordinary thin gas engine oil, will keep the action perfect, no matter what the temperature or weather condition. In theory of design the main points are, first to make the surface ample for the work to be done, and second, to provide a clutch spring strong enough to insure against any continuous slippage after engagement has once been effected. A satisfactory form of construction on a car of from thirty to fifty h.p. would consist of twelve to thirteen steel discs of say eight inches outside diameter by perhaps six inches inside hole, thus giving discs of one inch face width.

These plates would alternate thick and thin—the thin ones micrometring to about one-sixteenth to three-thirty-seconds of an inch, and the thick to three-sixteenths or a bit more, more or less. In each of the thick plates may be punched from forty to fifty half inch holes, slightly staggered, so as to distribute well over the entire face of the disc.

Into these half-inch holes are then forced by machine operation under pressure, ordinary straight corks of a diameter greater than the hole—a diameter of perhaps eleven-sixteenths or even three-quarter inch.

The corks should first be soaked in oil or warm water and should be allowed to dry in the air or in a dryer after the discs are filled. The projecting cork surfaces may then be trimmed and sanded down to within a thirty-second of the metal on any ordinary machine sander.

The discs are then ready to assemble. A clutch of this sort should cost complete—in any reasonable quantities—not exceeding three pounds and wear for years in any reasonable service. This cork inserted surface, may, of course, be used in cones or brakes if desired, with equally good results.

Accessories.

Power type pumps are being fitted to many of the higher and medium-priced cars—the demountable rims may be said to be becoming almost the regular equipment of all but the very cheapest machines.

There is a marked tendency toward the use of roller bearings, of which several makes are well known in the States. Used in connection with gearing, they give a quieter effect—or at least some makes of them do—and the load carrying capacity is considerably in excess of the ball bearing of equal diameters. The roller bearing bids fair to replace the ball for all heavy truck work.

The extreme quietness is just as much in demand with us as with you. This demand is being carried to foolish extremes in some instances, but its net result is undoubtedly good for the industry. Valve mechanisms are being enclosed, and better gear work and fitting done. The worm final drive is receiving a good deal of attention, although not nearly so much as yet as here in England. Sheet metal—both aluminium and steel—have come to be the universal, or nearly universal, practice for all panels. The quantity production of bodies has brought about some wonderful press and die work for the forming of entire front and rear seat panels from one sheet of metal. One concern, the Pierce, is notable for its bodies of cast aluminium seat back and panel construction.

Demountable wheels—wood—have been for several years fitted as regular equipment by our American makers, but it does not seem probable that this practice will spread.

Wire wheels have lost favour, and in 1903 and 1904, when the public made up its mind in favour of wooden spokes, American manufacturers were forced to scrap thousands of sets of wire equipment because the turn of the tide came so suddenly. The objection to wire is one of appearance, and I doubt whether the American buyer can be soon brought to favour the lighter and stronger type. Educational work is wonderfully expensive, and American makers are not apt to push it.

In one great thing our American engineers have been particularly fortunate. I refer to the number, the variety, and the character of the motor car contests which have been held in every part of the country and under every possible condition of road and climate. There can be no better school of design than the racing camp during the two weeks' practice period just before some big two or three hundred mile road race. But here, as in every other line, a definite system must be followed if the best returns in an engineering as well as a publicity way are to be had.

Many American makers have spent considerable sums annually in the maintenance of contest crews. The cars entered in the racing and endurance events, have for several years been generally stock cars. Even the special cars entered in the "open" or so-called "non-stop" events have been made up largely from standard parts used in regular production.

The advantages accruing to the engineer from such practice are too obvious to need emphasis. A not unusual method of procedure is to make the chief engineer, either directly in charge of the contest work or at least in close working contact with contest crew management. A crew is more than frequently made up of a business manager and from two to three cars with drivers and mechanics. If the engineer is wise, he will have placed upon his desk an almost daily detailed report of the performance and troubles of these contest cars. I do not know of a more searching trial of steering connections, wheels, motors, and in short, of the entire driving mechanism than is the three-hundred mile road race or the two-thousand mile endurance run, covering every possible road condition.

When these words were written, I had little idea how sadly this statement was to be emphasised. A telegram received just before sailing brought me the news of the death of the honoured chairman of our National Contest Board—brought about by the breakage of a steering connection a thousand miles out on a touring contest which would correspond in many ways to your Prince Henry event.

The importance of properly governed contests, and the full recognition of the dangers and abuses likely to follow a laxity in contest government, have brought about a unity of action upon the part of our manufacturers entirely independent of the views of the individual concern.

I don't imagine that you in England are troubled with professional promoters or outlaw meets, but with us they have been a source of much that has been disgraceful—and all, of course, to the end of separating the spectator from his dollar and giving a minimum of value in real racing in return. Really they have caused more of a row and more of a waste of printer's ink during the past couple of years than have all the Indians left in the North American continent.

Three years ago the makers put into motion strong machinery for the regulation of all things connected with motor car sport. We now have a Manufacturers' Contest Association of a hundred members. From this membership each year are appointed by the President the representatives of twenty-five concerns to serve as a general rules committee. In this body are discussed all rules having to do with contest legislation, the will of the majority being passed on to the National Contest Board for incorporation in the annual Rule Book of the American Automobile Association.

The inter-relation of all the various organisations, both trade and amateur, are clearly defined by executed contract, and from the fact that order has been brought out of the absolute chaos of a few years ago, one may judge that we have struck near to the solution of the trade relation to the sport.

In any event, the main fact is that the encouragement of the stock car contest has been of unusual value to the engineer, and hence to the progress of the industry in the States. Not only has the American engineer been constantly able to judge of the performance of standard cars under severe service conditions, but he has been no less the gainer in that he has had his whole attention riveted upon a "bread and butter" model, as we say, rather than upon some monster racing car. All of you know how the production of a racing car disrupts a factory organisation.

Enough for the racing subject, upon the various angles of which I might talk to you for the rest of the evening. A sport in which there is spent annually in America, directly and indirectly, some

two millions of dollars is apt to present its interesting phases.

And just now we are hearing much over here of another thing—the American invasion of the English market. There seems to be a good deal of worry, at least upon the part of the press.

I do not quite see why England fears an invasion. The invasion of the cheap car I believe it is called. If the cheap car has a field in England, the probabilities are that it is a field for which you have not yourselves catered. It seems to me that you may meet the issue either by building low-priced cars yourselves for this demand, or by keeping out of this field and sticking to the class of product to which your preferences incline.

We have been through the same thing in America exactly. The production of the five hundred dollar car by Ford was croaked to be the death knell of all high-price-getting. There have been several "death knells" since Ford, and Ford himself is still at it to the tune of seventy-five thousand cars for 1912.

And yet all these "death knells" have been the greatest of missionaries. You know that there is in the make-up of the average man very little of the anti-climax. A better or a bigger car is the dream of the owner of every "road-louse." I do not use this term in disparagement of the small car. Your larger car business will be better when your roads are alive with the little ones. Our records compiled show that within the United States there are thousands of families whose annual income would warrant the purchase of a four or five thousand dollar car. There are some hundreds of thousands of families amply able through income to purchase the two thousand dollar car. But there are actually millions of families amply able to meet their obligations in the purchase of the one thousand dollar car. Not so long ago we heard the cry that the ownership of a motor car must be an indication either of great wealth or of the reckless spendthrift. The automobile could never be the property of the ordinary farmer, the tradesman, or the village plumber. But wonder of wonders—when quantity manufacture had brought the price within the reach of the quantity market—the market was there to take the goods, to pay for the cars in cash, and to put them into a service to bring quick returns both social and financial. Look carefully at the man behind the wheel of the next low-priced American car you meet on the road. The chances are you will find him no millionaire. There are six million farm owners alone in the States, and many thousands less than one million cars in service even yet.

Now as we figure it, the maker may choose the field or class to which he will cater. Low price means necessarily large production and a tremendous financial outlay in equipment, to say nothing of brain, organisation, and nerve. I have heard it estimated that you cannot hope to combat the American invasion. I believe rather that the Englishman prefers to think conservatively and in small numbers. In this connection I would like to say two things. First, that this same low-priced car invasion has conquered America with results only of good to the medium or higher-priced product of merit and, second that, with her high-class labour, her great factories, her wonderful resources, her centuries of manufacturing experience, and her world-wide market through her provinces and through her proximity to the Continent, it would take much to convince me that England were not potentially qualified to hold her own in manufacturing competition with any nation on earth.

And one last thing—there is a great need in the mental equipment of every motor car engineer for that intangible quality known as "horse sense." There is no more dangerous element in any industry than the purely theoretical engineer—one always seeking the ideal to the utter exclusion of the very practical and very commercial present. Some of the greatest financial failures of our American motor industry can be charged directly to a lack of the commercial sense (if the term please you) upon the part of the engineer.

Above all the motor car maker is in business for something more than his health. He is in business to make money for himself, for his shareholders, and, indirectly, for the men he employs. He has an obligation also to those who become the owners of his cars, for repairs and service must be supplied year after year.

Capital is for the most part ignorant in those matters pertaining to the detail of design, and it is unfortunate that the theoretical man is too frequently very well qualified to impress as to the soundness of his advice. The engineer is the originating element in any industry, and if his work as a foundation is faulty, the combined efforts of all the other departments of the business cannot be made to redeem his failure. More than five hundred millions of dollars are staked annually in this industry upon the work of the engineer. If in the other great business activities concerted engineering work has been found necessary, then certainly must we, as motor car engineers, work hand in hand in the solution of all those problems which effect the future of the industry.

EDITORIAL NOTES AND DISCUSSION.

The following is a summary of the interesting points raised during the ensuing discussion.

Mr. Legros, in opening, remarked that there was just one figure that Mr. Coffin gave in his paper which attracted his attention, that was the figure for the Ford output. In ten years, or rather it was more than ten years, since the passing of the Act, they had registered in London practically exactly the same number of cars that Ford will turn out next year.

Mr. Staner desired to know whether, under the abnormal road conditions in the States, any experiments had been made with equalizing levers similar to those used in railway work, because there had been one or two inconclusive trials of such apparatus here. Also whether with the increased stroke an increased piston speed was being used or tried, or had the gear ratio been altered to suit the new conditions? As to the cork inserts, mentioned by Mr. Coffin, what particular advantage resulted, since it would seem but a complication of the multi disc design? Over here we have no trouble with metal clutches. They are cleaned out with paraffin once in three months, then oiled, and that is all for some 15,000 miles. What particular advantage ensued from cork inserts?

One other point of interest which Mr. Coffin did not mention was the use of roller bearings. The type which interested us most was that in which thrust and load are taken on the same roller. Such a bearing would seem peculiarly suitable for steering boxes, pivots and front wheels.

Unfortunately, limitations of time prevented Mr. Coffin replying to Mr. Staner's questions, which is the more to be regretted because they were of considerable interest, and should have lead to some most interesting comparisons.

Mr. Berriman touched on the interest attached to the increased stroke engine in America by reason of the fact that he had usually found such engines were suitable for hard pulling at low speeds rather than developing great power, as they did here. As a result, the American long stroke engines picked up on top with great rapidity, a quality which was desirable considering the usual type of driver in that country. Mr. Berriman also desired information on the gearbox axle design, as such a combination was undoubtedly quieter, an effect which might be obtained by underloading the gears very considerably.

Mr. Davis, in replying to this question about gear construction and noise, made a remark which is of some interest, namely, that the tendency in the States is to produce gears having the standard tooth instead of what is known as the stub tooth, and that it had been found easier to produce quiet gears with the longer tooth and very accurate forming.

Mr. Davis also asked for information on worm gearing, a subject of great interest to the American industry. Mr. Lanchester, in replying, went through the early history of such gearing, mentioning some amusing examples of the difficulties met with. Turning to the gear itself, he suggested that the angle for maximum efficiency is 45 deg. minus half the angle of friction. With this or even 40 or 35 degrees approximately maximum efficiency can be obtained. In criticising the different forms of worm, namely, the parallel and the hollow types, interesting mention was made of the efficiency curves peculiar to both, showing the falling off of the curve belonging to the parallel worm at an earlier point than that given by the hollow worm.

Colonel Crompton asked for information on the average annual mileage of an American car under the very bad road conditions there prevalent. Remarking that eight or nine hours of speed on the said road was too much for anybody's endurance, he had an idea that the manner in which American cars stood to their work must be due to the low average mileage. In Europe there was no limit to the mileage of an enthusiastic motorist, road conditions being comparatively ideal, and European cars had to be designed to stand prolonged running and high mileage as a consequence. This point depended on the precision of workmanship and high-class fitting. The average mileage was put by Col. Crompton as no less than the remarkably high figure of 12,000 miles.

In reply, Mr. Tomlinson remarked that an average American mileage might be 5,000 miles.

Mr. Coffin's general reply contained an interesting account of the work accomplished by the Standardisation Committee of the Society of Automobile Engineers of America, and he put considerable stress on the immense utility such a plan must inevitably contain; further, to the discussion on mileage, he raised the average of cars in his country to 6,000 miles.

In touching on society meetings, Mr. Coffin gave an account of the initial stages of the S.A.E. and spoke at some length on the wisdom of the adopted policy concerning the open mouth, stating that in every works he knew, all but the experimental department could be inspected by anyone.

WORKS EQUIPMENT.—We have received a catalogue of screw cutting and turning lathes issued by F. M. Frye and Co. All sizes of lathes are dealt with, and the list in question states thoroughly the price of all the spares necessary for the type of lathe handled by the firm. A speciality is made of the new six inch model suitable for small accessories or other minor motor components.

LUBRICANTS.—A very interesting pamphlet has been issued of the John Dixon Crucible Co., which takes the form of a condensed account of the test taken by Professor Goss on the lubrication of ball bearings by flaked graphite. A diagram of the testing machine used is shown, together with curves with various lubrications, including kerosene, lard oil and vaseline. The booklet is of interest to all who are responsible for the running and up-keep of line shafting.

NEW INDUSTRIAL PROCESSES FOR CASE-HARDENING OF STEEL.

A paper by Dr. F. Giolitti before the Iron and Steel Institute.

As the result of the experiments and from the theoretical considerations elaborated by the author in his several reports, and chiefly in the more recent ones, several fundamental facts, besides many of secondary importance, have been proved with certainty. It has, moreover, been specially demonstrated that these facts are in perfect agreement with the work of previous investigators, although they were unable to arrive at exact conclusions on account of the lack of the necessary theoretical knowledge which has become available to science only within the last few years.

These fundamental facts, the discussion and proof of which the reader will find in the reports contained in the bibliography, may be briefly enumerated as follows :—

1. Where case-hardening is carried out with solid cementing agents having a carbon base, the carburising effect of the free carbon on the iron (the materials being in simple contact and without the intervention of gaseous carbon compounds), is exceedingly weak, and in any case is entirely negligible in industrial practice. This assertion is fully confirmed by the more recent investigations of Guillet, Griffith, Weyl, and Charpy. The last-named experimenter even comes to the conclusion, already dealt with by others, that the direct effect of carbon in cementation with solid agents is absolutely nil.

2. Where case-hardening is effected with the solid case-hardening agents ordinarily employed in the industry, the specific effect of the nitrogen is very weak, as admitted, and variously explained by many investigators. Only with the cementing agents containing a high proportion of the cyanogen compounds (alkaline cyanides, ferrocyanides, etc.), does the direct action of volatile nitrogen compounds have any marked effect.

3. As compared with the cementing effect of the solid agents ordinarily employed in the industry, the specific direct carburising effect of carbon monoxide preponderates enormously over every other carburising effect.

4. Pure carbon monoxide carburises iron at all temperatures within the range (700° to 1,300° C.), at which the process of case-hardening can be performed by means of any medium whatsoever. Moreover, the rate of case-hardening (by which is understood the depth of carburisation which can be obtained at a given time) when working under suitable conditions is greatest when carbon monoxide, or a mixture in which the carbon monoxide can efficaciously exercise its specific carburising effect, is used as the agent.

5. This specific carburising effect exerted by the carbon monoxide on the iron at high temperature is due to a series of chemical reactions, the course and state of equilibrium of which have been actually observed with precision. Moreover, the conditions of equilibrium of the systems in which these reactions take place are in general comprised within the ranges of temperature and pressures ordinarily employed in practice. It is therefore possible to obtain with certainty a predetermined result using case-hardening agents whose actviity is due, if not exclusively at least very largely, to the specific carburising action of the carbon monoxide. More particularly it is possible to obtain with such agents carburised zones in which the concentration of the carbon does not exceed a predetermined maximum limit and varies in a well-defined degree towards the inside of the carburised zone. Such definite results, variable at will within sufficiently wide limits, are obtained by varying, in accordance with fixed rules, the temperature at which the case-hardening is performed, the pressure of the carburising gas, and the amount of carbon monoxide which, in a given time, comes in contact with the unit surface of the steel to be carburised.

6. The results obtained by the use of carbon monoxide as an agent vary regularly, other conditions being equal, as the chemical composition of the steel to be carburised is varied.

7. It is possible to vary regularly and within fairly wide limits the characteristics of the desired product by subjecting the steel to the action of substances (along with carbon monoxide) which are capable of modifying the conditions of equilibrium of the chemical systems under which the reactions due to the specific carburising effect of the carbon monoxide are completed. Such substances may be gases, such as hydrocarbons or nitrogen, or they may be solids such as carbon in various forms, and their actions can proceed simultaneously with that of the carbon monoxide throughout the whole cementation period or during a portion only of it.

8. In particular, by means of the agents, the activity of which is due to the specific carburising effect of carbon monoxide, it is possible to obtain with ease and certainty—whatever kind of steel is being operated upon—soft case-hardenings and graduated case-hardenings; that is to say, carburised zones in which the concentration of the carbon, without being excessive in the outer layers, diminishes slowly and with regularity in the succeeding deeper layers. This is the essential condition for the avoidance of the dangerous phenomena of brittleness and peeling, which defects manifest themselves so frequently in steel pieces case-hardened by the processes ordinarily used in the industry.

9. The chemical reactions produced by agents in which cyanogen is the active element, are at present but imperfectly understood, particularly as regards their condition of equilibrium, upon which depends the concentration of carbon in the carburised zones. It is, however, certain in the conditions under which case-hardening should be performed in practice that the conditions of equilibrium just now alluded to should closely correspond to the strength of the very high concentration of carbon passing into solution in the iron. Thus it happens that the cyanides, ferrocyanides, and other derivatives of cyanogen, if used alone as case-hardening agents, always give rise to too rapid (energetic) a case-hardening. That is, carburised zones are produced in which the concentration of the carbon is excessively high in the outer layer up to a certain depth and is then suddenly lowered in the succeeding layers. Zones of that type, the formation of which has been fully studied and their cause explained by the author, produce brittleness and peeling.

10. Further, gaseous or volatile hydrocarbons, when used alone as agents, also give rise to too rapid case-hardening, the causes being identical with those referred to in the case of cyanogen and its compounds.

The author repeats that the experimental proofs and theoretical explanations of all the foregoing statements are fully dealt with in the memoirs, the list of which is given at the end of the paper.

In the light of the facts recited in the foregoing, the great advantage of the use of agents, the activity of which is due, if not exclusively, at least principally, to the specific carburising action of carbon monoxide, is clear. In order to obtain the best results with such an agent, the author has demonstrated in his publications previously quoted that it is necessary to satisfy the following fundamental conditions :—

1. The chemical composition of the agent should be absolutely definite, and should be accurately known.

2. The compounds should be as simple as possible.

3. The reactions which take place during the case-hardening process between the various constituents of the agent and those of the steel should be simple, and should proceed rapidly—under the conditions most easily obtaining in practice—to a well-defined state of equilibrium corresponding to definite concentrations of carbon in the carburised zones.

The author in his previous work has demonstrated from the theoretical point of view that the agent which best satisfies these conditions would be pure carbon monoxide, except that by its use the concentrations of carbon in the carburised zones, which correspond to the conditions of equilibrium, are in general too low, when working within the ranges of temperature and pressures ordinarily maintained in practice, and when the metal subjected to case-hardening is an ordinary mild carbon steel or a steel containing a low percentage of nickel or chromium.

On the other hand, that inconvenience can be avoided, and the three conditions indicated above can be sufficiently well realised by the use of case-hardening agents, in which along with the carbon monoxide small quantities of hydrocarbons of known composition, or solid carbon in a properly divided state, are allowed to act either throughout the whole period or during a portion only of it.

Further, of these two classes of agents, those of the second class, based on the simultaneous use of free carbon and of carbon monoxide, are the most suitable for the majority of ordinary technical applications, since by their use the operation can be performed with the maximum degree of simplicity and the certainty of obtaining perfect results.

Below are enumerated the principal technical advantages which the author has shown to be obtainable by the employment of case-hardening agents, which satisfy the three essential conditions previously mentioned and, in particular, of the "mixed" agent based on the simple simultaneous action of carbon and carbon monoxide. The following are the principal advantages :—

1. The great speed of penetration of the carburised zone. That fact, if, indeed, it does not constitute the chief value of any given case-hardening process, as many manufacturers still believe, is certainly most advantageous on economical and technical grounds too numerous to mention.

2. Great uniformity in the distribution of carbon in the carburised zones, on account of the fact, as already stated, that the peeling of case-hardened and tempered pieces is reduced to a minimum.

3. The possibility of regulating—either by diluting the carbon monoxide with nitrogen, or by limiting the contact of the solid carbon with the surface of the steel—or by suitably varying the temperature during the case-hardening process, the concentration of the carbon in the carburised zone so as to maintain it within the most suitable limits for conferring the maximum hardness combined with minimum brittleness. The extent of carburisation must, of course, vary according to the composition of the steel subjected to cementation.

4. The possibility of establishing with certainty from the start the necessary conditions for obtaining a predetermined result which may be obtained with great accuracy.

5. Continuous use of the same carburising materials (solid carbon and carbon monoxide), which do not become attenuated, but may be used up to their last residue. This also permits of carburising to any depth without the necessity of renewing the agent during the operation.

6. Absolute security against the introduction into the steel of any foreign substance apart from the carbon. This is an advantage of the highest importance in most cases, and cannot be realised in the case of most of the case-hardening powders habitually used, consisting of organic nitrogenous substances, such as alkaline cyanides, or ferrocyanides.

7. The ease with which the surface of the case-hardened pieces is preserved without alteration, thus obviating the necessity of any subsequent dressing of the case-hardened pieces.

8. The deformation and change of volume which the steel pieces may undergo during case-hardening are reduced to a minimum. In any case it is possible to determine from the start the extent of such volume changes as may occur.

For a full account of the experiments and their results the author would refer readers to his previous reports on his research work on this subject.

Besides the advantages enumerated above, which refer particularly to the quality of the product, the use of the mixed agent having a carbon monoxide base offers other practical benefits, consisting of the ability to modify very advantageously the method of operation. In one of the reports referred to it is shown by the author how the method may be modified even without discarding the ordinary horizontal muffle generally used for case-hardening. The same advantages, to a greater extent, may be more easily secured by using the mixed agent in a furnace the form of which differs radically from that of the ordinary case-hardening furnace, and by following a special method of operation.

The particular object of the present paper* is to present a short description of this furnace, of the accessory apparatus, and of the method of its operation which will enable the specific advantages of each to be readily understood. Nevertheless, the author will refrain from describing at great length either the special conditions which the substances used as case-hardening agents must fulfil (granular carbon and carburising gas), or the qualities of the products obtained, these having been made sufficiently public in the author's previous publications, in which the characteristics and conditions are fully discussed and illustrated.

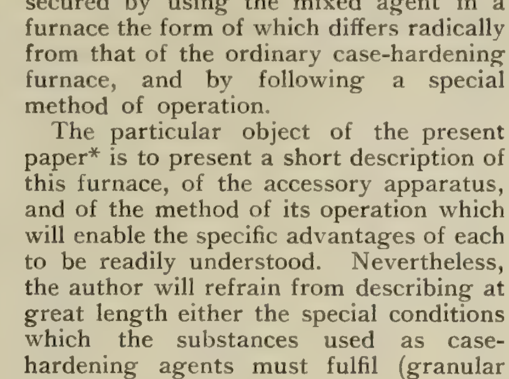


Fig. I.

The type of furnace which experience has proved to be best adapted to the case-hardening of small and medium-sized objects with the mixed case-hardening agent above referred to, consists of cylindrical muffles arranged vertically and heated by producer gas. It is a double-muffle furnace provided with regenerators, of the kind constructed by Messrs. C. M. Stein, of Paris, for the Sampierdarena Engineering Works of Messrs. G. Ansaldo, Armstrong and Co. With regard to the construction of the furnace, Fig. I. conveys a sufficient idea of its principal parts to render unnecessary any detail description. It may, nevertheless, be of interest to give some information concerning those parts more directly connected with the carrying out of the case-hardening operations, and to describe briefly the procedure. The parts of the apparatus particularly applied to the case-hardening process are represented in Fig. I., which shows a section through the centre of the vertical muffles of the furnace with the accessory apparatus completely mounted. For the sake of clearness, the dimensions of the several parts are omitted on the drawing, but a sufficiently rough idea of the dimensions can be obtained if necessary from the proportions of the parts. Inside each of the muffles of refractory material is placed a cylindrical retort of mild steel, A, the external diameter of which is about 10 to 20 millimetres less than the internal diameter of the muffle. Mannesmann seamless tubes serve very well for making the retorts, as they are easily obtainable in stock sizes up to 35 centimetres diameter. For retorts of larger diameter, as in the case of the furnace here represented, tubes welded by the autogenous process are used. The retort is supported on a bottom flange attached to a frame fixed to the brickwork of the furnace. The arrangement which serves to hold the retort, whether it is being supported by the bottom flange or by the upper flange, H, intended to receive the cover, is such as to enable one retort to be replaced by another in a few minutes. To the bottom flange is attached a cast-iron funnel B, closed at the bottom by a non-return valve Q.

The steel nozzle C, admitting the carburising gas, passes through an opening in the side of the funnel, provided with a good-sized boss. The gas, which, as has been previously stated, may consist of carbon monoxide or dioxide, in a pure state or mixed with other gases, is introduced through the screwed tube D, and through the nozzle C. This latter is connected to the hollow cast-steel dish E, intended to support the pieces to be case-hardened by means of the disc F, consisting of a steel plate covered with refractory material.

The carbon monoxide, after passing through the hole bored axially in the nozzle C, is delivered into a small distributing device placed inside the dish E, from which it passes to the case-hardening chamber G, through a number of holes provided in the disc F. During case-hardening the dish E is supported upon a series of projections on the same lower flange to which the funnel B is attached. This ring has thus to sustain the entire weight of the dish E, the disc F, and any articles to be case-hardened which may be resting directly or indirectly on the disc. The section shows exactly the position of the various parts during case-hardening.

Considering now the method of operating, beginning with the charging of the retort, the mode of this varies very much according to the dimensions and shape of the articles to be case-hardened. The author will therefore confine himself to giving a few special examples.

In the case of spur wheels, the maximum diameter of which might be 100 to 150 millimetres less than the internal diameter of the retort, the first thing to be done is to unscrew the tube D; then by means of the plunger L, actuated by the hydraulic cylinder I, the whole of the

*Both the apparatus and the process about to be described are patented in many countries.

parts C, E, and F are lifted together, the blunt conical end of the plunger M engaging with these as it rises. The plunger LM is cooled internally with water circulation, so as to avoid the heating of it in case it should accidentally remain raised inside the retort for too long a period. When the piston L has reached the end of its stroke, the upper surface of the disc F being now only about 30 centimetres below the cover of the retort, the cover is removed, and the wheels to be case-hardened are piled horizontally one on top of the other upon the disc F. As the wheels are inserted one by one, the apparatus carrying the weighted portions C, E, and F gradually sinks, and the charge is complete when the apparatus has reached the lowest position indicated in the figure. In that position the last wheel charged should be at least 30 centimetres below the upper edge of the retort.

Generally, the articles to be case-hardened are charged hot at a temperature of about 800° to 900° C., this being a matter of prime importance, as it affects most favourably both the quality of the product and the economy and regularity of the operation.

For the preliminary heating of the pieces a small coal-fired muffle will serve, since the uniformity of temperature which can be obtained in such a furnace is quite sufficient for the purpose. If, however, the case-hardening furnace is not being worked to its fullest capacity, one of the vertical muffles may be used for the preliminary heating of the articles, while the case-hardening of the charge is proceeding in the other. In that case the best method of working the furnace is to use muffles alternately, one for case-hardening, and the other for pre-heating the pieces.

As soon as the articles to be case-hardened have been inserted as described, the cover is placed on the retort, and the bottom discharge pipe N, leading from the iron box O, is passed through the central opening in the cover. This box is full of the granular carbon still hot (generally at about 900° C.), which was discharged from the retort after use during the last case-hardening operation just concluded. The box is easily swung above the muffles, being suspended by a tackle from a swinging arm. The outer walls of the box, notwithstanding that its contents of granular carbon are at a temperature of nearly 1,000° C., do not become heated above 200° to 250° C. during the time required for the loading and unloading of the retort, even when working under the most unfavourable conditions. This is due to the small thermic capacity and the very low conductivity of the mass of granular carbon which, although at a high temperature, can only impart to the walls of the box containing it a small proportion of its heat in a unit of time. The heat is quickly dissipated simply by the cooling action of the air currents passing round it.

The box O having been adjusted as indicated, the butterfly valve which closes the lower end of the opening N is gradually opened, and in a few seconds the whole of the free space in the retort surrounding the pieces is filled with glowing granular carbon. The author has previously pointed out that the hot granular carbon employed by him for case-hardening with his mixed medium constitutes a mass which, particularly at high temperatures, is endowed with a mobility comparable to that of a liquid. On this account it is able to penetrate simply by its own weight into all crevices among the articles to be case-hardened, and between these and the walls of the retort. The filling up may be further assisted by means of iron rods introduced through holes in the cover and manipulated by the operator who stands on the furnace.

When the granular carbon has been filled in up to a level of 2 or 3 centimetres below the upper edge of the retort, the butterfly valve of the tube N is closed, the box O is lifted, the central opening of the cover is closed, the plunger L is lowered right down, and the tube D is screwed into the nozzle C, through which the carbon monoxide or dioxide, pure or mixed with air, is admitted gradually to the distributor E and thence into the retort. In order to regulate the current of carbon monoxide according to the method indicated in the report already referred to, an ordinary gas meter of the dry type is used. The case-hardening then proceeds without further trouble, the only thing that is necessary to be watched and regulated being the current of gas and the temperature.

For the observation of the temperature, an operation which, with the type of furnace described and when adjusted properly, does not often require to be made, an ordinary thermo-electric pyrometer is used by passing it through any one of the holes in the cover from time to time and inserting it into various parts of the mass of granular carbon. The temperature should be maintained quite constantly at a fixed point between 900° and 1,100° C., chosen in accordance with the rules laid down by the author in the reports previously referred to.

The gas issuing from the retort through the pipe fixed in the cover can be collected in a gasometer, and after suitable regeneration it will serve for further heats. The necessary length of the heating period can be determined with the greatest precision, and is dependent on the result which it is required to obtain. When the heating has proceeded long enough, the current of carbon monoxide is shut off, the tube D is removed, and the discharge of the retort is undertaken.

Where results of a very special nature are not aimed at, the discharging operation begins by withdrawing the whole of the granular carbon at once, and removing it through the non-return valve Q into the box O, now placed underneath. Where, however, exceptionally graduated case-hardening is required, the specific action of the carbon monoxide is isolated during the last phase of the case-hardening from the direct action of the granular carbon. In that case the box O is placed so that the neck of the funnel B is inserted into the open end of the tube P, the box for this purpose being placed in the chamber below the muffle, where it can be manœuvred on a small bogie.

The non-return valve is then opened gradually and a quantity of the granulated carbon is drawn out from the retort.* Only those parts of the case-hardened pieces are uncovered in which a well-graduated case-hardening is required. Sufficient solid carbon is retained in the retort to ensure the desired chemical equilibrium of the gas. The remainder of the carbon is removed from the retort after the second phase of the case-hardening with isolated carbon monoxide has proceeded long enough to attain the desired results.

In either case, as soon as the whole of the granular carbon has been evacuated from the retort, the cover of the latter is lifted. In performing that operation the operator must be careful to remain at some distance from the upper open end, because the air which enters forms, with the carbon monoxide contained in it, a mixture which explodes suddenly at the high temperature of the retort. The explosion, which in any case is light and not at all dangerous, may be avoided by rapidly scouring the muffle with a little carbon dioxide after the removal of all the solid carbon. Care must be exercised, however, to perform this operation as quickly as possible, in order to prevent the carbon dioxide causing a superficial decarburisation of the case-hardened pieces.

When the pieces rest direct on the disc F, as in the present supposed case, they may be lifted by means of the appliances C. E. F, up to the open top of the muffle without first discharging all the granular carbon from the retort. In that manner the light explosion just referred to is avoided, but the picking up of the pieces for quenching is made more difficult, and a considerable quantity of the solid carbon is apt to get burnt in the retort.

At all events, whether the carbon is removed from the retort or not, as soon as the cover R is removed the plunger L is raised until its conical head beds itself in the recess provided for it in the part C. The hydraulic ram then continues to move slowly, gently lifting the weight-supports, C, E, F, with the case-hardened pieces resting on them. It only remains then for the operator standing above the furnace near the muffle to pick up the case-hardened pieces one by one as they come within reach and quench them in the tank, also placed above the furnace (in case double quenching is desired), or bury them in hot ashes. When all the pieces have been removed, the furnace is ready for a fresh charge.

As already mentioned, the method of loading the pieces to be case-hardened into the retort varies greatly according to their shape or size. Pieces of cylindrical shape can be piled on one another as in the foregoing example. If the pieces are of small diameter, such as pinions, cutters, cups and rings for ball bearings, several columns can be built up for one heat, but in that case it is desirable to keep the pieces in position by threading them on iron rods, assuming that the external surfaces only require to be case-hardened. For cylindrical pieces to be case-hardened internally, such as rings for ball bearings, the rods are placed outside the columns. Pieces of great length or of irregular shape have also to be suitably supported in the retort. Finally, very small pieces are placed in cages formed of iron framework held

*The carbon contained in the retort falls out from below, passing first through a series of holes in the bottom flange upon which the apparatus C. E. F rest. It was, however, not possible to show all these holes in the accompanying sectional view of the furnace.

together with iron wire-netting. Such pieces can also be laid free on successive layers of granular carbon.

As regards the necessary time for the purpose of the single operations above indicated, it is impossible to give it exactly for every case, for it must naturally depend on the shape, size, and number of pieces forming the charge. A few definite data may, however, be quoted here, giving the approximate time required for the several operations to be performed in the special case which has been under consideration in the preceding pages, where the charge was assumed to consist of about thirty cylindrical pieces, or spur wheels of such a size to form a single column in the retort. In that case the times are as follows:—

Table I.

No. of the Range of Single Layers Analysed.	Distance from the Surface of the Central Portion of the Layer Analysed, mm.	Amount of Carbon in the Single Layers.		
		First Determination per Cent.	Second Determination, per Cent.	Mean per Cent.
1	0·5	1·16	1·18	1 17
5	2·5	0 80	0 82	0 81
10	5.0	0·33	0·35	0·34

(*a*) Placing the pieces on the supporting disc; from 1 to 2 minutes.

(It is clear that this operation is longer according as to whether the pieces are smaller in size and larger in number; but even with quite small pieces packed in alternate layers of carbon, this being the least favourable condition, the time does not generally exceed 5 minutes.)

(*b*) Closing the lid of the retort and placing in position the box containing the granular carbon; about 1 minute.

Table II.

No of the Range of Single Layers Analysed.	Distance from the Surface of the Central Portion of the Layer Analysed, mm.	Amount of Carbon in the Single Layers.		
		First Determination per Cent.	Second Determination, per Cent.	Mean per Cent.
1	0·5	0·84	0·86	0 85
5	2·5	0·76	0·80	0·78
10	5.0	0·46	0·46	0·46
15	7·5	0·23	0·27	0·25

(*c*) Filling up the retort completely with granular carbon; from 1½ to 2 minutes.

(This operation is also prolonged when the pieces are very small and there is a large number. Generally in the least favourable case the time does not exceed 4 minutes.)

Table III.

No. of the Range of Single Layers Analysed.	Distance from the Surface of the Central Portion of the Layer Analysed, mm.	Amount of Carbon in the Single Layers.		
		First Determination per Cent.	Second Determination, per Cent.	Mean per Cent.
1	0·5	0·69	0·71	0 70
5	2·5	0·66	0 68	0 67
10	5.0	0·52	0·54	0·53
15	7·5	0·38	0·40	0·39

(*d*) Lowering the plunger of the hydraulic ram, adjusting the pipe for admission of carbon monoxide, removal of the box containing granular carbon, and complete closing of the cover of the retort; altogether 1 minute.

As will be seen in the case under consideration, the entire period for the operation of charging the retort may vary from 4½ to 6 minutes. In the most favourable cases it may nevertheless be increased to 10 or 12 minutes.

The following are the approximate times occupied by the discharging operations:—

(*a*) Removal of the pipe admitting carbon monoxide, placing the box for granulated carbon below the lower neck of the retort, and opening the non-return valve; 1 to 1½ minutes.

(*b*) Evacuation of the whole of the carbon from the retort into the receptacle and closing the valve; 1½ to 2 minutes.

(*c*) Raising the supporting disc, and unloading of the case-hardened pieces; 1 to 2 minutes.

(If the case-hardened pieces are very small and numerous, and particularly if they are laid out on layers of carbon, this last operation may be lengthened to 10 minutes. The time required for the complete discharge of the retort is therefore 3½ to 5½ minutes, though in the most unfavourable cases it may extend to 13 or 14 minutes.)

Further, in the case of the process described, and in almost all cases occurring in the practical production of case-hardened pieces for machine construction in which the carburised zone is seldom required to exceed 2 millimetres in thickness, the period of case-hardening never exceeds two hours, and may even be reduced to one hour, the time being reckoned from the moment of completing the charging operation to that at which discharging begins. It therefore includes the time necessary for bringing the temperature up to working point, which requires not more than 10 minutes, owing to the fact that both the solid carbon and the pieces are already hot when charged.

The foregoing particulars show that in practice, under ordinary working conditions, the time for a complete heat is less than 2½ hours, and since the weight of a charge, which retorts of the size shown in the accompanying sketch can conveniently hold, varies from a minimum of 100 kilogrammes in exceptionally unfavourable cases, to a maximum of over 500 kilogrammes, it will be seen that the productive capacity of each retort ranges from a minimum of 1 ton to a maximum of 5 tons of case-hardened steel pieces per 24 hours.

From the preceding statements, and leaving aside all considerations as to superiority of the product, which are due to the principle on which the process is based rather than to the method of its application, it is clear that the furnace

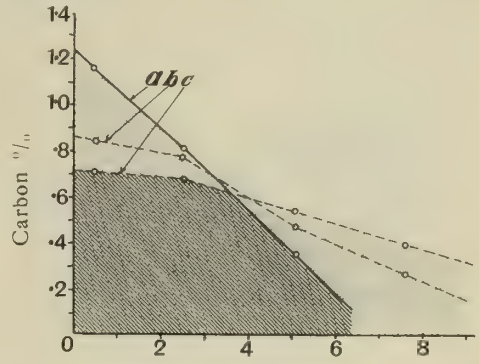


Fig. II.

with vertical muffles has distinct advantages as compared with horizontal muffles.

The advantages consist mainly in the greater speed of working for a given charge, greater simplicity of operating, and in the increased uniformity of the treatment of all the pieces forming the charge in the muffle. The uniformity is not limited to the temperatures of starting or of working, but, what is more important, it applies also to the distribution of the carburising gas (carbon monoxide pure or mixed with other gases). Besides, in the vertical furnaces arranged as described, the clearance spaces left in the retort are reduced to a minimum, whereas in the horizontal muffles the clearance spaces have a considerable effect, especially at the head of the retort, where they cause appreciable losses of heat and of carburising material, thus lowering the productive capacity. For these reasons, the vertical type of furnace is superior in every respect, especially as regards economy of production, as already stated.

Before concluding, it may be useful to discuss a little more fully some of the more interesting results which are obtainable with ease and regularity in the new type of furnace when using the mixed case-hardening agent having carbon monoxide as its base. Referring first to the special feature of the distribution of the carbon in the carburised zones, which can be graduated by isolating the action of the carbon monoxide during a part of the carburising period, it has already been

shown how that result can easily be obtained, when using the vertical furnace, by removing from the case-hardening chamber at the proper moment that portion of granular carbon which, during the first part of the operation, has been in direct contact with the surface of the pieces, leaving in the lower part of the retort a layer of carbon of sufficient thickness to maintain the chemical equilibrium of the gaseous mixture passing over it.

The effects of the isolated action of the carbon monoxide on the carburised zones obtained at the commencement of the operation, using the mixed agent, has been dealt with elsewhere. It may, however, be of interest to consider the effects to which that method of procedure gives rise. The following data refer to case-hardenings of medium depth (from 5 to 10 millimetres) in which the application of the process in question presents the greatest technical interest, particularly in the case of very deep case-hardenings. In the accompanying tables are collated the results of estimations of the carbon

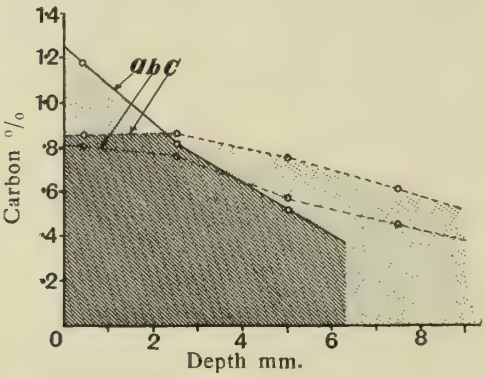


Fig. III.

in successive layers of a thickness of 0.5 millimetre, obtained by case-hardening with the mixed agent, followed by exposure to the action of carbon monoxide isolated in the manner previously described. The same results are graphically represented in the diagrams, Figs. II. and III., accompanying the tables of each series. In the tables the exact conditions are stated under which the two distinct phases of the operation are conducted.

Series I.—The material used was an ordinary mild steel of the composition :—

	Per Cent.
Carbon	0.12
Silicon	0.06
Manganese	0.47
Sulphur	0.02
Phosphorus	0.03

The results after each phase of treatment were as follows :—*

(I.) Case-hardening for ten hours at 1,100° C. with the mixed agent :—

(II.) After reheating the same carburised zone for five hours at 1,100° C. in isolated carbon monoxide :—

(III.) After reheating the same zone for another five hours (altogether ten hours of reheating) at 1,100° C. in isolated carbon monoxide :—

The numerical data contained in the three tables relating to the first series are graphically represented in the following diagram, Fig. II.

*Both in the first series of experiments, as well as in the succeeding series, similar pieces of steel were subjected simultaneously to each of the various treatments and under identical conditions, one piece of which was used after each operation for the purpose of making a quantitative determination of the carbon in the successive layers.

Series II.—The material used was a chromium nickel steel of the following composition :—*

	Per Cent.
Carbon	0.33
Silicon	0.06
Manganese	1.15
Sulphur	0.02
Phosphorus	0.015
Chromium	1.50
Nickel	3.17

The results after each single phase of treatment are as follows :—

(IV.) Case-hardening for ten hours at

lated carbon monoxide is sufficient to demonstrate clearly how, with the rational application of the process of case-hardening under consideration here, it is possible to obtain carburised zones in which are eliminated the principal causes of brittleness and peeling which always occur, in a greater or less degree, in steels case-hardened by the ordinary process. The principal causes of these defects are excessive concentration of carbon in the superficial zones of the case-hardened pieces, and the rapid diminution of the carbon in passing to the deeper

Table IV.

No. of the Range of Single Layers Analysed.	Distance from the Surface of the Central Portion of the Layer Analysed, mm.	Amount of Carbon in the Single Layers.		
		First Determination per Cent.	Second Determination, per Cent.	Mean per Cent.
1	0·5	1·18	1·14	1·16
5	2·5	0 79	0·82	0·81
10	5·0	0·49	0·51	0·50

1,100° C. with the mixed agent :—

(V.) After reheating the same carburised zone for five hours at 1,100° C. in isolated carbon monoxide :—

(VI.) After reheating the same zones for a further five hours (total reheating period ten hours) at 1,100° C. in isolated carbon monoxide :—

The numerical data contained in the last three tables, relating to the second series, are graphically represented in the accompanying diagram, Fig. III.

layers. It is also worthy of note that this rapid diminution contributes in a high degree to render more intense the phenomenon of liquation of the cementite or of the corresponding complex carbides of the various special steels, and of the ferrite. These phenomena in their turn are productive of the sudden local variations of the concentration of carbon in the carburised zones to which is due the peeling off of the carburised and quenched zones.

Table V.

No. of the Range of Single Layers Analysed.	Distance from the Surface of the Central Portion of the Layer Analysed, mm.	Amount of Carbon in the Single Layers.		
		First Determination per Cent	Second Determination, per Cent.	Mean per Cent.
1	0·5	0·80	0·81	0 80
5	2·5	0·76	0 78	0 77
10	5·0	0·58	0·58	0·58
15	7·5	0·45	0·44	0·45

In view of the detailed study, of which the author has given an account in his previous reports, upon the causes of the various phenomena of brittleness and peeling which may occur in case-hardened steel, any further comment in explanation of the data in the preceding tables and represented in the two diagrams will be superfluous. In fact,

Finally, the ability to alter the form of the curves of case-hardening, as shown in the diagrams, suggests that case-hardening can also be successfully applied to those special steels on the exterior of which the ordinary process of case-hardening yields carburised zones, which are practically useless on account of their excessive brittleness.

Table VI.

No. of the Range of Single Layers Analysed.	Distance from the Surface of the Central Portion of the Layer Analysed, mm.	Amount of Carbon in the Single Layers.		
		First Determination per Cent.	Second Determination, per Cent.	Mean per Cent.
1	0·5	0·87	0·85	0·86
5	2·5	0·85	0·88	0·87
10	5·0	0·75	0·75	0·75
15	7·5	0·60	0·58	0·59

bearing in mind the conclusions to be drawn from the author's previous research work, a simple examination of the gradual changes of the form and position of the curves of concentration and depth of zone due to the effect of the treatment gradually prolonged with iso-

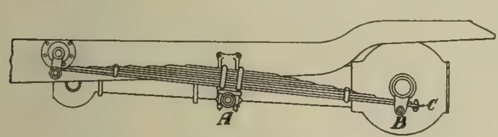
*As will be seen, this is a steel of a kind ordinarily used for Krupp armour-plates. On this account the data here given relative to case-hardening at moderate depth are of peculiar interest. But the author proposes later to present particulars of even greater interest relating to case-hardening to considerable depths as carried out upon this steel by the process described in the paper.

RECENT AUTOMOBILE PATENTS.

By Eric W. Walford, F.C.I.P.A.

Spring Suspension.

It will be remembered that in the Rolls-Royce cars the springs take no part of the axle torque or driving stresses. In the construction illustrated a single flat, inverted and laminated, spring is used at each side, being shackled at the front to the frame and fixed at the centre to a block A by means of the usual spring staples. The block is free to swivel on a pin and the rear end of the spring can slide on a roller B, which connects a pair of links depending from the axle. The

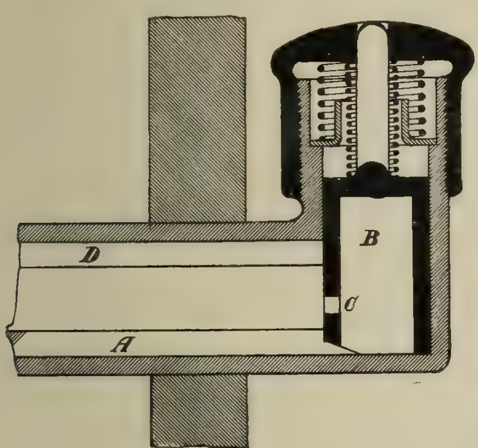


extreme end of the spring is provided with a stop at C, to prevent the spring from sliding out of the links.

F. H. Royce and Rolls-Royce, Ltd. No. 13516/11.

A Lubrication Device.

This device is located in the forced feed oil system and serves the purpose of an indicator as well as to prevent the pres-

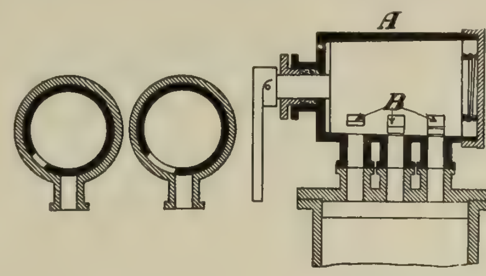


sure in the system rising too much. The oil enters by way of the tube A, and it has access to the underside of the hollow piston B. When the engine runs the piston B rises under the influence of the circulating oil, and its stem projects above the cover providing an indication that the oil is circulating. When the pressure becomes very high the piston rises to its extreme upper limit till an orifice at C comes opposite the outlet D, releasing the excess of pressure. It will be noticed that the piston is held down by a light spring and that outside this is arranged another spring, which only comes into operation when the oil pressure becomes too high.

W. F. Rainforth and D. Napier and Son, Ltd. No. 30019/10.

A Variable Compression Device.

Communicating with the cylinder is a chamber A, which contains a barrel valve controllable by the lever shown. This barrel is divided into three compartments, each having orifices B communicating with passages leading to the cylinder. The orifices B are of different lengths so that either one, two, or all of these can be opened up and the corresponding chambers in the barrel put in communication with the cylinder, increasing the

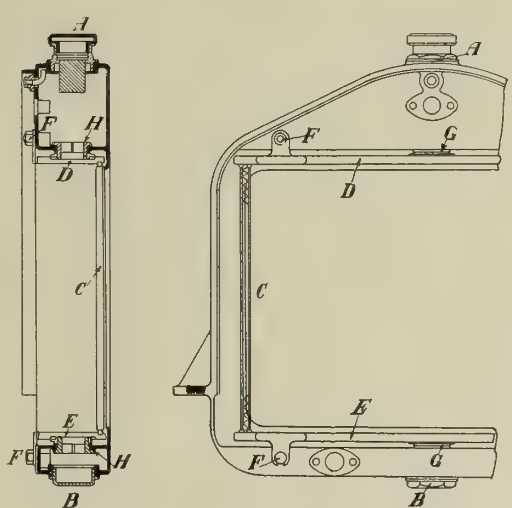


capacity of the compression space, and so lowering the compression in three stages.

C. Ransom. No. 5947/11.

Radiator Construction.

The top tank, side members, and bottom tank are all cast integral, being provided with the usual water pipe attachments, filler cap at A, and wash-out plug at B. The radiator tubes C are carried by flat tanks D and E, which tanks are bolted to the cast tank members by bolts shown at F. The water connections are shown at G, the connection taking the form of a screwed sleeve H, which screws through the edge of the cast tank into a screwed collar in the tanks D and E. This screwing is effected through the orifices A B, and it will be understood that the sheet metal tanks and the radiator tubes can easily be detached from the cast members by unscrewing the two screwed sleeves

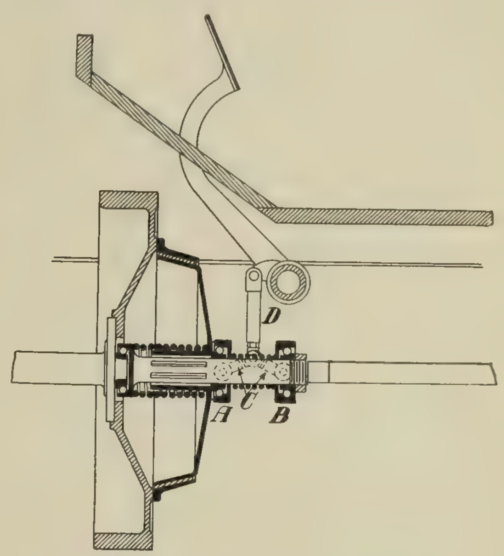


referred to. The cast tank is provided with a steam pipe or vent, as usual.

F. Lamplough and Son, Ltd., and W. R. Buckingham. No. 25,643/10.

A Friction Clutch.

It will be seen that this clutch is of the self-contained type, and that the male cone has to be moved to the left for disengagement. It is provided with a thrust ball race at A and on the clutch shaft is another thrust ball race B. The outer members of these ball races are connected together by toggle links C, the connections of which are acted upon by the link D connected to the clutch pedal. It will be seen that when the pedal is depressed the toggle links are flattened out and the clutch members separated. It is clear that the toggle mechanism is stationary, inas-

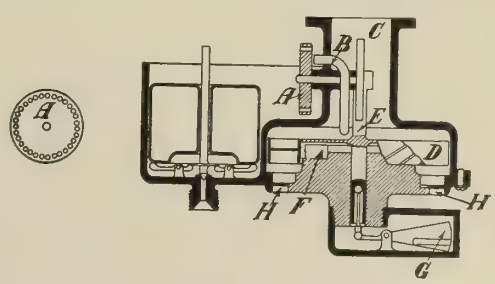


much as the outer members of the ball thrust races do not rotate with the clutch.

Daimler Motoren Gesellschaft. No. 24,151/10.

A Paddle Wheel Carburettor.

A float feed mechanism is provided which keeps the petrol at a certain height in the float chamber. Dipping into the petrol is a paddle wheel A provided with a large number of holes near its periphery. As the wheel rotates each hole in turn comes opposite the horizontal end of the jet tube B, the vertical part of which projects into the spraying chamber C. The suction in the spraying chamber causes air to be drawn along the jet tube B, with the consequence that the petrol is drawn out of each of the holes or short passages in the paddle wheel. At the bottom of the spraying chamber is arranged a turbine wheel D fixed to a spindle E, which is connected by spur gearing with the paddle wheel. Thus the paddle wheel is rotated by the air current passing through the spraying chamber. The paddle wheel rests on a brake ring F, its pressure thereon being regulated by an adjustable balance weight G, so that just the required amount of rotation is permitted. The air enters at H, the air inlets being closed by a weighted



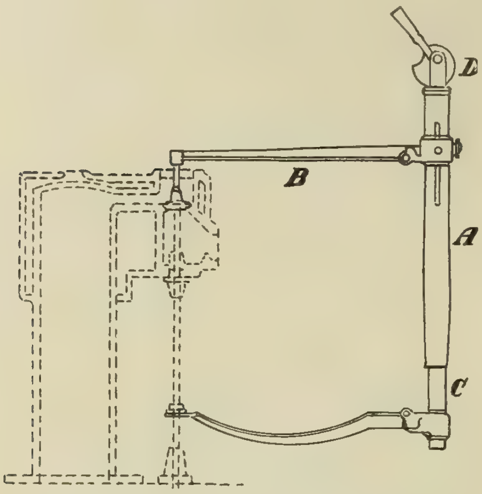
ring which ensures a reduction of pressure at all times in the spraying chamber.

H. Blackburn. No. 24,100/10.

A Valve Spring Compressor.

The body of this compressor is telescopic, and to the outer member A the upper arm B is adjustable attached. The

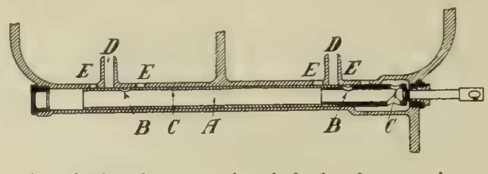
inner telescopic member C is acted on by an eccentric D, to which is attached an operating lever. The two arms are pivoted to their respective telescopic



members so that they can lie flat alongside the telescopic part A C, and the tool can therefore be packed away in a small space. J. Zehnder. No. 7,654/11.

A Lubrication Device.

This invention relates to a tap which enables the crank chamber either to be emptied of oil or the oil to be drained away to a certain level. The tap takes the form of a tube A movable endwise and provided with orifices B and C. In the position illustrated the tap is in its in-operative position, but when moved slightly to the left the opening B comes opposite the overflow pipes D, and if the oil level is above the overflow it is obvious that the excess of oil drains away through the tube A. By pushing the

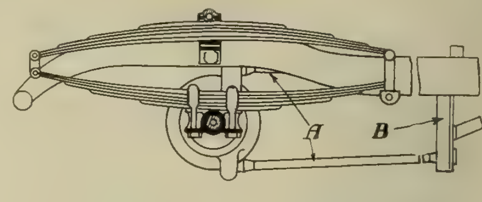


tube A further to the left both openings are brought opposite the outlets E and the crank chamber is consequently emptied.

Société Anonyme des Automobiles et Cycles Peugeot. No. 12,847/11.

A Torque Rod System.

Above, and also beneath, the axle is arranged a torque rod A, the rods lying parallel and being universally connected at each end. At the rear they are connected to the differential casing, whilst at the front they are attached to a rigid hanger B, which is supported by struts from the frame. Apparently the drive is taken through these rods as well as the torque and the axle ends are free to move backwards and forwards slightly according to the resistances met with, such movement being opposed by the main springs. It will be see that these are of full elliptic shape, but the front end is attached to the frame so as to support the springs against any twisting action to which they are subjected as the axle ends

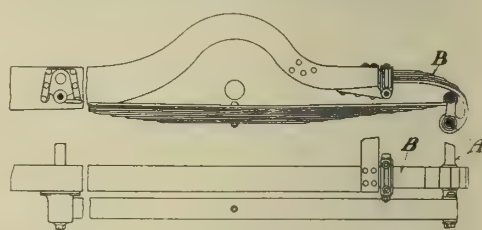


move backwards or forwards, and the result is a resilient resistance of such movement.

No. 25,708/1910. Austin Motor Co., Ltd., and H. Austin.

Improved Spring Suspension.

The main springs are shackled to a bar A, which projects across the rear of the car and is carried from the main frame by a spring member B. The object of em-



ploying this construction is to free the main springs from side strain and torsion.

No. 21,107/10. H. N. Ogston and D. Napier and Sons, Ltd.

CORRESPONDENCE.

No responsibility is accepted by the Editor for the opinions of correspondents, and the right is reserved to omit any portion of a letter if it is considered desirable to do so. Letters to be inserted with a nom de plume must be accompanied by the name and address of the writer. If publication in the next issue is desired letters should reach the Editor in no case later than the 20th of the preceding month.

A CRITICISM ON "VARIOUS USEFUL JIGS AND FITTINGS," BY C.T. SCHAEFER, M.S.A.E.

Sir,—Being a jig and tool designer, I was interested in the article entitled "Various Useful Jigs and Fittings," in The Automobile Engineer of Nov. 3rd, 1911. I am not quite clear on some of the points in this article, and I do not agree with others.

Referring to Fig. II. of the above article, I have found that it is better to have one fixed screw for locating the lever mentioned. With all the screws adjustable it is difficult to know when one is a revolution—that is, one pitch of the screw—out. I have tried this method and have had the heads of the screws divided equally so as to get a finer adjustment, and an indicator fixed to the jig so as to set the screws all at one level, and have found then that errors arise.

The author failed to mention how to set the milling cutter at the correct distance from the centre of the seating. A satisfactory way of doing this is to machine the web of the jig .015 in. less than the distance of the bore of the seating from its base, and to use a 15 thousandths feeler in setting cutter.

The jig for drilling jaws as shown in Fig. III. is not quite clear.

Is the piece in the "black," that is, as it comes from the dies? The author leads one to believe so—with regard to the length, at any rate.

Is the piece located by the rounded head and the flat of the jaw resting on the jig? This is scarcely possible, even if the piece is first of all machined—as some limit of error must be allowed.

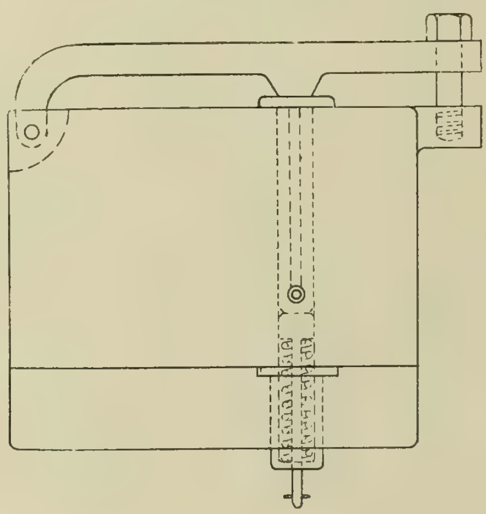
Is it possible to drill the lower side of the jaw without supporting it? The wedge shown will not supply the need. Screwed bushes rarely give satisfaction, as the thread wears badly, allowing the bush to vary its position. A sliding bush worked by a clamp gives greater satisfaction.

Why not try the job by first milling the jaw, and then drilling and tapping it on a turntable in a hand turret lathe?

The next jig I wish to draw attention to is illustrated in Fig. IV.

I think there is an error in the drawing of this jig. As shown, the taper pin projects through the base of the drill jig—which cannot be intended.

With the pin at right angles to the position shown, it would be better to have the jig made



of square stock so as to facilitate the drilling of the hole for this pin.

Is it possible to withdraw the pin after it is drilled? Will not the burr prevent it? I have found it necessary to have a slot both at the top and bottom of the hole to clear the burr.

The pin is removed, I presume, by pushing it out with another pin—a slow job.

The jig sketched herewith will be worth the extra first cost.

A hole is tapped in the base, directly under the bush, so that the jig may be bolted to the machine table. This type of jig can be worked rapidly and easily.

The pins to be milled, are to a minus two thousandths limit.

J. A. M.

SOME FACTORS IN TYRE ECONOMY.

Sir,—The question of the effect of the type of wheel upon the life of a tyre is an interesting one, which has often been mooted, and the figures given by Mr Mackle are rather striking; but before the critical mind can accept them as reliable evidence it must be satisfied that the only difference in the conditions under which the two sets of tyres were run, was that of wheel construction.

The only other information given about the cars was that "all were heavy covered cars," which is hardly sufficient.

One would like to know:—

(1) Whether all the cars had propeller-shaft or chain transmission, and, if not, whether the two types of drive were equally represented on the wood and wire wheeled chassis?

(2) Whether the engine power was the same for both types of wheels?

It is to be hoped that more information regarding the test will be forthcoming in order to give the figures a real value.

CRITIC.

BINDING CASES AND INDEX.

As this issue concludes Volume I. of the Automobile Engineer, arrangements have been made for the issue of a cloth-board binding case of the requisite size and strength. This cover will be supplied at 2s. 6d., post free 2s. 9d.

The index, which with the binding cover will be ready December 14th, will be supplied post free for 3d.

The publishers will undertake the binding of volumes for the sum of 5s., this including the supply of binding case and index. All orders should be sent to the Publishing Department, The Automobile Engineer, 20 Tudor Street, London, E.C.

ROLLER BEARINGS.—A catalogue of the Cooper Roller Bearing comes from the Unbreakable Pulley and Mill Gearing Co., Ltd. The catalogue contains a short description of this special type, some illustrations of its application, and the sizes of the standard bearings manufactured by the company.

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CONTENTS.

THE PROGRESS OF AUTOMOBILE CONSTRUCTION IN 1910, AND THE PRINCIPAL PROBLEMS OF THE DAY:

LOOKING back over the events of the past twelve months it cannot be said that development or improvement is particularly marked in respect to any one detail or any one type of automobile, save only the aeroplane, which is dealt with fully and separately, as it deserves, on another page. Improvement has been no less noticeable in 1910 than in former years, nor even has the rapidity of improvement slackened perceptibly, but it is becoming increasingly difficult to judge the quality of a chassis by inspection only, as the difference between goodness and indifference lies now more in detail than in general. Recently in *The Automobile Engineer* we remarked on the undoubted fact that manufacturers are beginning to pay more heed to the results of laboratory research or experiment, with great advantage to the efficiency of their cars, but it seems that they are giving equal attention to petty detail wherein it affects the convenience or the comfort of the driver or passengers. Thus, in not a few instances, it may be observed that a car, which appears unaltered when viewed casually, has had its few troublesome, if unobtrusive details improved very greatly, while it is usual to find that the running is distinctly better than before, either by reason of greater power or greater smoothness and greater ease of control.

For experimental study makers have most commonly chosen the engine, their investigations being directed towards obtaining the maximum power with definite cylinder dimensions. Many other manufacturers have given more attention to the removal of noise in the transmission, and a much smaller number have tried to improve both parts. As regards the former case, the result has been to increase very considerably the average piston speed of car engines, while the results of the second line of inquiry have so far been so small as to be little more than nothing. Also, of course, there has been considerable experiment with new engine systems throughout the automobile industries of the world.

Concerning engine improvement, it may be questioned whether the time has not come to change or modify its direction somewhat. By comparison with three years ago, or even a year ago, it is now possible to get very high power from very small cylinders, but the higher speed of rotation and piston translation thereby entailed usually causes quite perceptible and distinctly unpleasant vibration *when the engine is developing its full power.*

If not yet, it is certain that quite soon the elimination of vibration will be one of the chief tasks before automobile engine builders, and this may fittingly be given first place in the list of the principal problems of the day.

The use of the worm and the great strides that have been made in bevel gear manufacture have almost solved the problem of obtaining a quiet "direct" drive from the engine to the road wheels, but the effect has merely been to divert the issue, which is now to get the indirect gears as soundless as the direct—a much more difficult task. We have already remarked that but little progress has so far been made, and it is therefore reasonable to give this question a position of equal importance with the first named. It is not our present purpose to suggest ways and means, but there is, perhaps, some reason for the belief that the evolution of a silent change gear system lies deeper than the mere making of accurate gear teeth, and may indeed lie so deep that the way out will be found to be the abolition of the present type of gearbox altogether, with the substitution of an entirely different mechanism. It is apt to be forgotten that it is not long since everyone cried out against the unmechanical brutality of the sliding gear system, and though the brutality has been softened by detail improvement, it is still there. In the hands of a poor or indifferent driver the gearbox is now the least satisfactory and most easily damaged part of a chassis, and that some radical change will take place in the near future with regard to taxi cab transmission is very probable. Epicyclic change-speed gears have been tried thoroughly both here and in America, and it is safe to say that they have been found wanting because of their elaborate nature. If they are made well enough to be quiet and efficient they are very costly, while if the really essential four speeds are given the complication is very great indeed. The chain may have a future for change-speed gear work, but cost is again against it—though to a lesser degree—and the durability of short chains is too little assured for it to be possible to forecast events.

Meanwhile, the most striking fact in connection with power transmission is the establishment of four speeds instead of three as the standard arrangement, and this is a splendid instance of the inevitable ultimate survival of the best form of construction.

The subject of transmission introduces another problem which is the determination of the best ratios of the four speeds with regard to general convenience and utility. Reference to the gear ratios given in the tables published elsewhere in this volume shows conclusively that makers of similar engines are by no means agreed as to the best average speed (or normal speed) for those engines. For any engine and any normal load there should be a best series of ratios, and it appears that there is room for further investigation with a series of different conditions. Possibly an equation could be found whereby the ratios might be determined from the bore of the engine, the bore/stroke ratio, the number of cylinders and the weight of the chassis.

The next question is one which is less a matter of theory than the accurate observation of successful practice, with correct deduction of cause and effect therefrom. The somewhat indefinite quality commonly known as "holding the road" is not possessed by every car, and there are but few that hold the road thoroughly well. Cars of older types with much greater weight by comparison with their power than present patterns, were not usually sinners in this particular to the same extent as is now noticeable. Thus improvement in one direction has led to some retrogression in this one way. It is probable that the liability for a car to skid, sway, or roll is wrapped up with errors of steering, distribution of weight and flexibility of springing, but most likely steering error accounts for the greater part of the trouble.

"Holding the road" is perhaps a little difficult to define, but it may be taken to mean that there is no conscious effort needed to keep the car in a straight line or on a moderate curve when travelling at high speed. As has been already hinted, some of the latest "high efficiency" chassis are far from good in this respect—in fact, some of them are so bad that they would almost be dangerous in the hands of an inexpert driver. The increasing tendency to cant the steering pivots inwards at the top does not seem to exercise any definite influence, and, also, it is certain that flexible springs do not necessarily act detrimentally, though hard springs generally seem to minimise the tendency towards discomfort at high speeds. Much concerning this matter has been learnt at Brooklands, but it is not quite conclusive for road work, because the type of chassis vibration set up by the unevenness of the cement is not the same as that caused by a rough road, where the inequalities are mostly smaller and more frequent in occurrence.

To what extent skidding in mud is related to the chassis proportions rather than to the surface still remains one of the unsolved problems, but it is probably precisely the same as that of "holding" a dry road, and there is some reason to think that a car with radius rods to the back axle is less liable to skid than one which is not so fitted, although, of course, the influence of such axle staging is a very great deal less noticeable than of bad steering.

The points so far considered chiefly affect the comfort of the car, and the next item which needs particular attention is the efficiency, not in terms of the power developed by comparison with the weight or size, but in terms of work done per unit of fuel and oil consumed. It is, of course, true that petrol forms but an insignificant part of the total of the yearly running expenses of a car, but it is scarcely wise to neglect it altogether, as too many carburettor designers have done. It is still unfortunately true that very many cars are improved by their purchasers by the alteration of the existing carburettor or the substitution of a different type, while if manufacturers acted wholly in their own best interests, no improvement should be possible. Things are now much better in this particular than they were not long ago, but much remains to be done, and it must be done by organised research. There must be a best general type of carburettor for general all-round purposes, and the very large number of designs in use proves that the best type has either not yet been found or nor yet been appreciated. Several well-qualified investigators are giving their attention to the carburettor problem, and it is to be anticipated that some valuable facts will be ascertained from their experiments, but it behoves manufacturers to encourage such research, and to endeavour to profit by it rather than to disbelieve deductions therefrom because they may happen to disagree with their own much more rough-and-ready experiments.

Standards and Materials.

Turning to a quite different theme, there is also great need for simplification with respect to the numbers of different bolt sizes and the number of different threads used by chassis makers. It is a time-worn subject, but it is even more important to-day than it was a year ago, because the number of cars in use has increased so greatly. It affects the convenience of the owner, the professional repairer and also the manufacturer, because the latter needs to keep far more spare parts in stock if all his parts are peculiar than he does if many of them are standard. With the organisations available, it ought not to be hard for makers to agree at least to adopt metric standards and metric threads, or even British standards and threads, for all British cars, though the former are to be preferred to the latter on many grounds.

Another similar action that might advantageously be taken would be to reduce the number of different diameters and sizes of pneumatic automobile tyres, for they are now existant in quite unnecessary profusion, and are nothing but a trouble to makers and users alike. At the very utmost six sizes should be sufficient for all types of car, and in all probability four would meet the case. The innumerable sizes have appeared much more by accident than by design, and it needs only concerted action by the car manufacturers to put an end to many of them, while no one would or could have any cause for regretting their disappearance for ever.

The last-named matters, of course, concern the car as a whole rather than the chassis alone, and there is another matter to which engineers may be urged to pay some attention, though it concerns them only remotely. That is the accessibility of the parts of the chassis where accessibility is desirable, after the body is fitted. At the recent Olympia Exhibition the difference between the bodies made by the same factory as the chassis to which they were attached and the bodies made by coachbuilders was most marked, for whereas the latter were often better finished and better from the point of view of the occupants' comfort, they were seldom such as to attract the mechanically minded. The advantages of a hinge or some other swinging or sliding form of attachment are so enormous that it is truly wonderful that one or another of them has not become universal long since. Like the four-speed gear box, the easily removable body must survive in the end, and it is most surprising that the end is so long in arriving. This and similar evidence of lack of introspection on the part of manufacturers is probably owing to the fact that no member of the staff of a manufacturing concern can ever place himself quite in the position of a consumer—he has always the resources of the works at hand—and it is certainly this fact that leads to many an omission obvious to the experienced car user who has no connection with any constructional establishment.

In the materials of construction there has been no great change in the past year, except perhaps that some new aluminium alloys have made their appearance. The endeavour of the automobile metallurgist has always been to produce materials of very great strength, and some of these new alloys are most promising both in this respect and also as regards lightness. In all probability the growth of the aeroplane industry will tax the ingenuity of raw material producers far more than ever the road automobile has done, but leaving the air machine out of the question, it appears possible that steel may be replaced for many purposes by one of the aluminium preparations of greatly less specific gravity. For shafts and for all bearing purposes development tends towards the use of ever stronger steels, but such parts as the frame may soon come to be made of a lighter and softer but otherwise equally strong substance. Wood for constructional chassis purposes has almost disappeared, though it has advantages peculiar to itself, and even for body work metal is being used more and more every day, and it is not altogether improbable that the car of the future will be constructed entirely of metal.

Centralised or specialised manufacture does not seem to be gaining at all rapidly in popularity. That is to say, the number of firms that specialise on the production of one or two parts only does not increase perceptibly outside the manufacture of those parts which have almost always been made by specialists, such as radiators, carburettors, and wheels. Many arguments can be brought forward in defence of specialised manufacture and the reduction of complete car making to a mere selection and fitting together of ready-made parts, but the system is one that does not appeal to British sentiment, and rightly so because unless each part of a chassis is designed to suit its neighbours it is unlikely that the best possible whole will be obtained, however excellent each individual portion may be.

Considering all the problems now before automobile makers, it does not seem probable that any startlingly rapid developments will take place during the following year, but it may be anticipated confidently that the average excellence of British chassis will be as much improved by December, 1911, from what it is to-day, as it has been improved in the past year. The ultimate solution of most of the questions we have selected as being the most insistent might be found any day, but the discovery will far more likely to be so gradual, and the advances be made by such slow stages that to recognise the steps afterwards will not be possible. It is by the comparison of ideas, by the discussion of theories, and by the most minutely detailed observation that results can most quickly be obtained. The days of casual or rule-of-thumb methods have gone from the automobile industry for ever, and undoubtedly for good.

On several previous occasions we have urged the immense importance of quality in comparison with the much smaller importance of cost, and the present is a fitting opportunity for reiterating the belief that the present high position of the British automobile industry is due almost entirely to the foresight and ability of its professional side. There are men in this country to-day who are capable of designing and making as good cars as can be produced in any other part of the world, and to pay too much attention to cost of production is to waste the opportunity of becoming the leading nation so far as average excellence of product is concerned. In the British motor trade alone there are several historical instances of the effect of the placing of cost in a position of superiority to efficiency and durability. It is to be hoped that the trade has learnt its lesson, and that no more misfortunes to individuals will be needed to firmly establish the fact that it is quality and quality alone that carries with it lasting reputation and lasting commercial success. After quality price is of paramount importance, and always must be, but it is far the best policy to ensure the quality first and leave price for subsequent consideration.

ENGINE DESIGN IN 1910-11.

Being a consideration of the engines fitted to chassis exhibited at the 1910 Olympia Automobile Show with numerous detailed descriptions of especially interesting constructions.

ALMOST every writer on automobile topics has lately exclaimed upon the sameness of modern designs, on the extraordinary similarity between one chassis and another, and even between one detail and another, but this general close approximation to the average applies only if chassis are considered class by class, and is not by any means so impressive if the whole range are considered together.

Strictly speaking, there is little doubt that the last Olympia exhibition was distinctly misleading to the casual visitor because but few makers showed more than one or two models, often from a list of seven or eight, and in almost every case they chose similar types as regards price, which is equivalent to saying similar in respect to power and design. An examination of the tables which appear elsewhere in this volume will show that bores and strokes for similar cubic capacities vary most in their proportions amongst the very small and the very large cars, those of medium size containing the longer lists of precisely similarly dimensioned engines. Roughly 250 cars are dealt with in the tables, and of these 20 per cent. have a total cubic capacity of from 1,000 cc. to 2,000 cc., 27 per cent. from 2,000 cc. to 3,000 cc., 27 per cent. from 3,000 cc. to 4,500 cc., 15 per cent. from 4,500 cc. to 6,000 cc., and 11 per cent. over 6,000 cc. These have been dubbed classes A, B, C, D, and E for convenience in reference, and it is in classes B and C that the greatest similarity of design is to be found. Class A is a hetrogeneous mixture of single-cylinder, two-cylinder, and four-cylinder cars. Classes D and E are a collection of cars with widely-varying cylinder dimensions, and four or six cylinders, while it is in these last two classes that most of the short-stroke engines are included.

Once these considerable class differences are made evident, it appears that in making any deductions, or in determining the average of design (standard practice) it will be far more instructive to deal with the classes separately than to allow the strong likeness in certain classes to be rendered less striking by broadening the basis of calculation and dragging in the large variation classes. In the following pages, therefore, it is proposed to deal with the chassis in order of size, and to sum up briefly in conclusion by a comparison of class with class. As regards the method of classification, i.e., cylinder capacity, it ought, perhaps, to be explained why this has been chosen, though the tables are sufficient to show that it results in a reasonable form of grouping. The principal reason was the lack of a formula which would deal with each type of engine with equal fairness, owing to the neglect of a factor for the piston speed, and a volumetric comparison gives a truer measure of the power than does the bore alone. Secondly, it provides a ready means of comparing four and six cylinder engines, and shows up with emphasis the way in which the small six-cylinder is contending with the large four-cylinder.

Lubrication Systems.

Taken as a whole, engines have improved very considerably during 1910, especially in classes B and C. Detail improvement has been wonderfully rapid, and no retrograde tendencies are to be noticed. Foremost amongst improvements is the immense increase in the number of engines having some form of forced lubrication; in fact, it may be said that the recently universal plain splash has disappeared to a sufficient extent to almost put it on a par with chain drives, though it is less likely to linger on for commercial work after the manner of the latter. The trough system of lubrication is very popular, probably partly because it is less troublesome to make than a full forced system; but it may be observed that several makers who commenced last year with a simple trough arrangement have now provided forced leads to the main crankshaft bearings, allowing either the overflow therefrom or separate leads to maintain the level in the troughs. The trough system has the disadvantage that it requires very delicate setting in the first instance, and it is very difficult to protect the cylinder walls sufficiently while at the same time allowing enough oil to be splashed by the dippers or scoops to supply all the bearings of the engine. A pump is, of course, an essential portion of it to keep the troughs supplied, and it is thus both easy and desirable to dispose of the main bearing lubrication by direct feed from the already existent pump. The scoops can then be set to a finer depth of dip, having only to supply the big ends, small ends and cylinders. The completely-forced system is troublesome in that it calls for a drilled crankshaft, but, somewhat mysteriously, it is not so difficult to control; there is less danger of over-lubrication of the pistons, and a greater bearing durability is to be expected. The principal charge brought against the forced system as compared with the trough is that it is wasteful of oil, though the record oil economy has been obtained from the latter. However, practical road experience with a number of different cars goes to show that there is very little difference under ordinary conditions, while there is no doubt that the higher the pressure of the oil when it reaches the bearings the greater will be their life and the longer will be the time during which the engine may be run at full power.

Thus there is some reason to believe that the completely forced form of lubrication is likely to become the standard form as time goes by, though it is probable that it will be long before the trough system disappears, because it is an excellent compromise for engines which seldom run at their full power for more than a few minutes together.

It is in small detail that the greatest improvement with regard to lubrication still remains to be effected, and here a study of current design shows that many makers have changed their lubrication systems without a full appreciation of the possibilities of pump-fed oil. There are two essential points which should be borne in mind in connection with lubrication; the first, that the higher the pressure of oil fed to a bearing the longer will the life of the bearing be, and the second, that the purer the oil the longer will it last and the better is it for the bearings. Though these two facts are known to every engineer they are utterly disregarded by very many automobile constructors. A few are using high pressure oil and have overcome the tendency for it to produce a smoky exhaust, and a few are providing proper filtration for the used oil before it returns to the bearings, but they are in each case only a few, and thus it may be well to examine particular instances as typical of good or bad practice.

When oil is supplied to the main bearings of a crankshaft under pressure it flows from the ends of the bearings, comes in contact with the crank webs, whence it is whirled off in quantities more or less directly dependent upon the original pressure of the oil. To protect the cylinder walls from receiving a too copious supply baffle plates have always been used, but they are limited in value, as they also act as oil traps for such oil as does pass through the connecting rod slot which must be left in them. Where baffles have been found insufficient, two other methods have been tried, these being directed not so much towards preventing the deposit of oil upon the cylinder walls, but to its removal before it can be passed up to the combustion chamber. The first is to provide the piston with an additional ring near the bottom of the skirt, which acts as a scraper, and the other is to drill a number of holes in the skirt, the oil then passing to the inside of the piston, where it can do no harm. As an example of the former method the piston of the Sheffield-Simplex engine (shown on another page) may be studied, and the drilled piston process has been used for the racing Vauxhall cars which have been so constantly on Brooklands track during the past year. It may also be observed in the large portions cut from the Sunbeam engine pistons as shown elsewhere. A few firms use both scraper rings and holes, and a few both rings and baffle plates, but it is noticeable that the two cars with the highest pressures—Vauxhall and Lanchester—find that baffles are not essential by any means. It may, therefore, be taken that there is no special difficulty in using oil at fairly high pressure—from ten pounds per square inch upwards to as much as thirty or forty—and that pressures are likely to increase as engine speeds go up.

The next point in connection with lubrication is the ensuring of a pure oil supply, which is equivalent to discussing the filtering arrangements with circulated oil. These are, as a rule, hopelessly inadequate and to leave them in this state is spoiling the ship for the proverbial ha'porth of tar. In considering a filter, it should be remembered that its purpose is to remove every particle of solid matter, and also it must not offer too much resistance to the

passage of thick oil. Thus single layers of gauze are useless, because if they are fine enough to clear out every piece of grit they will clog at once. Probably at least six different layers should be used, and the last filtration should be through finer gauze than the first, while it should be possible to clean the filter without the loss of any oil from the base chamber, and without the removal of awkwardly situated nuts. A common form of filter with the trough system is a large gauze tray covering the whole of the cross section of the crank case, but the value of this is very doubtful, as it is of necessity awkward of access and cannot be made fine enough in mesh to do more than remove the grosser pieces of carbon. The equally common cylindrical gauze, with two or three layers of material, placed over the pump intake is much more effective and more easy to clean, though it is wonderfully seldom that this operation can be performed with a full sump. Also the gauzes of this type are generally so small in area that they require frequent cleaning, and are often so arranged that any deposit formed by sludge tends to stop up the channel in which they lie, as there is no drain pit into which it can fall. There are many things in favour of an entirely external filter, such as is used on the N.E.C. cars, for this is both easy to reach and easy to clean while offering a large filtering area, and there is no doubt that the device on the engines fitted to Commer Car chassis (whereby a valve closes the entrance to the chamber on the release of the gauze cylinder) is also strongly to be recommended to the attention of designers.

At present the disposition of parts tempts an ignorant driver to believe that his only duty is to fill the sump every few hundred miles, and but few attempts are made to call his attention to the existence of a filter, let alone the need for cleaning it.

Amongst oil pumps the gear pattern is used far more than any of its rivals, owing to its simplicity and cheapness. It is but little inferior to the plunger type as it can supply oil at any pressure likely to be desired for automobile work, and it is infinitely superior in every way to the centrifugal or vane type. As a rule, its dimensions are distinctly meagre, especially as regards the width of the wheels. The Lanchester, which is claimed to give a feed at forty pounds per square inch pressure, is a notable exception, being of a width equal to three times the diameter of the wheels. Accessibility of oil pumps is commonly as poor as that of the filters, but the few exceptions will be mentioned in due course, and a good deal may be learnt from the different styles of fitting.

Still, as a rule, the inaccessibility of the oil pump does not matter very much because it is but seldom that it should need any attention.

Oil ways and oil pipes are generally a little larger this year than they have been previously, and they may be expected to increase in size still more. Here the crankshaft of the Armstrong-Whitworth engine serves as a good example and so do the pipes on the Lanchester, despite the high pressure. The chief advantage of large pipes is their freedom from resistance when the oil is cold, and it is an admitted fact that the majority of bearing wear takes place during the first few thousand revolutions of an engine each time it is started up with cold and therefore viscous oil in its bearings.

Crankcases, Shafts and Bearings.

The introduction of forced lubrication systems has had the effect of considerably complicating the crankcase, both the upper and lower portions being affected. The upper half now not only carries all the bearings, but it is rapidly becoming standard practice to cast—in copper pipes, which afterwards form the oil leads to the main bearings, while provision has to be made for the oil pump drive, this usually being situated between the two rear pairs of cylinders and commonly consisting of a skew gear. This means that there must be a chamber for the gears, and also bearings for the vertical shaft. Externally when there are four supporting arms these are most often joined together by a web or tray on which control fittings, the magneto, etc., are mounted. The strong desire for quietness has also led to the use of a large number of cross-shafts at the front of the engine, for driving the pump and the magneto, though it is doubtful whether the small gain is worth the cost, as the case is naturally complicated considerably by the addition. As regards the lower half this now needs to be provided with a sump and a filter chamber, if not an oil pump chamber as well, so it is less simple, both as a foundry and as a machine shop job.

Crankshafts are still usually made from forgings on which an enormous amount of stock remains to be removed, the stamped shaft having gained but little ground. As stamped shafts can now be made of good quality material and with such accuracy as to require grinding only before they are ready for use, this is the more surprising, and the same applies with equal force also to camshafts with solid cams. As strokes have increased, the need for greater strength in the shafts has given rise to the use of very strong steels which has perhaps militated against the stamping somewhat, but there is no doubt that strength for strength, a stamped shaft is much cheaper than one of the old type, while it need be but little heavier if it is made hollow.

Camshafts more often than not have solid cams, but many of the reputed first-class makers still prefer the loose cam on account of its manufacturing advantages. As regards efficiency, there should be nothing to choose, if the jigging is equally good in each case, while the cost also should be much the same. Probably the cheapest of all ways of making cams is to use a drop forged solid shaft, and to grind direct from the rough, where the necessary special machine is available.

Owing, perhaps, to the influence of the Knight engine, several engines are now made with a chain drive to the camshaft instead of the more customary spur gearing. This is, of course, quite silent, and some such engines have now been running long enough for their durability to be assured, and in one or two cases, two, three, or even more chains are used for driving different parts. Further, there are one or two ingenious methods of driving the camshaft, without the use of either chains or spur gearing, and these are mentioned in detail further on, but the fact that the once rare cross-shaft at the front of the cylinders has now become quite common for driving the magneto or the pump, or both, is further evidence of the great importance which is now given to silence, because the appearance of a cross-shaft most frequently means that the manufacturers in question prefer to use a skew gear rather than an extra spur gear which may be noisy, and will rarely be quite quiet. Methods of obtaining silent running spur gears for camshaft drives are very numerous, and in certain cases the large wheel is now made from a fibrous material, otherwise, where the gears are both metal, they are almost invariably single helical, but this appears not to have a very great effect in silencing, and creates an undesirable thrust.

The standard position for the gear for driving the oil pump is between the last two cylinders on the camshaft, and the vertical pump shaft is usually made with a telescopic joint, so that the pump can be dropped and replaced without trouble. In such cases the horizontal skew gear wheel is provided with bearings, and the connecting shaft to the pump is free to run slightly out of line if necessary.

As considerable interest is now being taken in the vibrations in six-cylinder, and even in four-cylinder crankshafts, mention might be made of the fact that so far but little practical effect is noticeable. There are a surprising number of six-cylinder engines with three bearing crankshafts, and it would be hard to say whether these are noticeably less smooth running than engines with seven crankshaft bearings, especially as the latter are usually larger and slower running. However, it may be taken as an undoubted fact that the seven bearing shaft, being more rigidly supported, is less liable to vibration of any kind, than the shaft with a smaller number of points of support.

For the crankshafts of four cylinder engines three bearings are commonly used, and they are roughly proportioned so that their total length is from six to seven times the diameter of the shaft, big end bearings usually having a length equal to from one and a half times to twice the pin diameter. Where five bearings are used the total length is increased from ten to twenty per cent., the big end proportions being unchanged. In only a very small number of instances are less than three bearings used for a four-throw shaft, and where this has been done it is admittedly for the sake of cheapness. With very small cylinders—not above 70 mm. bore—it may be satisfactory, but for larger powers than the two bearing shaft needs to be enormously heavy if it is not to whip. There are many manufacturing advantages in the three-bearing arrangement, and probably a slight increase in the size of the shaft will make a three-bearing crank practically as satisfactory as a five-bearing design.

Ball bearings for engine work have made but little headway, though they are coming into use for the camshaft. There are, of course, a few more engines with ball bearings for the main crank journals and in most cases these bearings are of very great size, and must be extremely costly. It is known that a ball bearing engine is almost invariably noisy, and seeing what an extremely small amount of trouble arises from the wear in a white metal bearing, it is not easy to see in what way the ball bearing has advantages. On a very large engine it makes starting less difficult, because the engine is noticeably more free to turn when cold,

but when warmed up the difference is by no means so striking. In most cases where ball bearings are used extreme precautions are taken to guard against any thrust coming upon the journals, and it seems that the disintegrating action of axial load is far more pronounced with large bearings than with small, or—it may be—is more pronounced with impactive loads.

Crankshaft diameters do not appear to vary according to any definite rule, very few engines having shafts more than 50 mm., while the average for the car classes B and C would be about 40 mm. to 45 mm. The crank pins usually have the same diameter as the other shaft journals, and the webs commonly have a cross sectional area roughly equal to that of the shaft, though the exact proportions vary considerably, according to the stroke/bore ratio. For small ends the fixed gudgeon pin with a phosphor bronze bush is practically universal, the most common exception being to use a hardened steel bush. Where oil is not forced up the connecting rod the phosphor bronze is less likely to give trouble, but with a pressure oil supply the steel should be more durable. In general there is no sign of a tendency to use larger small end bearings as engine speed increases, though where the ring fixing is used for the pin some makers reduce the width of the lugs on the piston, making the bearing wider.

Bearings of subsidiary shafts are commonly plain, but ball bearings are often used, especially on vertical shafts, and shafts that are driven by a skew gear usually have a ball thrust bearing, though this is not always the case.

Pistons and Cylinders.

Taking the smaller subject first, the steel piston has not found many fresh supporters, especially the pressed steel variety. This is probably due to the expense of the latter form, and its small practical advantages, for when it is remembered that 80 mm. cast iron pistons are in use weighing little over a pound complete, with rings and gudgeon pin, that the advantage of the steel is very small becomes at once obvious, and steel pistons are more liable to seize. Malleable cast iron is used for a few of these light pistons, and so is cast steel, but the staple material is good cast iron. Very few pistons are given more than three rings, unless the fourth be a scraper ring at the bottom of the skirt, and it is probable that the increasing use of grinding for finishing both cylinders and pistons will result in only two rings becoming standard. With long stroke engines (by which is meant engines with a stroke/bore ratio exceeding 1.4) there is a tendency to cut large holes in the skirt, so as to reduce the weight while leaving enough material to act as a guide. It would be unsafe to make any prophecy concerning piston development in the near future, but it is safe to say that the important bearing which the weight of the reciprocating parts has upon the smooth running of a high-speed engine is now being fully realised, and greater reductions may be expected.

Gudgeon pins are most often fixed by the old-fashioned set screw, or by a ring placed over the ends. The latter method is one which has always been found satisfactory where the gudgeon pin is keyed to prevent its turning, and is often reliable where this precaution is omitted, although the lack of a key has been known to cause breakage of the ring.

Very few pistons have arched heads with a pronounced curve, but there is often a web to support the head. In most engines with any pretence to be considered first-class, care is taken to balance the pistons against each other till they are of equal weight.

Cylinders cast singly and cast in pairs have changed but little in design, but great improvements are to be found in the castings of three or more. Firstly, the water ways are now larger than they were commonly a year ago, and greater efforts are made to keep the whole casting at an even temperature. Valve seatings and pockets almost always have water jackets, and in many cases the cylinders are quite separate, as instance the Argyll engine shown on page 9. The practice of making the exhaust pipe also part of the casting is not increasing; in fact, several instances can be found where this habit has been given up, but the inlet passages are more often cast-in than not, which is probably to be recommended, as it ensures warming of the mixture on its way to the cylinders.

In a few cases the portions of the cylinders below the jacket are separate, with a space between each, but more often there is only a single opening between the middle pair of cylinders in a four-cylinder casting, this being made more or less necessary if a reasonably wide central bearing is to be provided.

Now and then the tappet guides are part of the block casting, but more often loose guides are secured, either in holes in the flange of the casting, or in the crankcase. Valve guides are still commonly simple holes drilled in the casting, but there is evidence of an increasing tendency to use bushes, as they allow the casting to be simplified a little, and are also a help in machining.

A single inlet and a single outlet for the cooling water is usual with multi-cylinder castings, and there is only one inlet flange on most of the small engines of this type. With large multi-castings the tendency is to treat them as pairs so far as piping or passages are concerned, but in only isolated instances is there more than one exhaust passage external to the casting, a single loose-ribbed pipe taking the four exhausts being standard practice. This is open to some criticism, for the reason that some exhaust interference must take place, with the usual sequence of firing, and one or two attempts have been made to overcome this defect, as will be seen later.

Of course, the valves are always all on the same side of the cylinders, in multi-cast arrangements, and six holding-down bolts are the rule, though eight are sometimes employed. With the six these are often carried through to be used for holding up the caps of the main crankshaft bearings—a highly commendable design where other proportions permit.

Much has been written, and much will probably be written in the future as to the wisdom of this present popularity of multi-castings, and the subject is too wide to be opened here. It may, however, be said that from the point of view of the user of a car, who has now and then to remove deposit from the combustion chambers, the pair casting is the most convenient (having due regard to the greater number of pipe connections necessary with single cylinders). For manufacturers the multi-cast form is cheaper if there are special tools for boring, and the assembling is also cheaper. The percentage of wasters is little higher, now that founders are becoming accustomed to accurate and complicated moulding, and the machining of the crankcase is simplified a little by having all the cylinders in one block.

Valves and Valve Gear.

It has already been remarked that the valves are commonly all on the same side in a multi-cast cylinder engine, and the same applies to pair casting, though with less force. It is a little surprising that the opposite inlet and exhaust, lately so common, has fallen off so quickly, especially as the diameter of valves has increased very much indeed. Usually the valves are as large as the cylinder casting will permit, and it is obvious that opposite placing would enable them to be larger than is possible when they are all on one side. A good deal has been said concerning the better thermal efficiency to be expected from an engine with only

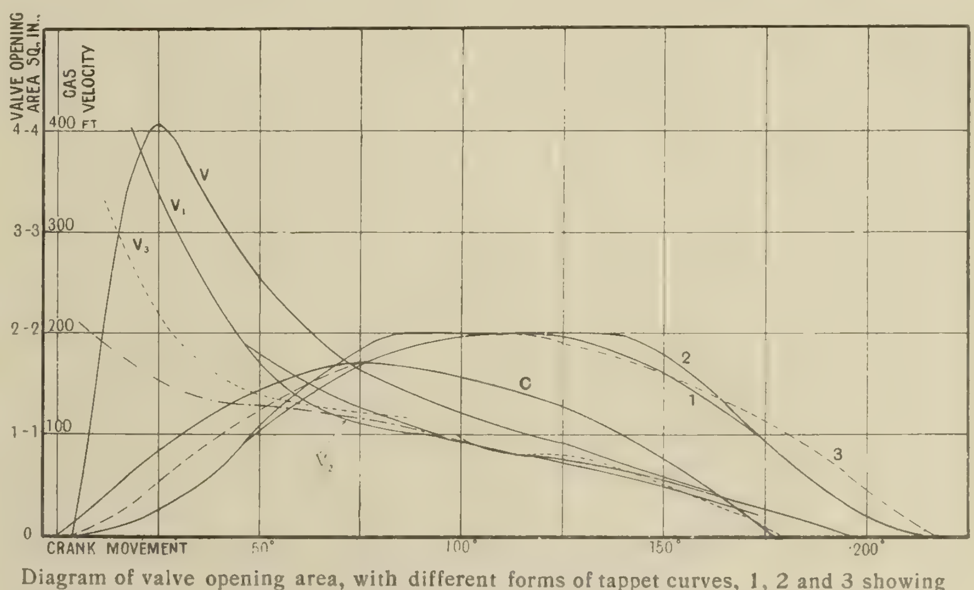


Diagram of valve opening area, with different forms of tappet curves, 1, 2 and 3 showing the comparative areas of opening of valves with a roller ended, a plain ended, and a mushroom ended tappet respectively. Curve C shows the corresponding momentary piston speed. V_1, V_2 and V_3 show the corresponding theoretical gas velocities for the three tappets, and V is a probable actual curve.

one pocket to each cylinder, but the real reason for the change is most likely to be found in the fact that one method needs two camshafts, and the other only one. The extra shaft carries two objections, namely, it is, of course, an extra expense, and its extra driving gear is liable to produce noise. It seems that cylinder castings, whether in pairs or higher multiples, can accommodate large enough valves on one side only, if the water spaces are as large as they should be, or if the casting is flared on the pocket side, but if a short engine is desired, then opposite placing has advantages.

To give any rule for valve diameter would be misleading, as it must depend greatly upon the purpose for which the engine is needed. For racing work the valves cannot very well be too large, but for ordinary road driving a very free entry and outlet is not necessary. Low lift is desirable, because it causes much less noise than high lift, on account of the cam form being easier. This leads to the consideration of profile, and here the flat-sided cam is still the most common, though a good many with concave flanks were to be found on the engines at Olympia. As a matter of general interest we re-publish a diagram which appeared in *The Automobile Engineer* for October last, and which gives the valve opening diagram for three forms of tappets, with a straight-flanked cam, namely, the V-ended type, the most usual roller-ended variety, and a tappet with a mushroom foot, the last-named being almost unknown here, though used in some other countries. With a roller tappet the cam with concave flanks gives a diagram more like the ideal form than either of those in the illustration, and it would be possible to design a profile so that the velocity of the gas would always be constant throughout the stroke, though to obtain a mechanism which would ensure the valve following the profile might cause some difficulty.

Tappets are now invariably adjustable, the exact arrangement varying, but generally being allied to the old set screw and lock nut. In a good many instances there is a small cushioning spring, to take up any slack between the valve stems and the cams, and most tappets are faced with fibre let into the head of the adjusting screw.

Valve encasement is common on all small engines where new designs are frequent; in fact, it might be said that almost every engine designed during 1910 has been provided with covers, whether the valves are on one side or two sides. Valves with flat faces instead of the once universal conical seat appear to be gaining in general esteem. Whether the gas flow is more easy in such cases or whether the seating is more easy to maintain in good order is a matter which must be left for time to decide.

If it is safe to judge by averages it might easily be said that the sleeve valve engine had made but little progress, and other types than either the poppet or the sleeve, no progress at all. But in a case of this kind it is not safe to deduce from average alone, because the widespread interest in new valve systems which now exists has had a great effect upon the private investigations of many leading manufacturers. It is certain that various forms of valve gear for internal combustion automobile engines will exist side by side for a good many years to come, and it is very doubtful whether the ultimate survivor has yet been made, or even conceived. Regarding the only common type other than the poppet, which is the Knight engine, this has been improved in detail by the original manufacturers, and the foreign Knight engines follow much the same lines as the British example. Thus a description of any one example applies for them all so far as practice, as distinct from experiment, is concerned. This description has been given very frequently, and is re-capitulated briefly later, so no further mention of it need now be made.

Carburettors.

There is no doubt that, on the average, the carburettor is the least satisfactory part of present-day engines. Except for the Lanchester wick pattern, the float-fed spray type is universal in this country, but the methods of attempting to obtain a constant mixture at different engine speeds are legion. Last year a large number of manufacturers were using carburettors of the Wolseley pattern, that is to say, with a single or double jet provided, with a very small choke tube, and from six inches to four times that amount of small copper pipe leading to a mixing chamber where air is added through an automatic valve. In the better examples the small pipe was water jacketed, but it was more often not so provided.

Another almost equally common design was the multi-jet arrangement, which sometimes had an automatic air valve and a mixing chamber, but perhaps more often had a positively controlled hand air admission valve connected with the throttle. This year the multiple jet type has gained greatly in popularity, but the Wolseley type has decreased. There is also evident a strong tendency for manufacturers to adopt one of the numerous fairly satisfactory patent carburettors which are now being made by specialists. The subject of the theory of carburettor action is far too large to be dealt with here, though certain interesting facts concerning the carburettors fitted to some small high-power engines have recently been published. It may be stated as an undeniable fact that very few, if any, carburettors at present in use give anything at all close to a constant mixture at varying speeds, that the spring controlled automatic air valve is most difficult to deal with, and in short, that finality in carburettor design seems almost as far off as ever. Up to the present most carburettors have been the result of trial and error, the experiments being pursued till a reasonably good result was obtained, and then ceased, leaving the mechanism far from perfect. Much laboratory investigation still remains to be made, and till there is closer agreement between those who have been conducting experiments, it would be a bold man who would predict the ultimate survival of any of the existing types. It may, perhaps, usefully be emphasised that the only satisfactory method of testing new carburettor designs is by means of analysis of the mixture or exhaust gas throughout the whole range of engine speed, and that nothing is more misleading than road tests, at any rate in the earlier stages of investigation. It may also safely be said that most makers are trying carburettor designs more or less haphazard, substituting almost any new type which appears to give better results than the old style. It can be noticed that this is to some extent causing makers to follow each other's procedure, though sometimes in the reverse direction, one firm taking up a type which a rival has just dropped.

Cooling Systems.

It has already been mentioned that there is a tendency to increase the size of cylinder casting water passages, and no doubt the popularity of thermo-syphon, or natural circulation, has had something to do with this. Although more makers than ever have given up the use of a water pump, there are still quite a large number who prefer to retain it, even on quite small engines, and there is no doubt that a car can be made decidedly lighter if it has a good and fairly powerful pump. In singularly few instances is there sufficient head of water to make a thermo syphon system really safe, for usually it would be essential to keep the radiator full to the extreme top, in order to ensure circulation, and the loss of less than a pint of water would often cause the outlet from the cylinders to the radiator to become uncovered. Water pipes for natural circulation have grown in size, and a rough and ready rule would appear to be to make the cross-sectional area of the outlet and inlet pipes, at their smallest, equal to from a quarter to a third that of the piston. Where very tapering pipes are used with pair cast cylinders it has sometimes been found difficult to obtain good cooling of the hinder pair, owing to the water taking the easier path through the front pair where the pipes are at their largest. In at least one case this has been so pronounced that it has been found necessary to restrict the outlet from the front cylinders by the insertion of a baffle plate, which has been found to cure the trouble. Engines with pump circulation exhibit alteration in that the pipes are now generally larger, and the pumps also have increased in size, which is a most curious fact by comparison with the popularity of natural circulation. The advantages and disadvantages of the two different systems are not very pronounced, and it is almost entirely a matter of choice for a manufacturer. He can make all waterways large, and carry a good supply of water in a big radiator, or he can use a pump and cut down the dimensions and the weight. Contrary to popular opinion, there is no reason why large engines should need pumps more than small engines, if the proportions are kept the same, but where pumpless large engines are seen, it is also to be observed that the radiator and pipes are enormous, and consequently heavy. Had it not been for the ridiculously small, poorly-glanded and inaccessible pumps often fitted a few years ago, probably thermo-syphon circulation would never have gained as it has. All things considered, it is not a true simplification, and it is likely to be long before either system displaces the other completely.

Radiators still vary in design very greatly, though the vertical flat tube type without gills is becoming very common, so much so, in fact, that the gilled tube bids fair to disappear, leaving the honeycomb as sole competitor. In details of radiator construction enormous advances have been made, owing to a number of

factories having come into existence which specialise on radiator making. More and more car makers are supporting their radiators on swivel mountings, so as to relieve them of all frame stresses, and the radiator makers are attaching the brackets with greater security, so that the once troublesome cooler can now be obtained in a perfectly reliable condition in almost any pattern. After the flat tube and the honeycomb, the next type is that with horizontal gilled tubes, and this may even be more common than the flat tube pattern, the latter being placed first in the list as being the one which is making its way most rapidly.

Finally, one of the most noticeable facts in connection with cooling systems generally is the greatly improved accessibility of water joints, and of pumps when they are fitted. Rubber hose clips are stronger, and are designed so that they could be used again after detachment, while it is but seldom that a convenient drain is not supplied, so that the water can be run off completely.

Methods of Suspension.

Although the methods of securing an engine in the frame may more properly be considered under the heading of "Frames" rather than that of "Engines," it may be touched upon here, as it affects the whole design slightly. The most common method is to use four cast crankcase arms to carry the engine from the main side members, or, less frequently, from a sub-frame. A few engines only are supported by three arms, though three points of support are usual for engine-and-gear units. Still fewer makers attach the engine to cross members, though this makes a very neat job, and simplifies the crankcase casting. The adoption of worm drives has led to the sloping of the engine so as to obtain a straight line through the whole system when the loading is normal; in most cases this in done with an underhung worm, by dropping the rear end of the engine (the engine simply being placed high with an overhead worm) but in at least one case the front of the engine is dropped to give the same effect. This leads naturally to brief mention of the height of engines relatively to frames, and it might be said that the top of the crankcase (or the bottom of the cylinder castings) are usually about two or three inches below the level of the top of the frame sides. If a pious hope may be expressed, it would be that designers will not drop engines still deeper in their frames, as the present style of body design is not suited by a low bonnet, while a comparatively high engine gives improved accessibility of all its parts, and a better clearance. Many cars are now quite low enough to give trouble from lack of clearance on English by-roads, let alone the rougher ground in other parts of the Empire.

Silencing.

The removal of valve mechanism noises has already been dealt with, and so there remains the silencing of the exhaust and intake to be considered. Most present-day exhaust silencers create a good deal of back pressure, as they are still constructed upon the baffle plate principle, and are often very small. From a silencing point of view alone they are of a high average of excellence, but there is an occasional suggestion of humour when it is found that an engine with very large valves has a tiny silencer, which must quite counteract the effect of the size of the former. Silencing the intake is less often done, and the faint whistle of the carburettor is still the dominant note of most cars on the road. Oddly enough, but few makers concern themselves about this noise, because it is not noticeable at slow speeds, but it is so easy to prevent that it is wonderful that it is not got rid of more often. The only necessary is that the entering air shall not be drawn at high speed through a small orifice with sharp edges. The fitting of a funnel on the main air intake, and the arrangement of other ports to make the entry as smooth as possible is all that is needed to cure the sound.

General Conclusions.

Engine design has advanced enormously in the past year, the tendency being towards an ever-increasing stroke/bore ratio and growing piston speed. At present many engines run with a piston speed of over two thousand feet per minute in ordinary touring cars on ordinary service, and fully double this high rate of travel has been attained on the track. It is likely that piston speeds of three thousand feet per minute, and upwards, will be the rule rather than the exception in quite a short time. Whether the very high speed engine is the best for all purposes is a debatable point, and it is very doubtful whether the worship of high power (with respect to size) is not being carried too far, with probable harmful influence on smoothness and durability.

This fact has been attributed to the influence of taxation, but it is certainly not so in reality, as these small, high-power engines were in embryo long before the new taxes appeared, while development on the untaxed Continent has been more rapid than here.

Forced lubrication will soon displace all other forms, the trough system lingering beside it for slow speed engines.

Six cylinders seem likely to become the standard form for engines larger than the 90 mm. four-cylinder, though this development will be slow, and it is not certain.

Very strenuous efforts are being made to obtain smooth running by reducing the weight of reciprocating parts and improving the rigidity of shafts.

As regards new types of engines many experiments are being made and some types with rotary valves are certain to come into use for a time, if not for longer. So far no form of engine has been made in quantities which is likely to displace the poppet valve entirely and, if the latter does disappear, it will be a very long time before it has vanished completely, even for pleasure chassis work.

Much investigation is being made with two-cycle engines as well as with new valves, and the two-cycle system is gaining supporters amongst motor-cycle makers. This may be significant, or, on the other hand, it may not, but some of the more recent designs for two-cycle engines exhibit such great improvements upon former work of the kind that it is not improbable that the near future will see a contest not only between valve and valve, but between system and system as well.

Aeronautical engine development is mentioned elsewhere, but it is divided from car engine progress by many essential differences in the requirements. It has been suggested that the possibilities of the rotary cylinder engine are not yet realised at their full value, and that the rotary engine may come into terrestial use: at present it is very far from doing so, and there are many grave difficulties in the way. Still, this is a type which is developing more rapidly than any other in the aeronautical world, and there circumstances change with such intense rapidity that it is unsafe to forecast by even so much as a day.

Having thus considered the generalities of engine design and touched upon the tendencies of the day, some typical and some peculiar designs will now be described in greater detail. No attempt at a description of every chassis at Olympia has been attempted, because it is believed that no useful purpose would be served thereby, having regard to the special interest of our readers. Petty variations in detail design are so greatly matters of personal prejudice that to itemise the whole of them would merely be confusing. The system of classification mentioned in the beginning of this article will be adhered to throughout, and therefore commencement will be made with Class A. That is, with engines having a total cubic capacity of less than two thousand cubic centimetres.

CLASS A.—ENGINES WITH A TOTAL CUBIC CAPACITY UNDER 2,000 cc.

Single cylinder engines are few in number, except amongst the motor cycle type of light cars, which cannot fittingly be dealt with here. The De Dion Bouton Company, who have been the upholders of this type in France, have made practically no alteration in their well-known engines, and their various copyists have adhered to this same course of action. Turning to a completely different type, the same is true of the 10 h.p. Adams, with its Benz or Oldsmobile design of slow speed large cylinder engine. It is still a unique example of its type so far as this country is concerned. The only really novel single cylinder being made by a well-known manufacturer is the 8 h.p. Knight-Rover, which is allied very closely to the Daimler engines, and is exactly the same as the two-cylinder 12 h.p. Knight-Rover, with the one exception of size. Both these engines are carried high in the frame by arms from the main side members, and there is an arched aluminium web over the flywheel of the larger, see Fig. I. The lubrication system is the customary Daimler trough arrangement, the carburettor is a two-jet Daimler, and there is a very large water pump with a honeycomb radiator.

The smaller car has the flywheel in front and the gearbox bolted up to the crankcase as in the older Rovers. A plunger type of oil pump is used, driven off the valves shaft, and the pump and magneto are at opposite ends of a cross-shaft placed in the usual position. The 8 h.p. poppet valve Rover was described in detail in the issue of *The Automobile*

Engineer for last June, and since then but only the smallest of changes have

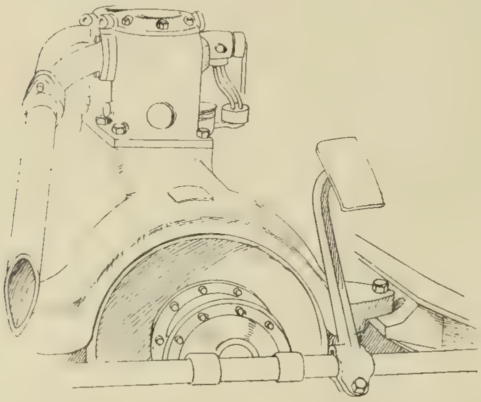


Fig. I. The 12 h.p. Rover engine.

taken place, so there is no need to recapitulate the features on the present occasion.

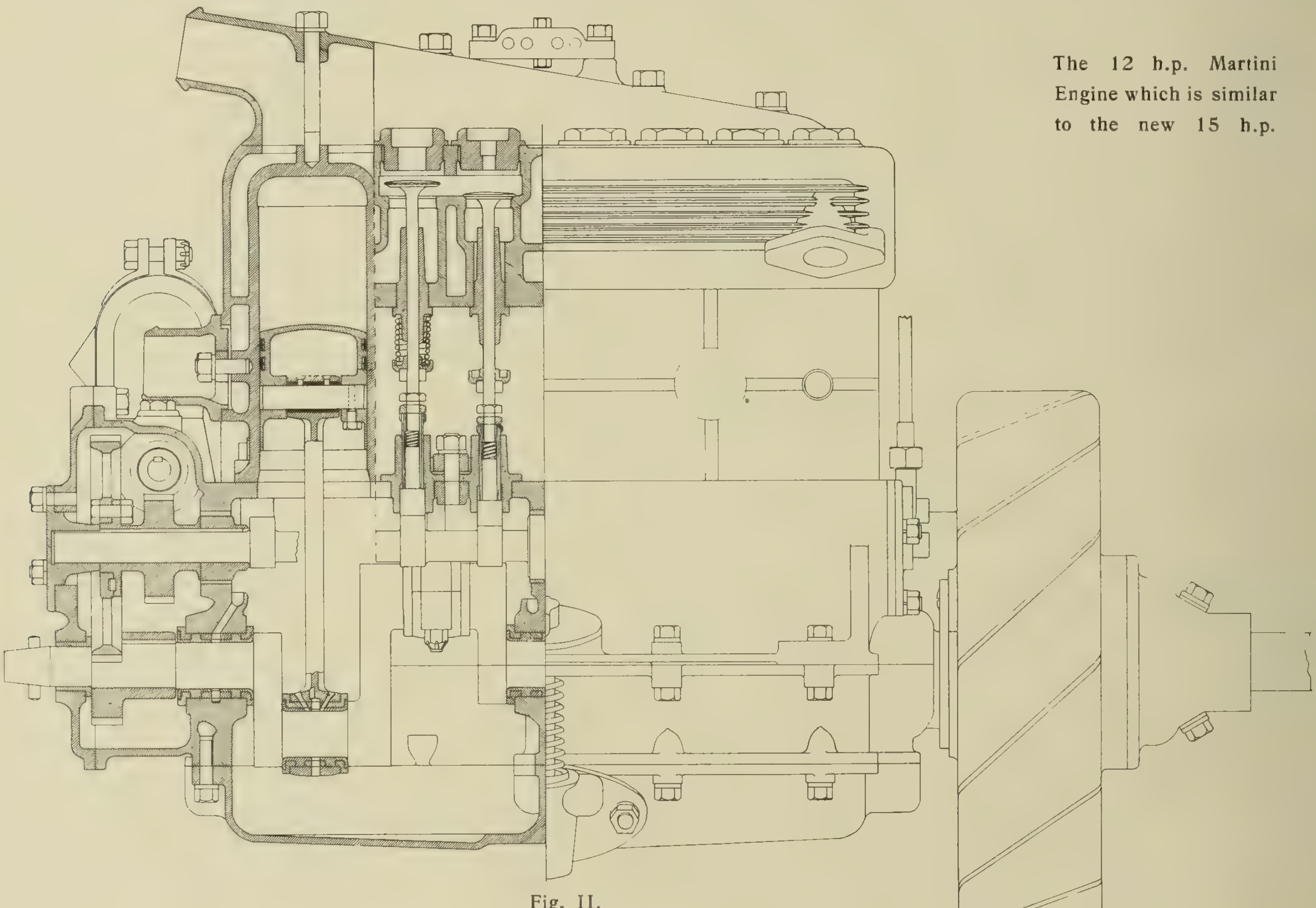
Another well-known two-cylinder design which has been similarly little altered is the 10-12 h.p. Swift, and the same applies to the 7 h.p. single-cylinder. These differ from the small Rovers in having thermo-syphon cooling, and are both splash lubricated. The crankcase arms are connected by the customary webs of aluminium, carrying the magneto on the off side, and each engine rests directly upon inward and downward extensions of the side members of the frame pressed in one piece with it.

The single-cylinder 8 h.p. Thames is a more peculiar design than any of the above-mentioned, having a ball bearing crankshaft and the valves arranged in front of the cylinder (facing the radiator). It has thermo-syphon circulation, but instead of using a single large outlet pipe, it has been thought better to use two separate pipes of smaller area, they entering the vertical flat tube radiator some distance apart. The camshaft takes the usual position relative to the crankshaft, the exhaust cam being operated through the medium of a bell crank lever, while the magneto is placed transversely and is driven by a skew gear, itself driven from the camshaft.

It had been anticipated by some that this last exhibition would disclose the existence of a number of two-cylinder V engines, but this was not so, there being only three or four in the building, principal amongst them being the two patterns of Riley engine. These again have not been altered, and have no features of a sufficiently novel nature to render their description necessary, as the old designs are very well known.

Passing on to the small four-cylindered cars here a large number of entirely new designs are to be found, but there are only small differences between the majority of them and the next Class—engines of from 2,000 to 3,000 cc. capacity. A typical example of Continental practice in this Class A is the 12 h.p. Martini, Fig. II. The four cylinders, the exhaust pipe and the inlet branches are all cast together solid, being mounted on a crankcase with four supporting arms, and the tray-form webs which have been mentioned as now being almost standard practice. The crankshaft is 30 mm. in diameter, and has three main bearings with an aggregate length of 200 mm. Lubrication is by splash, the oil level being maintained by drips on the dashboard, themselves supplied with oil by a belt-driven pump of the old Dubrulle type. It might be remarked in passing that French makers seem still to have an affection for this form of lubrication, for it was practically standard on the larger French cars at Olympia, though it is to be anticipated that this will be its last year of general popularity, except for quite cheap cars.

A very neat level tap is, however, fitted to the Martini engine, the handle being situated on one of the "trays," and being provided with a three-way indication giving the positions for overflow, correct level opening, complete draining of the crankcase and the closed or running position. It is shown in Fig. III., and is typical of the taps now standard practice. The carburettor, which is a single jet type, is a Claudel, and is attached directly to a single centrally placed flange on the off side of the engine cylinder block, so the whole of the inlet passages are internal and water-warmed. Aluminium cover plates enclose the valves and the cooling is by thermo-syphon, the water outlet pipe being a separate iron casting held in place by a pair of studs from the cylinder casting. The valves are 30 mm. in diameter, and the tappets are of the customary adjustable type with set screw heads.



The 12 h.p. Martini Engine which is similar to the new 15 h.p.

Fig. II.

Another good example of Continental design is the 12 h.p. Imperia, but it has an entirely peculiar lubrication system. The cylinders are 75 mm. bore by 100 mm. stroke, and are, needless to say, all cast in one piece. The engine is put in the frame by rivetting a pair of steel pressings to the side members, in order to deepen them after the fashion of the specially pressed frames which have already been mentioned. The engine crankcase has the usual four arms connected by webs, and this rests on the

inner edges of the steel pressings. The effect should be precisely the same as that of the deep pressed frames, and might, perhaps, be slightly less costly.

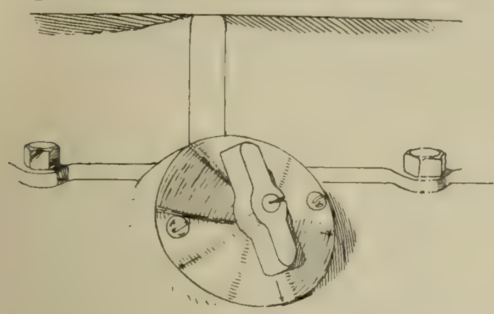


Fig. III. The Martini oil level indicator and drain tap.

For lubrication, the bottom half of the crankcase is divided into two halves, beneath and separate from which there is a sump. On the extreme rear end of the camshaft high up, and therefore in an accessible position, there is a plunger pump which sucks from the sump, and supplies the two halves, whence the oil is distributed by splash, while the height of the overflow orifice on each of the divided halves can be adjusted from outside to suit different oils or different working conditions. Thermo-syphon cooling is used with a honeycomb radiator and a fan with a very neat adjustment. The carburettor is a Zenith, and is provided with a very short inlet pipe connecting to the usual single flanged opening on the off-side of the cylinder casting.

The 12 h.p. Sizaire is an example of a totally different class of Continental design, and is, perhaps, specially interesting owing to the successes which the makers have obtained with their large single cylinder. It might be remarked that the latter is still being made, but has undergone no alteration whatever since last year. The new engine has a bore of 70 mm. and a stroke of 120 mm., the four cylinders being cast together. The inlet valves are arranged above the exhausts, and are operated by long tappet rods and exposed rocking levers, the same cams being used for both valves of each cylinder. The exhaust valves are enclosed by the usual form of cover plate. Lubrication is by splash, and the oil level is supposed to be maintained at such a height that the long pistons actually dip in the oil at the bottom of each stroke. The engine is mounted very low down on the frame, to which it is secured by brackets, and the magneto is carried in the position shown in Fig. IV., being driven by an exposed bevel gear from the crankshaft and revolving at twice the crankshaft speed. The ex-

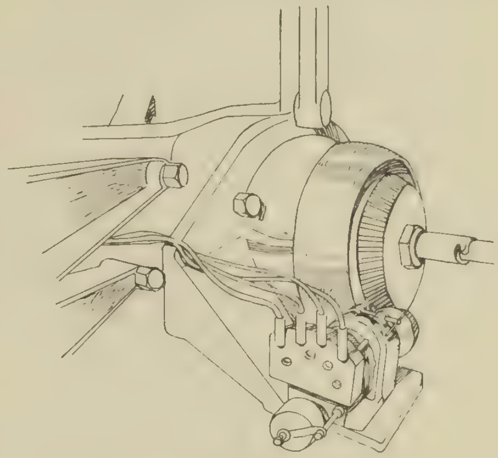


Fig. IV. The Sizaire Magneto Drive.

haust passage is cast in, and the inlet passages are dealt with similarly, the carburettor being carried on the off-side.

The 14 h.p. Metallurgique has been quite recently described in *The Automobile Engineer,* but as it is a particularly good example of Belgian construction, the sectional drawing is reproduced further on. The three-bearing

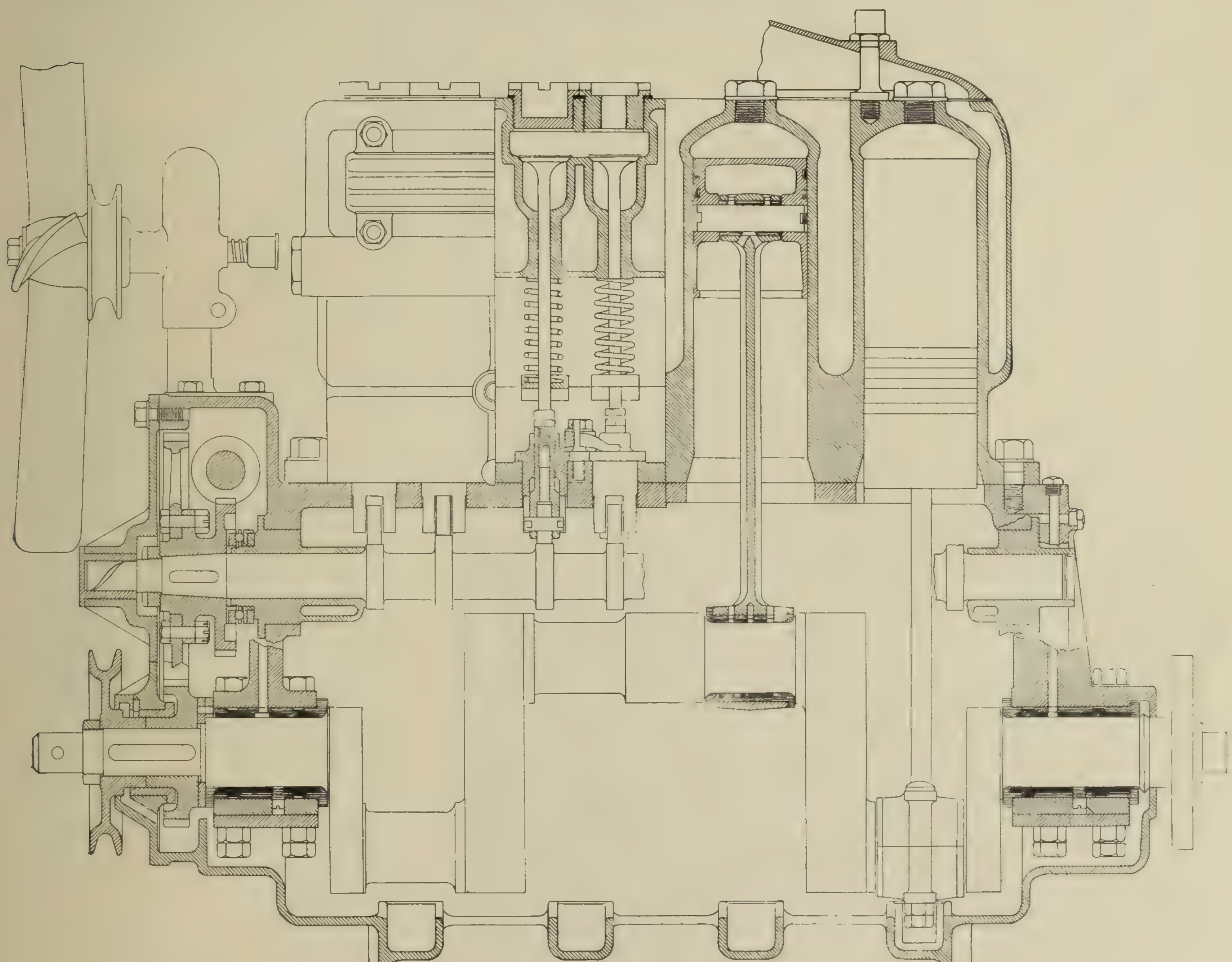


Fig. V. The 12 h.p. Argyll Engine.

crankshaft is off-set 25 mm. from the cylinders, the shaft being 37 mm. in diameter, with an aggregate length of main bearing of 175 mm., the big ends each being 54 mm. long. The oil pump is in a particularly accessible position, being carried at the front end of the camshaft, and is, of course, a gear pump, while provision is now made for priming it, should it be necessary to do so. The valves have exceptionally large dimensions, being about 35 mm. mean diameter, and it will be noticed that the seatings are very nearly flat, this being claimed to give clear passage to the gases. The shape of the valves is also worthy of special notice as it is typical of modern racing practice and may also be observed on one or two other engines illustrated elsewhere in this issue. The tappets are loose, and separate from the cylinder casting, the valves being exposed, and there is a cushioning spring between the top of each tappet and the underside of the valve spring washer, the purpose of this being to keep the tappet roller in continuous contact with the cam. This engine, although supported by four arms, has not got the usual webs, but is encased by a shield through which the bottom of the crankcase projects, exposing the filter which can be detached if needed. The magneto is situated on the off-side of the cylinders, being driven by a separate gear from the main timing wheel. The exhaust pipe is separate, with four flanges, and the carburettor is carried on the near side with a two-flanged inlet pipe.

Cooling is by thermo-syphon, with the usual V-shaped Metallurgique radiator, which by giving a considerable free space at the front of the engine still further enhances the accessibility, which taken all round is very good.

An engine of quite a different type to the last-mentioned is the 12 h.p. Argyll, shown in section in Fig. V. This has a bore of 72 mm. and a stroke of 120 mm., the compression pressure being between 90 and 100 lbs per square inch. Naturally, it is capable of very high rates of revolution, and is normally geared low. The cylinders are provided with exceptionally large water spaces, the space between each pair being as much as 30 mm., and the valves are distinctly large by comparison with the cylinder bore. The two-bearing crankshaft is a feature which is open to some criticism, but it will be noticed that the diameter is large, and the thickness of the webs is also considerable. The oil pump is carried on the side of the crankcase, and driven from the crankshaft in the usual way, but, instead of being at the bottom of the sump it is mounted comparatively high up, and can therefore be detached without the loss of any oil. It forces oil to the two main bearings, and supplies troughs for the big ends. The cylinders are not specially protected, the ring which secures the gudgeon pin having sufficient scraper action to render this unnecessary. The magneto is driven by a cross shaft, and is situated on the off-side, there being ample ball thrust bearings behind it for the skew gears. Like the crankshaft, the camshaft has only two bearings, and it is solid with the cams.

The Austrian-Daimler 12-15 h.p. is interesting, in that it is the only example of its nationality. Generally, it follows the usual lines of French design, and it differs very materially from the Mercedes designs to which it might be expected to be more closely allied. It is perhaps more properly included in Class B than in Class A, as the pump and magneto are driven by a cross shaft which, though common in the former class, is rare in the latter. Otherwise, it follows much the same lines as several of the engines already described. The lubrication system is quite peculiar, consisting of a pair of piston pumps supplying all the bearings through a drilled crankshaft. The two pistons are actuated through a beam, and are so arranged that no additional valves are needed, as one piston moving faster than the other opens and closes the intake and outlet parts. The exhaust pipe is cast solid with the cylinders and the valves are not enclosed, the tappet bushes being screwed into the crankcase—a most unusual method of attaching them.

CLASS B.—ENGINES WITH A TOTAL CUBIC CAPACITY OF FROM 2,000 cc. TO 3,000 cc.

Figs. I. and II. show both sides of the 12-16 h.p. Lorraine-Dietrich, with the valve covers removed. As regards outward appearance, this is in many respects quite typical of its class. Although not too clearly shown in the photographs, there is a clear space between each of the cylinder barrels, though only a three-bearing crankshaft is used. The carburettor is of the type supplying saturated gas to a mixing chamber, and it does so by means of a half-inch copper pipe, a single jet being situated centrally in the float chamber with an annular float. The duplex water outlet is perhaps not quite so common, and it will be noticed that there is only one intake, this being situated immediately underneath the inlet passage.

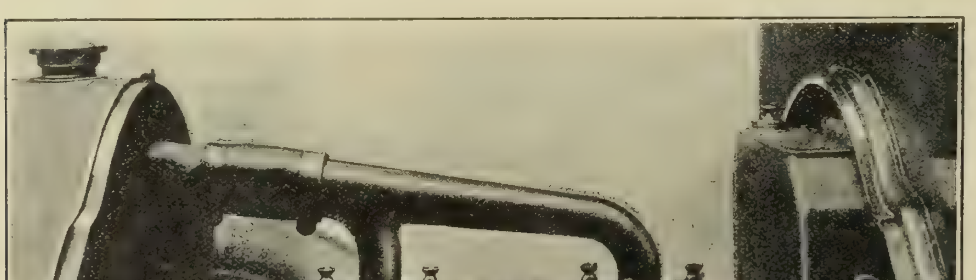


Fig. I. 12-16 h.p. Lorraine-Dietrich engine with valve covers removed and showing spring and lever fan adjustment.

Another French 15 h.p. chassis is the Vinot or Gladiator, and here again the engine is supported by four arms, and has one piece cylinders, but the casting is a little unusual in that it is made completely open at the front and rear ends of the water jacket, closure being effected by plates attached by numerous small screws. The lubrication system is interesting, as it is one of the very few in which the pressure gauge is not in direct communication with the pump, as the oil has to pass through at least one bearing before reaching it. There are separate feeds to each of the three main journals, and drilled passages in the shaft to the big ends. Spray is relied upon for gudgeon-pin lubrication, but there is an external pipe feeding to the bottom of each cylinder, and thus lubricating the pistons directly, as is illustrated in Fig. III. This is so antagonistic to ordinary practice that it is not surprising to find a tap is provided whereby this cylinder-feed may be cut off altogether, and it is normally closed. The pump itself is well above the average size, and should be capable of almost double the normal output for an engine of this size. It is situated in the usual position at the near side rear end of the sump, and driven by very substantial skew gears from the camshaft. As usual, the valves are enclosed by two plates, and the tappets are peculiar having large cup-shaped heads containing small rubber plugs. Over each of these plugs there is placed a steel cap, and the adjustment is supposed to be set so that there is always a little compression of the rubber taking place, thereby keeping the tappets in contact with both valve stems and cams. Cooling is by pump circulation, there being a pump on the off-side driven from the timing gears by an extra pinion, and the pump has a gland on each side, the shaft passing completely through it, and driving the magneto by a universally-jointed extension.

There are but single water inlets and outlets placed centrally on the casting, and the carburettor connects by a single short

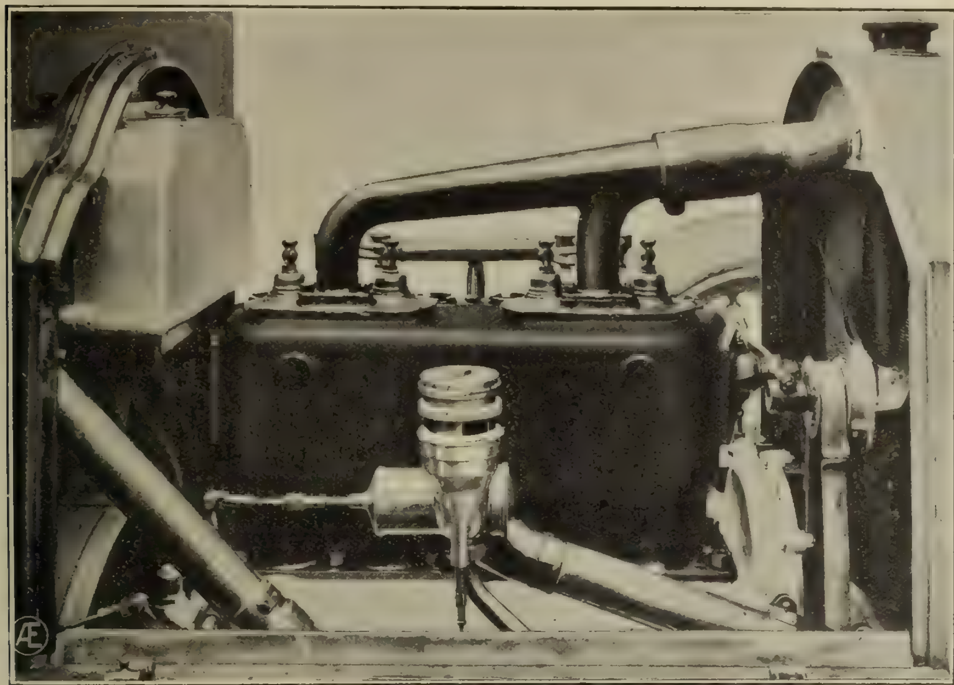


Fig. II. The 12-16 h.p. Lorraine-Dietrich engine.

pipe to the branched passages within the casting. This is one of the comparatively few small engines for which an underframe is provided.

The new type of Itala is of nominal 14-16 h.p., with a bore of 77 mm. and a stroke of 120 mm. This has the usual one-piece cylinder casting, but with centrifugal pump circulation, and a honeycomb radiator. Both the pump and the magneto are on the near side, and the

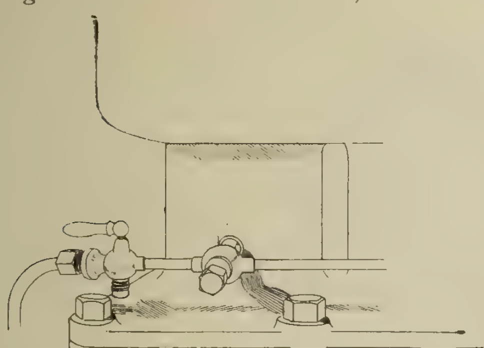


Fig. III. The Vinot oil feed for cylinders.

pump has two glands, a driving shaft passing through and being carried on to the magneto from a universal joint. This design will be found to exist on several other engines, the object presumably being to avoid the use of an additional timing wheel for driving one or other of the two fittings. Seeing that a pump gland is always a potential source of trouble, reproducing it in duplicate does not seem to be very desirable, but in this case both glands are of good size and easy to get at. The valves are exposed, and the tappet guides are bored directly in the cylinder casting. The carburettor is situated in exactly the same position as on the Lorraine-Dietrich, and the water arrangements are similar. The oil pump is enclosed, and carried in the upper part of the crankcase, being slightly above the usual level: longitudinally, its position is beside the middle of the camshaft. The crankshaft is drilled, oil being forced to the three main bearings, and the gudgeon pins are supplied by spray; the sump is rather larger than is common, and the filter is of the ordinary cylindrical type. This is another of the few engines mounted on an under frame, the crankcase having four arms, and the under frame being hung fore and aft with sheet metal guards connecting it to the main frame.

The 15 h.p. Iris, with a bore of 80 mm. and a stroke of 114 mm., has a ball bearing crankshaft with three bearings, as shown on page 19. It will be noticed that the diameter of the bearings is extremely large, and that they are very firmly secured to the crankcase. We understand that this is the first ball bearing engine to be turned out by the manufacturers in question, and that the experimental patterns have been found satisfactory and not unreasonably noisy. We hope to be able to publish details of tests of this engine in *The Automobile Engineer* before long. It should also be noticed that there is a large double thrust bearing to guard against any tendency for the withdrawal of the clutch to disturb the main bearings. The crankcase is provided with exceptionally large inspection doors, in fact, they are sufficiently big to be really useful which cannot often be said of devices of this nature. The oil pump, which sends all its supply to the troughs, is in the usual position at the rear left-hand near side of the crankcase, but instead of being entirely in the sump is external and can be detached without taking down the base chamber. The cooling is by thermo-syphon, the pipes being made solid with the cylinder casting and connected practically direct to the radiator. The carburettor is on the off-side, fitted close up to a single intake port, and is of the type in which the air is drawn horizontally across the top of a single jet. The exhaust passage is also part of the main casting, having a single outlet at the rear end. All other details are given perfectly clearly in the drawing.

In some respects the 15-9 h.p. Thames is one of the most original engines now being made in this country, and, in fact, the chassis as a whole is particularly ingenious. The cylinders are cast together, but the crankshaft has five bearings, and the greatest originality is observable with regard to the lubrication system. The oil pump is illustrated in Fig. IV., and its position is fully explained by the valves which belong to the rearmost cylinder but one. The oil pump is carried in the cast pot as shown, and can be withdrawn completely, together with the shaft and driving gear, by the detachment of a bridge piece which hooks on to the pair of studs also shown, the one behind being raised above its true position in order to show it more clearly. The oil is forced to the main bearings and passes from them to troughs under the big ends, and in addition, there is a by-pass valve for the purpose of reducing the pressure of the main feed.

For big end and piston lubrication troughs are provided, and are filled by the overflow from the bearings, but to ensure their complete supply the by-passed oil, instead of returning direct to the sump, as is usual, is led to a pipe which can be seen in the illustration at the left-hand side of the containing pot. This pipe has been broken off short, but in reality is curved over so that any oil which comes up it is delivered to a duct at the back of the casing whence it runs across the crankcase to a distributing trough on the far side which leads to each of the dip troughs.

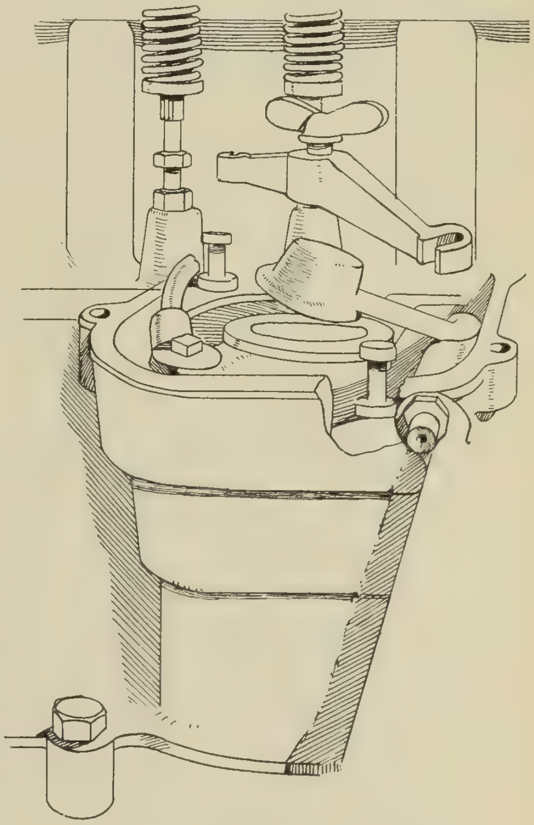


Fig. IV. The Thames oil pump.

It will be realised that the whole pump can be taken out for inspection or adjustment in a very few minutes, and if the crankcase is empty a full supply of oil could be put in in this way in a very short time. The valves are rather larger than is usual for an 80 mm. engine, being 38 mm. in diameter, and their stems are not enclosed. The tap-

pet guides are dropped into recesses in the crankcase in the usual way, but instead of being held down by dogs there is a single casting which is dropped over the tappets after they are in position, and thereby holds down a whole row together. The carburettor is a Polyrhoe, fitted without any inlet pipe right against the offside of the casting. The cooling is by thermo-syphon circulation, with a plain tubular radiator with vertical tubes and a grid facing, which while giving the radiator the appearance of a honeycomb also serves a useful purpose of acting as a radiating surface. The starting handle is carried on a special aluminium casting of a V shape, bolted to the front of the crankcase. The petrol feed is by air pressure, a not very common feature with cars of this class, and there is a plunger pump fixed on the rear-end of the camshaft for this purpose.

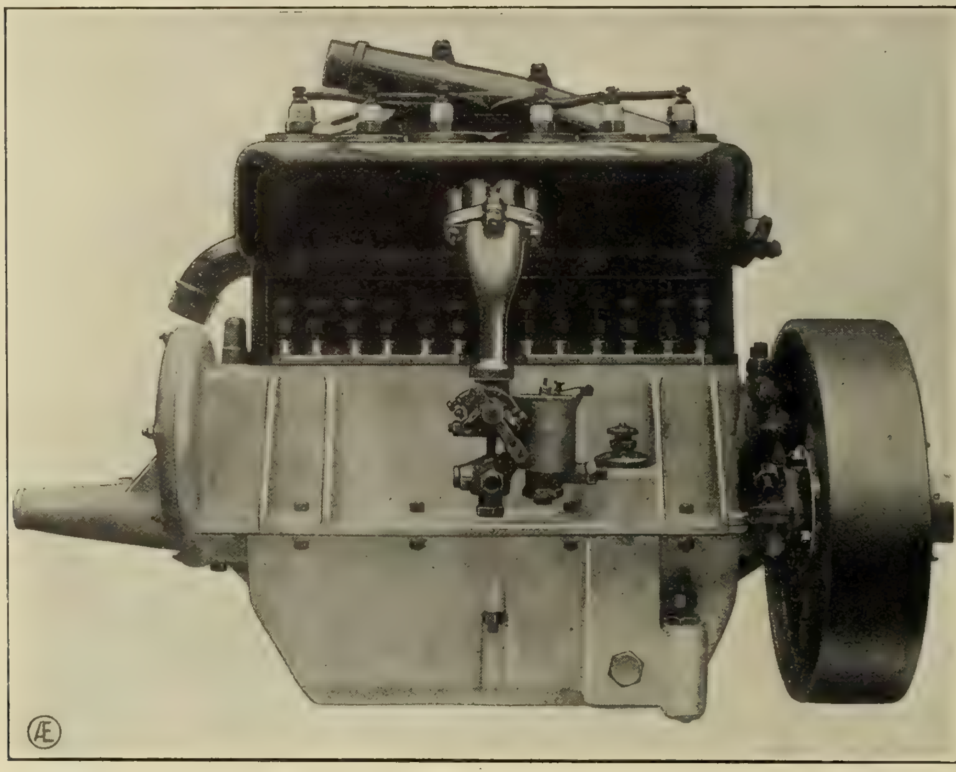


Fig. VI. The six-cylinder Delage engine, showing the multiple mixture intake.

In Class A we have already described the 12 h.p. Martini engine, and would again refer to the 15-8 h.p. as being a good example of French practice. In general outline and particulars it is very much the same as the 12 h.p., save that it is one of the large class of engines with bore of 80 mm. and stroke of 120 mm. The lubrication in this case is not simply by splash, as there are troughs supplied by an engine-driven pump, and in this respect the engine is not typically French. In addition to the troughs there is a pressure supply to the main bearings, the gear pump being mounted in the sump in the usual position and driven from the crankshaft by skew gears. Thermo-syphon circulation also is not deemed sufficient for this larger engine, and the pump is fitted on the off-side end of the cross shaft, the magneto occupying the corresponding position on the near side. The piping arrangements are, however, much the same as on the smaller engine and all other details are precisely similar.

The petrol feed is by means of an air pump on the rear end of the camshaft, and a single cover plate is used for all the valves.

Another interesting engine is the New Arrol-Johnston. This has three main crankshaft bearings, the rearmost being divided into two halves, as the crankshaft carries the skew gear for the oil pump at this point instead of the former being on the camshaft. The pump itself is thus situated almost centrally at the back of the sump, being entirely contained by a large gauze cylinder, and both the filter and the pump are withdrawable from beneath. The oil is fed, by a drilled crankshaft, from each of the main bearings to the big ends, and to protect the cylinders the top of the crankcase is cast with slots only just sufficiently wide to allow the small end to be placed through before the big ends are assembled. The camshaft has loose cams separated by distance pieces, and runs on three ball bearings, there also being a spigot bearing in the outside of the larger timing wheel, as this is some distance forward of the cylinders owing to the length of the front crankshaft bearing. The tappets have divided guides, the upper part taking bearing in hollow lugs cast on the cylinder block and the lower part being provided with the usual gunmetal bush let into the crankcase. This engine is a unit with the gear box casing, and is therefore suspended at three points while the radiator is behind the engine, the draught being assisted by a vaned flywheel and the circulation, of course, is natural. The water pipes are aluminium castings, the exhaust pipe is loose with four junctions to the casting, and the inlet pipe is also carried on the valve side leading to two main ports, being held up, together with the exhaust pipe, by dogs and studs.

The 15 h.p. Valveless, although it is volumetrically to be included in this class, cannot very well be compared with any of the other engines. It is to be presumed that the principle on which the engine works is well-known, and it has undergone no alterations, the system still being to compress pure air in the crankcase, to take up the petrol as the compressed air passes to the inlet port, and to complete the compression in the cylinders. Detail improvements connected with the exact size and position of the ports has enabled this engine to be run at considerable speeds of revolution, it being claimed to give increased power up to 1,800 r.p.m. The balance is necessarily good, owing to the two crankshafts revolving in opposite directions, and such vibration as arises is almost entirely due to variations in torque. The lubrication system is peculiar, there being a plunger pump working in conjunc-

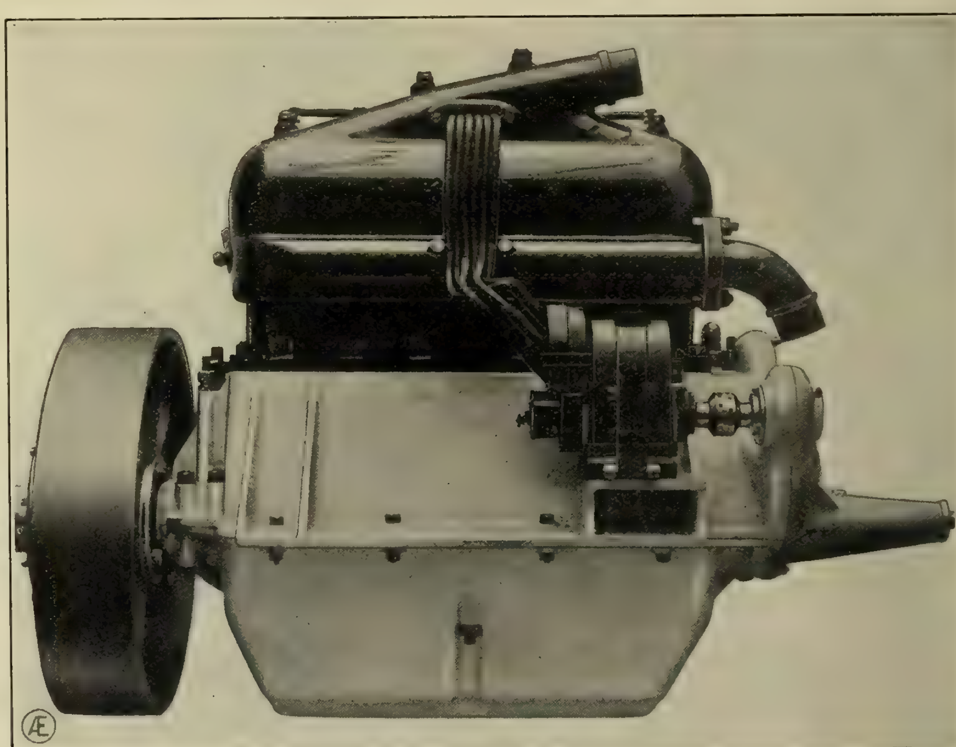


Fig. VII. The six-cylinder Delage engine, showing water pipes and the method of taking the ignition leads to the plugs.

tion with a distributing gear whereby the four main bearings and the gear between the two flywheels are fed in succession. Particulars of the pump are given in the section devoted to details.

Thermo-syphon cooling is used, and the

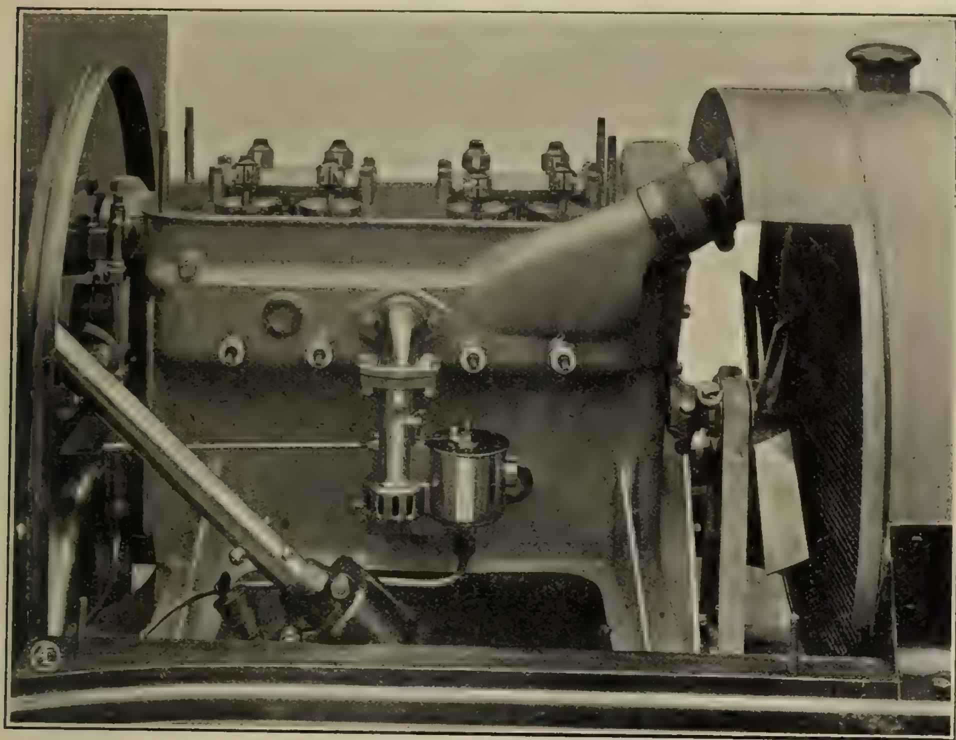


Fig. VI. The new 15 h.p. Germain engine.

dispositions of the two cylinders, of course, makes it an extremely short engine, so it is a little surprising that this has not been taken advantage of for traction vehicle work.

An interesting engine of a quite different type is the small six-cylinder Delage, two views of which are given in Figs. V. and VI. This is one of the few engines having six cylinders cast together and, considering the small total length of the engine, the casting must be an exceptionally awkward foundry job. There are only six valve caps, two valves being withdrawable through each, though the valves are not particularly small for the size of cylinder. The crankshaft runs on three ball bearings, the centre bearing being almost as large as the crankcase in mean diameter, and the housing may be seen to be recessed in a special chamber. A gear pump is situated at the rear end of the camshaft, and, drawing its supply from the sump, it feeds the inside of the crankshaft, which is drilled to the big ends. The water piping is all cast solid with the cylinders, and the inlet pipe is a manifold connected directly to a Claudel carburettor. Though an interesting type, this engine is somewhat too highly original to enable any definite opinions to be formed as to its probable performance in the hands of the average user. Ball bearings are not usually reliable for crankshaft work, but the altogether exceptional size of those in this engine may cause them to be satisfactory. It might reasonably be expected that considerable periodic vibration would occur at certain speeds of revolution owing to the lack of rigidity of the ball bearing crankshaft support.

One of the most novel and also the most interesting engines in Class B is the 15 h.p. Germain, and it is therefore illustrated by Figs. VI., VII., and VIII., showing the two sides and a top view respectively. The camshaft lying along the top of the cylinders is chain driven from the front end of the crankshaft, and operates all the valves through the medium of rockers, the whole eight valves being flat faced and contained in cages very similar to those of the Maudslay, shown on page 20. The ingenious part of the design is the way in which it can be taken down quickly. Each rocker arm connecting the cams with the valve stems has a sleeve, which takes its bearing on the corresponding shaft. There are four of

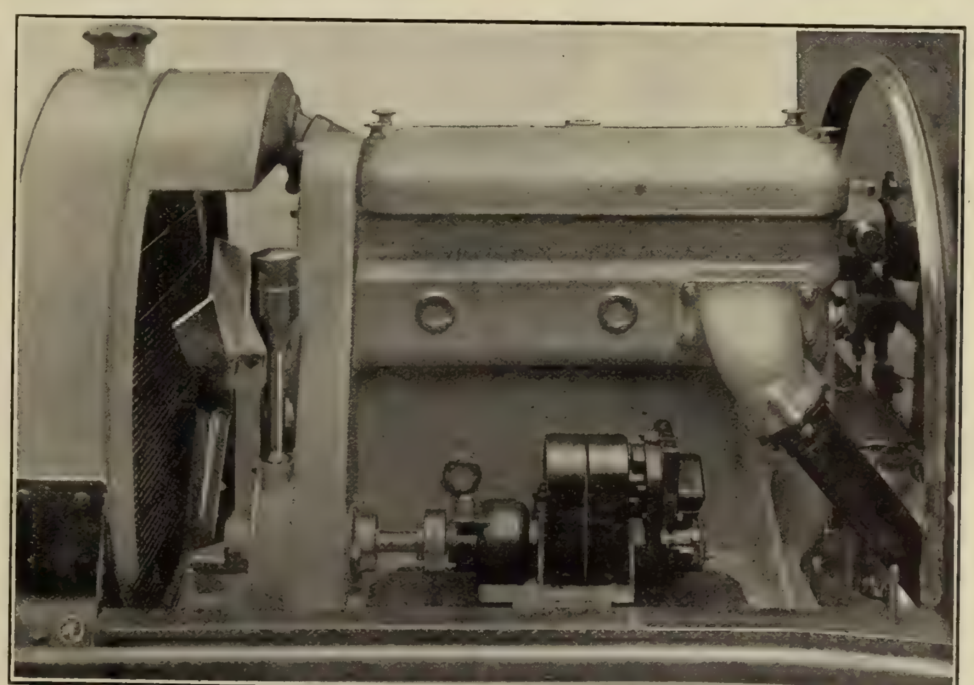


Fig. VII. The new 15 h.p. Germain engine.

these shafts, and they are gripped at the ends by the bridge pieces shown. Detachment of four nuts enables a pair of bridges to be lifted off, and then the valves can be removed with a special spanner. Both the inlet and exhaust passages are, of course, cast-in with the main block, and can be seen in the form of channels at each side. With the aluminium top cover plate in position, the whole of the valve mechanism is not only enclosed, but is running in an oil bath, as the camshaft can be almost covered with oil without ill effects. For the bearing lubrication (there are five main crankshaft bearings) there is a gear pump in the crankcase sump, and this is actuated by a long vertical shaft driven by a skew gear at the

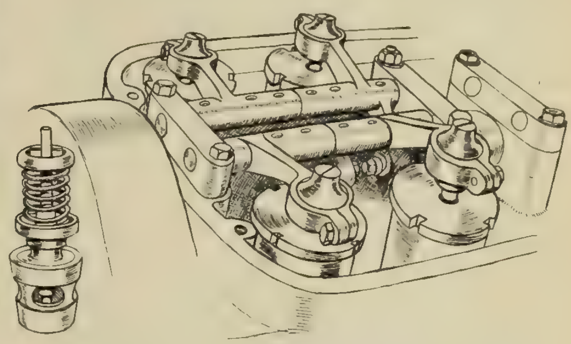


Fig. VIII. The overhead valve gear on the Germain. One valve is shown detached.

rear end of the camshaft. Also at the rear end there is an air pump for supplying pressure to the petrol tank. The oil from the gear pump is, we understand, fed to each of the main bearings, and from them passes to the big ends through drilled holes, and again from the big ends goes to the small ends by external leads on the connecting rods. From the illustrations the method of mounting the magneto is obvious, and it need only be added that it is chain driven and provided with an automatic or governed variation in timing, the governor being a part of the magneto machine. For the cooling natural circulation is not relied upon, the pump being placed correspondingly to the magneto on the off side, and also chain driven. A honeycomb radiator is used, and the fan is also chain driven through a spring coupling.

A Zenith carburettor is fitted on the off side, and the method of feeding it with warmed air is most ingenious, there

being a passage cast right through the cylinder block, and ribbed inside, so as to give the maximum of radiating surface. For external cleanliness for accessibility and for completeness of equipment this engine takes a prominent place amongst new designs, and its behaviour in actual use will be watched with great interest by many. There is no doubt that the form of combustion chamber obtainable with overhead valves is thermally better than the pocketed head, and it is noticeable that few makers who once adopt overhead valves appear to give up the practice. As applied to a long stroke high-speed engine they are a new departure for a touring car, and this new pattern Germain makes a somewhat curious con-

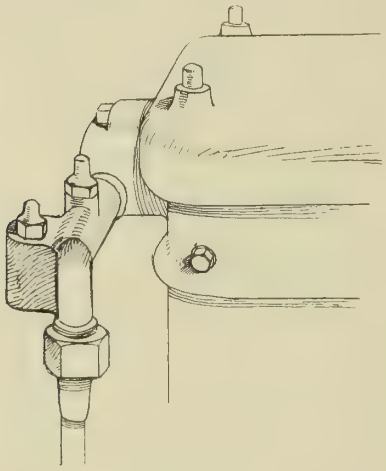


Fig. IX. Air pump on the Germain Car.

trast to the other models with their oppositely-placed valves and steel cylinders, which have now been in use for years, and are still employed for all other Germain types. Also, it may be noted that the crankshaft of this new engine is offset as much as 40 mm.

The 15 h.p. Calthorpe engine, which has one of the biggest stroke/bore ratios to be found amongst British cars, has one or two interesting details, although it follows standard practice in generalities. The bore is 75 mm., and the stroke 150 mm., and the cylinders have the appearance of a monobloc, but are actually cast in pairs, and bolted up together, somewhat similarly to the Maudslay engine, and not altogether unlike the 18 h.p. six-cylin-

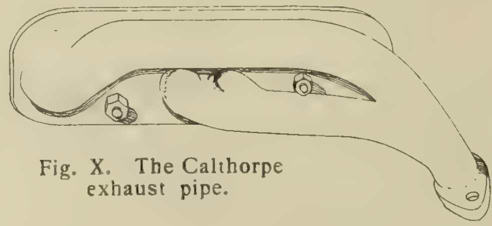


Fig. X. The Calthorpe exhaust pipe.

der Panhard. The exhaust pipe is shown in Fig. X., in diagrammatic form, and it will be seen that with an order of firing of 1, 3, 2, 4, or 1, 2, 4, 3, no two exhausts follow each other in the same pipe. This is claimed to have a noticeable effect on the back pressure, and therefore on the efficiency of the engine. The carburettor, which is White and Poppe, is carried on the other side of the engine, feeding two passages through a cast V pipe, while another good feature of the engine is the five main bearings which are lubricated by a plunger pump, the latter also supplying big end troughs.

The 16 h.p. Adams engine, shown in Fig. XI., has cylinders of a rather unusual shape, cast in pairs, with large openings at each end of the valve pocket portion of the castings, closed by screwed-on plates. The valves are exposed, and the whole of the piping is carried on the near side, the carburettor lying in a deep channel, which is formed between the crankcase arm on each side of the engine. Fig. XII. shows the position of the oil pump rather more clearly than the first view, and it will be seen that it is driven from the extreme end of the camshaft, while it is considerably less inaccessible than usual. This second view also shows the way in which the carburettor is set in the casting. Oil is supplied to the main bearings, and also to troughs, the feed channels being internal. The cylinders are slightly off-set, and the valves normally exposed, cooling being by thermo-syphon, with a vaned flywheel.

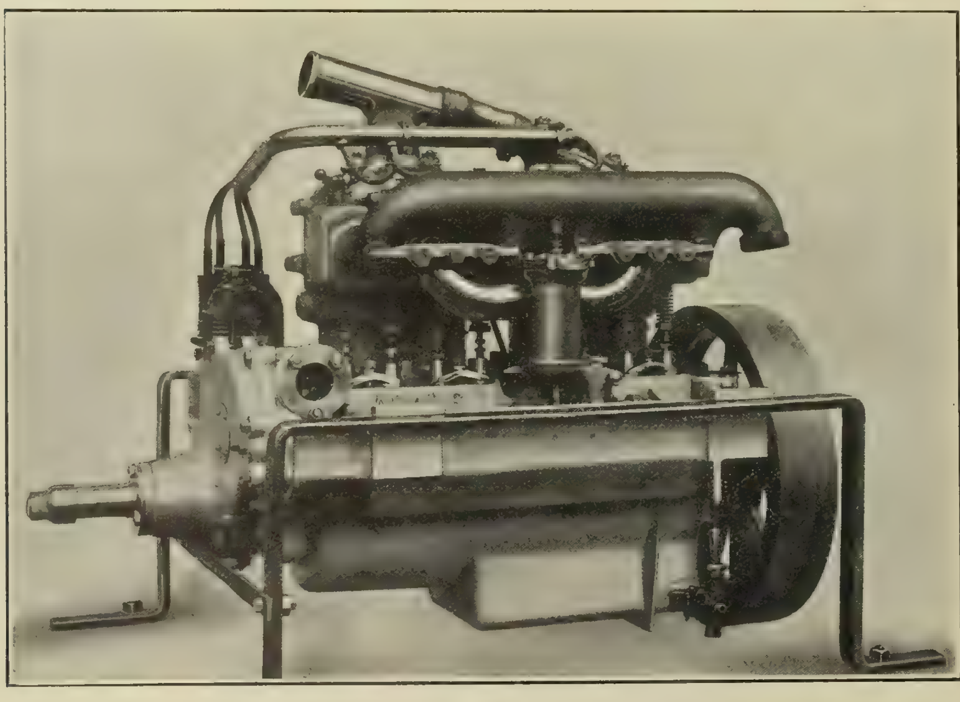


Fig. XI. The 16 h.p. Adams engine.

The 17-9 h.p. Armstrong-Whitworth, together with the Adams just mentioned, belong properly to a sub-division of Class B., as the 85 x 120 mm. engines are distinctly more powerful than the 80 x 120, notwithstanding the comparatively small difference in capacity. The Armstong engine was described in complete detail in

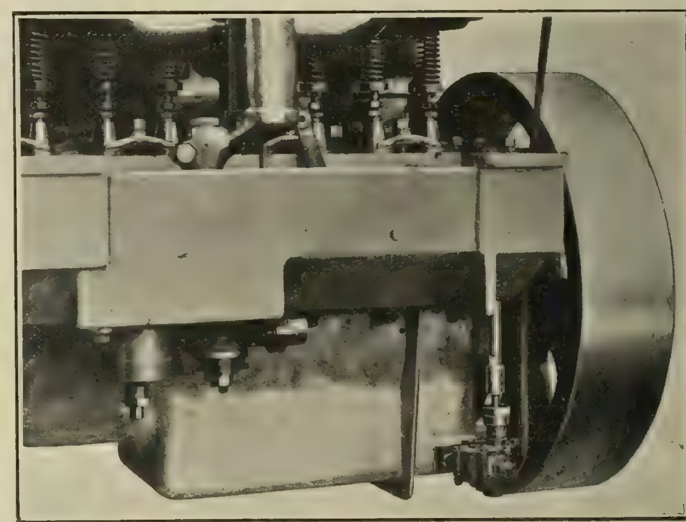
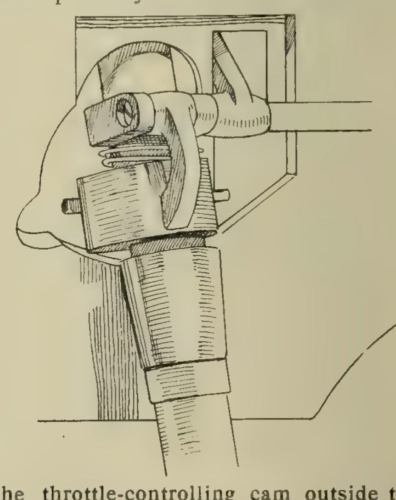


Fig. XII. The arrangement of the Adams oil pump.

the last issue of *The Automobile Engineer*, and will therefore not be dealt with again here, being merely mentioned in passing as a good example of modern British design.

The 15 h.p. S.C.A.T. again has a one-piece cylinder casting, and in this case the exhaust pipe is also cast in. Cover plates enclose the valves, but they are held in place by set-screws instead of the



The throttle-controlling cam outside the steering column of the 16 h.p. Adams.

more usual fly nuts, sharing this peculiarity with the Armstrong-Whitworth and several others. The magneto and the water pump are both on the valve side, and are placed one above the other, similarly to last year's 15 h.p. Napier arrangement, only in this engine the timing gears are at the front of the crankcase instead of at the rear. Oil is forced to all

but the gudgeon pins by a gear pump situated in the usual place, but driven by a coil spring external to the crankcase, instead of by the usual enclosed shaft. There is an adjustable by-pass to allow the oil pressure to be regulated, and the customary three-way filling, draining, and level cock. A two-jet carburettor is fitted on the off-side, and feeds a single entry port in the casting. The most interesting feature of the engine is naturally the self-starting attachment, which, as may be remembered, consists of an air-cooled ribbed cylinder pump mounted on an extension of the crankcase in the forward direction, which is used to charge a compressed air bottle. When needed for starting, special valve gear is brought into operation, and the air is fed to the main cylinders of the engine, causing it to revolve when, as soon as sufficiently rapid rotation has been obtained, the valves are returned to their normal setting, and the engine commences to run on its own power.

The 15 h.p. Bentall is an example of a totally different type, being one of the few small engines with separate cylinders. Curiously enough, it shares with the Vinot the peculiarity of taking the magneto drive right through the pump. The valves are exposed, and placed oppositely there, of course, being the necessary two camshafts. This engine has an interesting built-up crankshaft, the sections consisting of crank-pin pieces and mainshaft with web parts. Each piece is separately hardened and ground before assembling.

CLASS C.—ENGINES WITH A TOTAL CUBIC CAPACITY FROM 3,000 cc. TO 4,500 cc.

The 20 h.p. Vauxhall has been altered very little, the changes being principally matters of more or less small detail. The valves are exceptionally large for a 90 mm. engine, being over 40 mm. in diameter, and the cams are slightly convex, the profile being such that the valve is lifted very rapidly, but is not allowed to shut

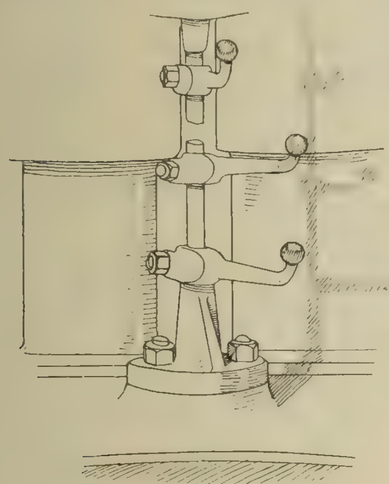


Fig. I.

violently, the concave flank allowing the tappet to drop very rapidly for the greater part of its fall, but catching it near the bottom, and slowing the motion very considerably during the last fraction of an inch. The valves are now all enclosed by a single cover plate, and the crankcase arms have been connected by the usual

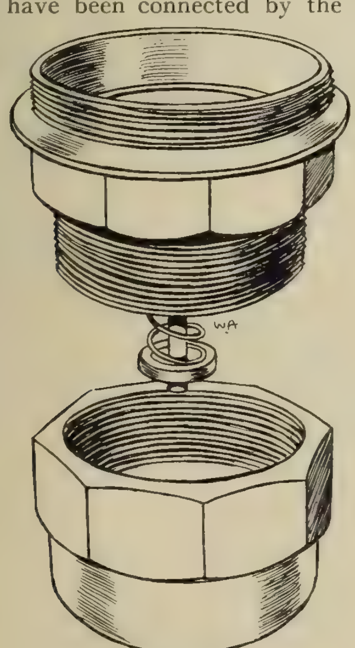


Fig. II.

webs. The control is very neatly fitted thereto, as there is a special lug cast on the centre of the tray on the off-side to which what might be called a distributing bracket is fixed, the throttle and ignition control levers being divided from this point. (See Fig. 1.) Another alteration is an improvement in the lubrication system by the addition of a dirt trap. It may be remembered that there is a plunger pump driven from a crank at the extreme rear-end of the camshaft, which sucks from the sump and forces oil, under very considerable pressure, from a drilled crankshaft to all the bearings. Immediately beneath the pump intake there is a large brass pot screwed into the crankcase, and containing a mushroom valve, which is held open when the pot is screwed home, but closes when the latter is loosened, the idea being that as it is the lowest point in the whole circulating system (see Fig. II.), impurities will gradually accumulate therein, while on unscrewing the pot, the bulk of the oil is retained in the sump by the closing of the valve, and the oil can thus be cleansed without loss. A small point in connection with this device, not unworthy of mention, is that it is designed to take the usual hub cap spanner. As before, there is a single intake for the carburettor, which is a White and Poppe, and is situated on the off-side, while a small plunger air pump is situated immediately above the oil pump to supply tank pressure. The exhaust pipe is loose and attached by flanges, while the cooling is by thermo-syphon, with a vertical gilled tube radiator.

The 15 h.p. Star, with the same bore and stroke as the Vauxhall, is another example of a high-powered engine, and was described in full detail in *The Automobile Engineer* for November. The sectional drawing published in that issue is still correct, so only a few of the main points need be re-stated. The crankshaft is exceptionally heavy, and is offset $\frac{3}{8}$in. from the cylinders, which are cast in pairs with four exhaust outlets and two intakes. The water spaces are of good size, as is the honeycomb radiator, but a pump is used situated high up on the near side of the engine. The valves are $1\frac{1}{2}$in. in diameter, with a $\frac{3}{8}$in. lift, and the normal cam is flat-faced, while the oil pump is carried on the rear end of the camshaft in a most accessible position. Normally, the lubrication is by filling the two halves of the crankcase, the surplus overflowing through the centre opening, and so passing to the sump, being filtered on the way and again filtered at the pump intake, but a completely forced system with a drilled crankshaft is fitted alternatively to this system. The timing gears are exceptionally wide, having $1\frac{3}{8}$in. faces. This is one of the extremely few engines in which there is no special provision for holding the gudgeon pin, it being merely made a light driving fit. The total length of main bearing is 9ins., the big end being $2\frac{1}{4}$ins. wide, while the diameter throughout is $1\frac{5}{8}$ins., except for the webs, which are 2ins. wide. The White and Poppe carburettor is carried on the off-side, the inlet pipe passing between the cylinders and being held up by central set-screws in the dogs, which retain the exhaust pipe.

The 15 h.p. La Buire 85 mm. × 140 mm. engine has several peculiarities, one being that the camshaft is chain driven, the magneto and pump being situated on the near and off-side ends respectively of the cross-shaft. The valves are enclosed, the usual tray-form arms are provided, and a peculiar type of ribbed exhaust pipe is used, shown in Fig. III. The oil pump is situated in the sump, driven by a diagonal shaft from the camshaft, and forces oil to a reservoir in the top of the crankcase, whence

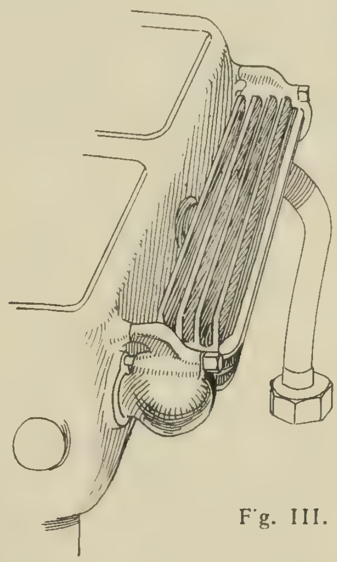


Fig. III.

it is taken by ducts to all the main bearings, their overflow filling the troughs. There is an unusually large filter placed in the customary and somewhat inaccessible position. It has been mentioned that, like the Star, a pump is used, but in this case there is only a single intake and outlet for the water, the former being on the near side, between the two middle cylinders, and the latter in the centre of the top casting. The inlet passages are two in number, and pass between the front and rear pairs, the V-shaped inlet pipe being secured by a single stud and nut, the arrangement being exactly similar to that used for retaining the valve caps on the 18 h.p. Thornycroft, as shown in an illustration on another page.

The 17 h.p. Maudslay, which was described in *The Automobile Engineer* for

September last, has undergone no alteration whatever, but for purposes of reference it is illustrated on another page. It is one of the extremely small number of engines with entirely overhead valves, and the general details are shown sufficiently clearly by the drawings. The cylinders are cast in pairs, troubles at all worthy of mention.

The 18-24 h.p. Sunbeam, which is illustrated in Figs. IV. and V. is a specially interesting engine, as it is an endeavour to combine the features of the small high-power type of engine with the smoothness generally attributed to six-cylinder construction. The bore is 80 mm., and the has an important influence upon the liability of the latter to periodic vibration. A curious point in the design is the fact that the two camshafts are not in the same plane horizontally, which is due to the off-setting of the crankshaft by 15 mm., and also to the necessity for providing an oil channel above the camshaft on the near

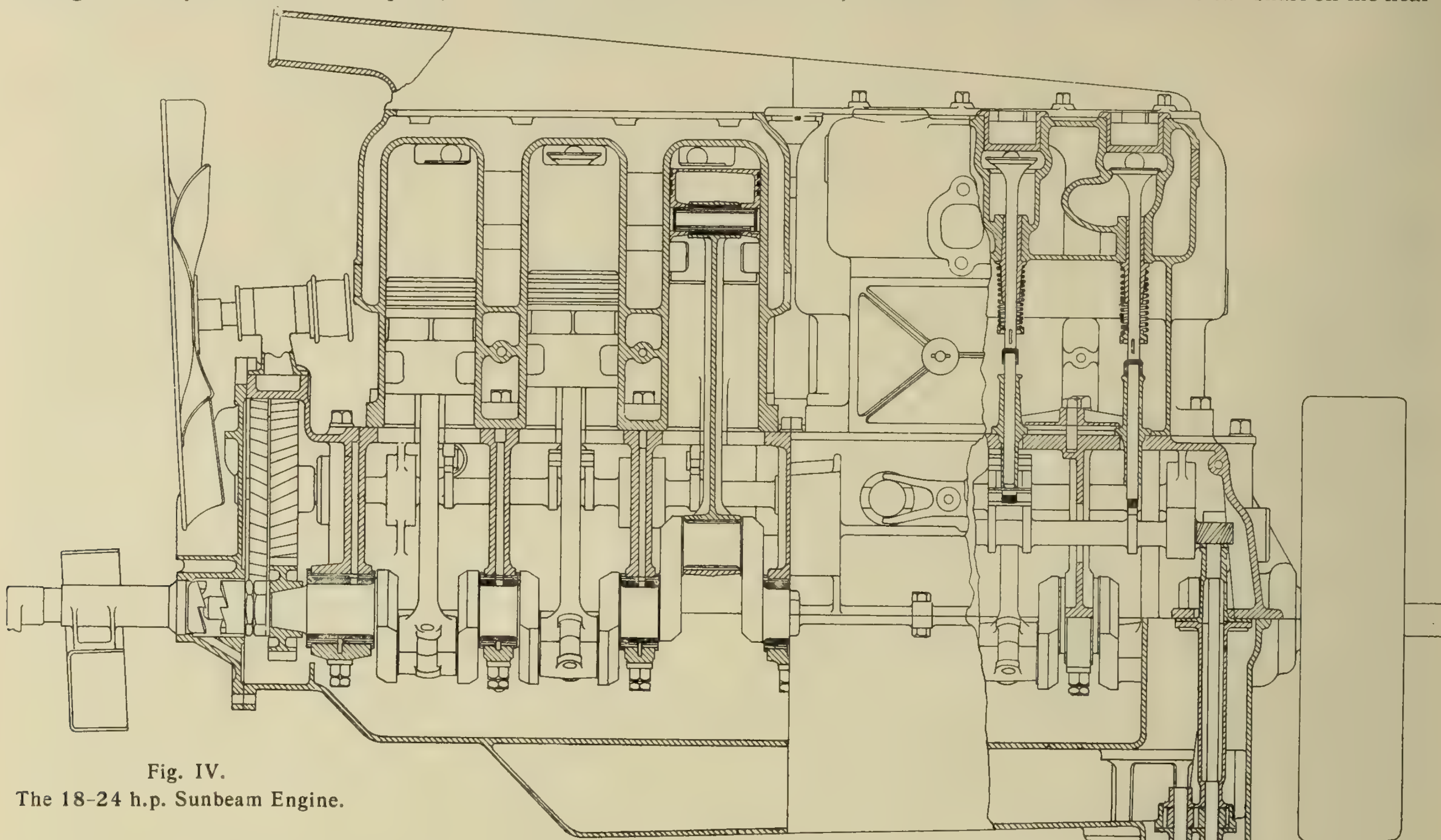


Fig. IV.
The 18-24 h.p. Sunbeam Engine.

being more or less rectangular in shape, and entirely open both front and back, making the removal of the core very easy, and also enabling the ends to be faced, the two pairs being then bolted together to form a single block. The plates to close both the front and rear ends are made to serve for other purposes also, the rear one carrying the magneto platform and the front end, the fan bracket, and some other small fittings, so the use of the plates, although they are in themselves an unnecessary addition, results in the elimination of a variety of other parts. The overhead camshaft is driven by a vertical shaft with a slotted ball type of universal joint at the top, and the same shaft also drives the oil pump at the bottom, oil being forced to each of the five main bearings, and thence passing through the crankshaft to the big ends, and from them to the gudgeon pins by way of an external copper pipe in one design, or through a Lanchester type hollow connecting rod in another modification. The engine is naturally quiet, owing to the entire absence of spur gears, and the parts are all very accessible, as the camshaft can be swung over to give easy access to the valves. It is carried in the frame in an unusual manner, the crankcase having a nose piece at the forward end, which can swivel in a hanger clipped to a tubular cross member of the frame, while a steel pressing forming another cross member is bolted up to the back end of the case. This makes the engine extremely easy to fit in the frame, there being no lining up

stroke 120 mm., the cylinders being cast in blocks of three, a system which seems likely to become universal for small six-cylinder engines. It may be noticed that the pistons are cut away in the skirt, to give lightness, and the holes also have the effect of scraper rings, allowing any surplus of oil on the cylinder walls to return to the inside of the piston, whence it drops back to the sump. The shape of the valves is also noticeable, this being typical of modern racing car practice, and it may be mentioned that the effective diameter is 40 mm. The solid camshaft operates the tappets through rockers, and this is peculiar because there are now extremely few engines with this form of construction. The rockers themselves can be seen in the transverse section, while the caps in which their pivots are retained are also shown in Fig. VI., this being a view from the outside of the crankcase. The valves are placed oppositely, but are enclosed, which is a most unusual practice, except when the valves are on the same side. A good point is the seven main bearings, because, although they are narrow, the intermediate ones being only 30 mm. wide, the amount of support which they give to the shaft

side. It will be noticed that the cams themselves are of the flat top concave type. In the sectional elevation the very

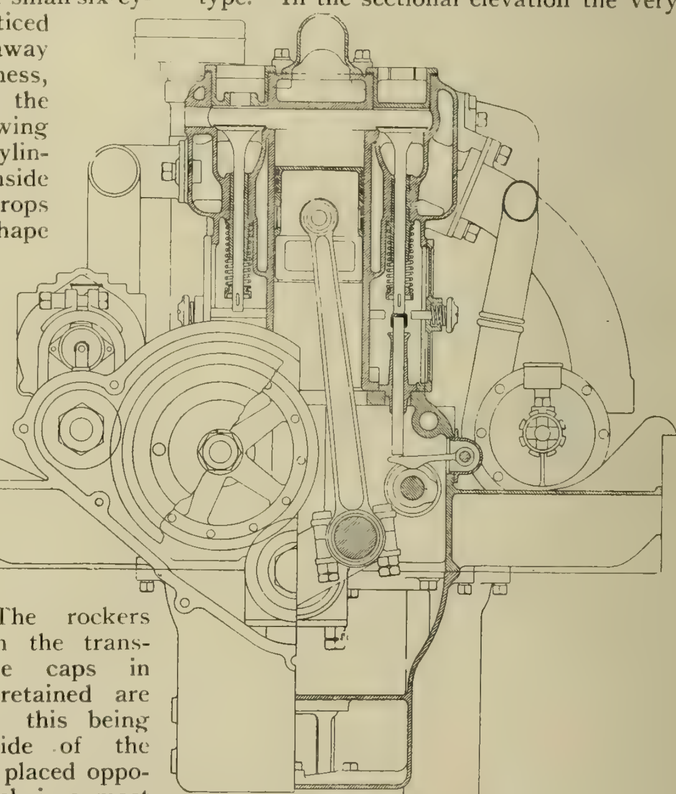


Fig. V. Transverse section of six-cylinder Sunbeam engine.

large water spaces are noticeable, and, when it is remembered that the cooling is by pump circulation, it will be obvious that it is unusually complete. The mag-

neto is carried on the off-side of the cylinders, to which it feeds by a two-branch pipe, there being a single intake in each cylinder block. The exhaust pipe is, of course, on the near side, while the carburettor used is a Claudel. The water circulating pump is of the gear pattern, now not at all common, as the majority of water pumps are centrifugal. It is situated on the near side of the cylinders,

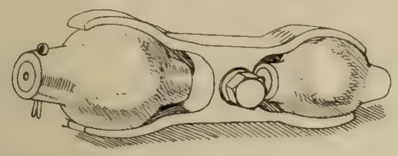


Fig. VII. Sunbeam valve rocker covers.

and is driven by an unusually long universally jointed shaft, from a separate timing gear, the magneto being arranged correspondingly on the off-side. Lubrication is by gear pump situated in the usual

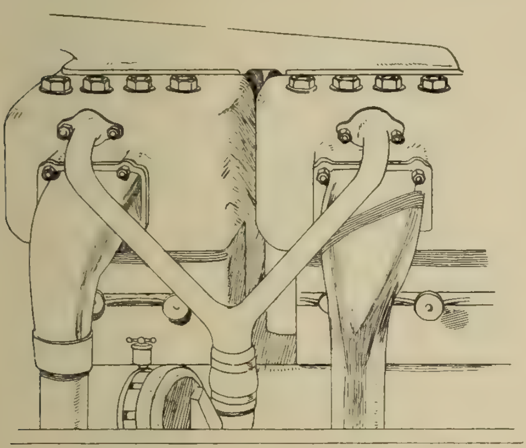


Fig. VIII. Sunbeam exhaust pipe arrangements.

position in the sump, forcing oil to each of the main bearings through a cast-in channel in the crankcase, and from the main bearings by a drilled crankshaft to the big ends. The filter and pump are enclosed in a cylindrical gauze, and both filter and pump are detachable from beneath. Fig. VII. shows the exhaust pipes on this engine, and it will be seen that they are of unusual proportions, having a rectangular shape at the point where they are attached to the cylinders, which gives an extremely free outlet. The radiator is a square tube honeycomb, and a spring-controlled jockey pulley maintains the tension on the fan belt.

The six-cylinder 23-9 h.p. Arrol Johnston is another car with the 80 mm. bore and 120 mm. stroke, and therefore forms an instructive comparison with the Sunbeam we have just described. As regards general arrangement this engine follows the line of the 15-9 h.p., which has already been mentioned, that is to say, it is built up on the unit system with the gear box, and has precisely the same lubricating arrangements. The cylinders, however, instead of being in one piece, are cast in pairs, and a triple manifolded pattern of divided inlet pipe is used. Although this unusual design of pipe has the disadvantage of slightly obscuring the valves, it is probably the ideal arrangement for a six-cylinder engine, as it gives an almost exactly equal effective length of piping between the carburettor and each of the inlet valves. It is made in cast aluminium, and secured by flanged joints. Exactly the same system of cooling is used as on the 15-9 h.p., the radiator being on the dashboard, and the circulation by thermo-syphon. Each pair of cylinders is cast with tubular jacket extensions, so that simple rubber joints are all that is necessary to connect the pairs, and the outlet is by means of a single aluminium casting, the size of the openings between the pipe and the cylinder castings being so proportioned that an even flow of water is ensured through each of the three jackets, there being no tendency for the rear pair of cylinders to pass more water than the other two.

The 28 h.p. Lanchester has a bore of 101.6 mm., and a stroke of 76.2 mm., with six cylinders, and is one of the last remaining few engines in which the bore exceeds the stroke. The reason for this proportion is two-fold, being partly that the makers consider smooth running to be one of the most important features of the car, and that this desideratum can best be obtained by the use of a short stroke, and partly because the position of the Lanchester engine between the driver and the front passenger necessitates a narrow crankcase, which in turn means that the crank throw must be small. Cylinders cast separately are used with horizontal valves actuated by flat springs. Extremely little alteration has been made since we described this engine in *The Automobile Engineer* in the issue for last August, probably the most striking change being in the situation of the oil filter, which is shown in Figs. X. and XI., and these completely illustrate the method by which it may be detached for cleaning. The lubrication system is a completely forced one, oil being supplied to each of the seven main bearings, taken thence to a drilled crankshaft to the big ends, and passing up them through hollow tubular connecting rods to the gudgeon pins. The pump is of the gear pattern, driven from a spur gear situated just in front of the rearmost engine bearing, the crank web at this point being made of disc form, so that the wheel can be bolted to it easily. The pump gears have a width equivalent to between three and four times their diameter, the latter dimension being normal, so the quantity of oil passed per revolution is very considerable, and, as the pump runs at twice the speed of a camshaft-driven pump, it is easy to believe that the claimed bearing oil pressure of 40 lbs. per square inch can be attained. The filtering is duplicate, the oil being both strained through a tray which covers the lower half of the crankcase, and passed through the detachable filter illustrated, on the way to the pump. Cooling is by thermo-syphon, the radiator being in two separate halves situated side by side in the dashboard at the front of the engine, and the fly-

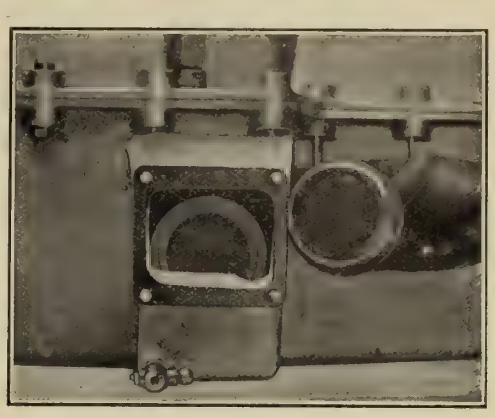


Fig. X. The Lanchester oil filter.

wheel comes immediately behind the radiator at the front end of the crankshaft, as the gear box is bolted up direct to the crankcase. The camshaft is partly exposed, and may be seen between each of the cylinders, while the carburettor is on

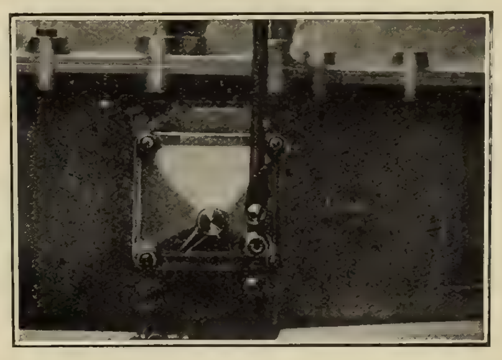


Fig. XI. The Lanchester oil filter, closed.

the same wick principle as the earliest Lanchester design. This description, it may be said, applies to the other two types of Lanchester, the only differences being a matter of size, as the details are very

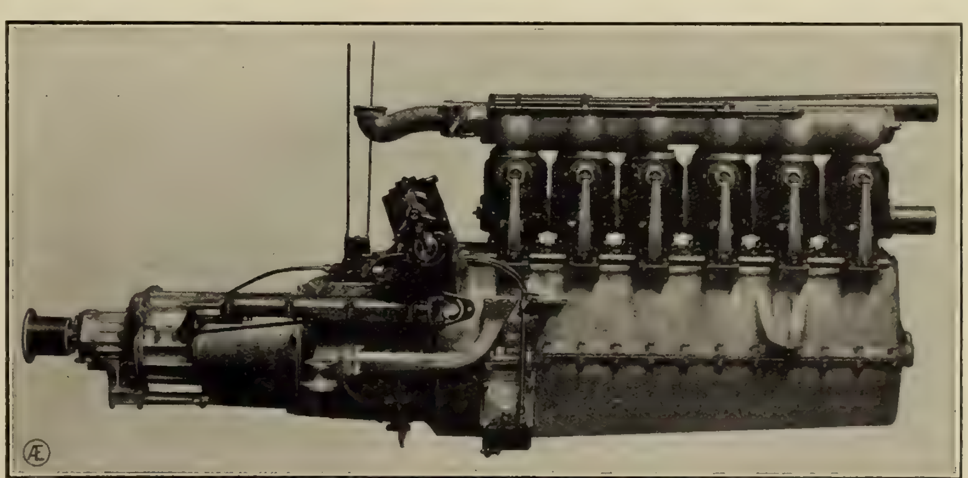


Fig. IX. The 28 h.p. Lanchester engine and gear unit.

nearly precisely similar for the 20 h.p. four-cylinder, the 28 h.p. and the 38 h.p.

The 18 h.p. Thornycroft, which is the next largest car in the list, has a bore of 101.6 mm., and a stroke of 114.3 mm., the difference in volume being a little over 30 cc. Although the stroke/bore ratio is very small, it is positive in this case, and the engine does not resemble the Lanchester in the smallest degree. The cylinders are cast together, and have enclosed valves arranged on the same side,

with a ribbed loose exhaust pipe and an inlet pipe on the off-side leading to two passages through the casting, the carburettor being mounted exceptionally high up. The flywheel is noticeable owing to its being a regular marine pattern of small diameter and great width, the latter dimension being about equivalent to the radius. In this engine the tappets are carried in a peculiar way (see Fig. XIII.), there being a detachable portion of the

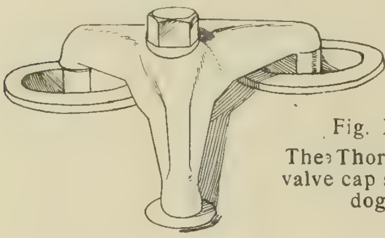


Fig. XII. The Thornycroft valve cap securing dog.

crankcase in which they are mounted, this forming a lid by the removal of which the whole of the camshaft may be exposed if desired. The valve caps are also retained in a unique way by means of a dog held by a single stud, and taking a bearing by one arm on the cylinder casting with the other two resting on adjacent caps, as shown in Fig. XII., this being essentially the same principle as that by which the La Buire inlet pipe is held up, as mentioned previously. Lubrication is com-

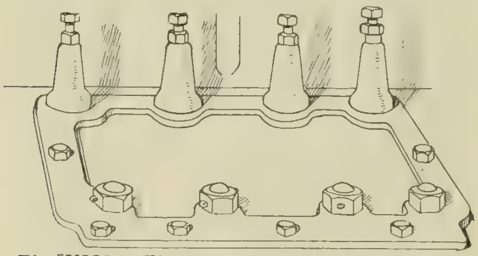


Fig. XIII. Thornycroft valve gear covers.

pletely forced by a gear-driven pump carried in the sump in the usual position, and feeding the five main bearings, whence the oil passes to the big ends through the crankshaft, and up the connecting rods to the small ends, but the pressure is not so high as that used by the Lanchester, the pump being much smaller and running at a lower rate of speed.

The 26 h.p. Minerva, with a bore of 102 mm. and a stroke of 150 mm., may be said to correspond to the 25 h.p. Daimler, though the stroke/bore ratio is slightly smaller. The general arrangement of the engine is precisely the same as regards the whole of the parts, with the solitary exception of the lubricating system, which is possibly slightly more neatly carried out. In both cases the whole of the oil distribution takes place from troughs, whence lubricant is splashed to every part of the engine, but the alteration for this year in both the Daimler and Minerva designs has been to make the level of the troughs adjustable, so that as the engine speed increases with throttle opening, the scoops dip more deeply, and so throw up a greater quantity of oil. In the Daimler, the troughs are separate castings, and are hinged on the off-side by a rod which passes from end to end of the crankcase, while on the near side there is another rod with a toggle lifting gear. The end of this rod passes through the crankcase, and carries a lever, from which a connection is taken to the throttle, so that as the throttle is opened the rod is turned, and the ends of the troughs are raised from a depressed towards a level position. In the Minerva system the troughs are mounted on a rod, to which they are pinned, while the outer ends are free; the rod passes right through the crankcase as in the Daimler, and it is at once obvious that rotation of the rod alters the inclination of the troughs. Cast on the sides of the crankcase are small cups, which are fed simultaneously by an external gear oil pump, and from these cups oil passes through a duct into each of the troughs, whatever their inclination may be. It will thus be seen that the Minerva device has a rather smaller number of parts, while giving almost exactly the same motion to the troughs. For the whole of the Knight engines now being made, pump circulation is provided, it being necessary, owing to the separated water jackets rendered essential by the loose head construction, and by the fact that the water in the centre of the heads lies at a lower level than the water which surrounds the outside of the top of the sleeves. In the Minerva and Daimler designs, there is a cross-shaft, driven by a skew gear from the crankshaft, and passing right across the crankcase, it being situated at the front end of the engine, and an extra hand-controlled air taking its supply from the crankcase, and so carrying a certain amount of oil to the sleeves when the engine is working at high pressure. Details of the mounting of the Mercedes engine, and of the nature of the radiator, are not yet available, but for the Daimler and Minerva a four-armed support is used, and the cooler consists of a vertical gilled tube in the latter case, and a vertical un-gilled flat tube in the former.

The 25 h.p. six-cylinder Sheffield Simplex is another example of a comparatively small six-cylinder, the bore being 85 mm. and the stroke 107 mm. The most distinctive feature of this engine is, of course, the camshaft drive, which is performed entirely by skew gears, although the camshaft and crankshaft are parallel. On the crankshaft there is a small worm wheel with a 45° angle of tooth, and there is a similar wheel twice the diameter, and with twice the number of teeth, on the camshaft. Between the two there is a worm or wide skew pinion (whichever phrase is preferred), and this acts in precisely the same manner as an intermediate gear in any other train. This

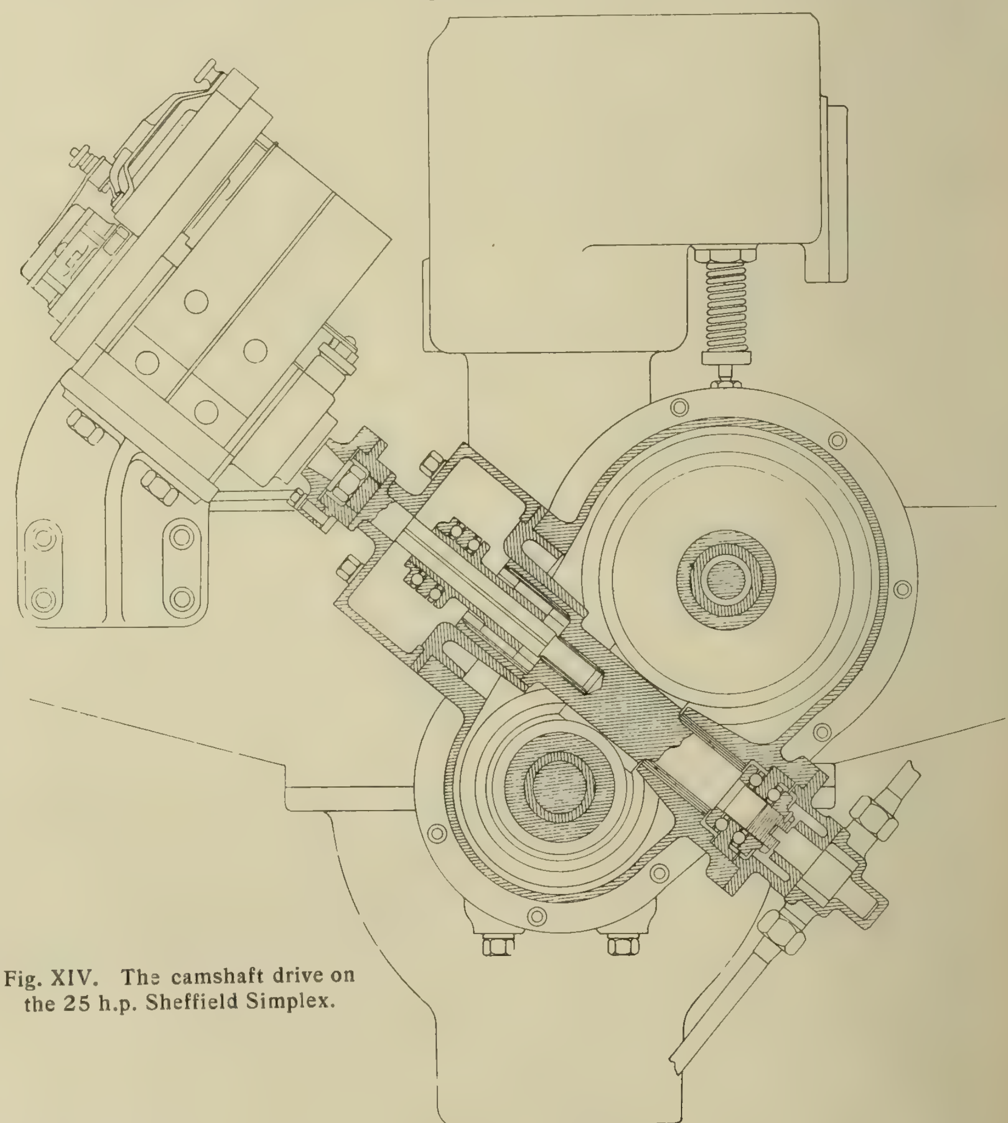


Fig. XIV. The camshaft drive on the 25 h.p. Sheffield Simplex.

sometimes between the front pair of cylinders. In the Mercedes-Knight engine, however, the camshaft, instead of being driven by a chain at the rear end, is driven through a gear at the centre, and the pump and magneto are then placed parallel to the crankshaft, being driven by a separate gearing. The carburettor for both the Daimler and Minerva is a two or three-jet pattern, with automatic air increase, construction therefore entirely eliminates spur gearing without the use of a chain, nor is the intermediate wheel simply an idler, because it is utilised to drive the magneto and the oil pump, it being inclined, as shown in Fig. XIV. Reference to the large view on another page shows the very thorough manner in which the worm wheels are mounted, and also explains the lubricating system, which is

forced to the main bearings, overflow thence running into troughs. Both scraper rings and baffles cast in the top of the crankcase are used to protect the cylinders, and the gudgeon pin is also ring secured, so it may assumed that the amount of oil which is allowed to be thrown by the scoops is exceptional. The pump, being on the lower end of the skew shaft, is, of course, external to the crankcase, and is much more accessible than it is on most designs.

CLASS D.—ENGINES WITH A TOTAL CUBIC CAPACITY FROM 4,500 cc. TO 6,000 cc.

The 22 h.p. Scat is in general respects similar to the 15 h.p. which has already been described. The cylinders are cast in pairs, the water intake and the gas intake being arranged in the ingenious manner shown in Fig. I. The passages are brought

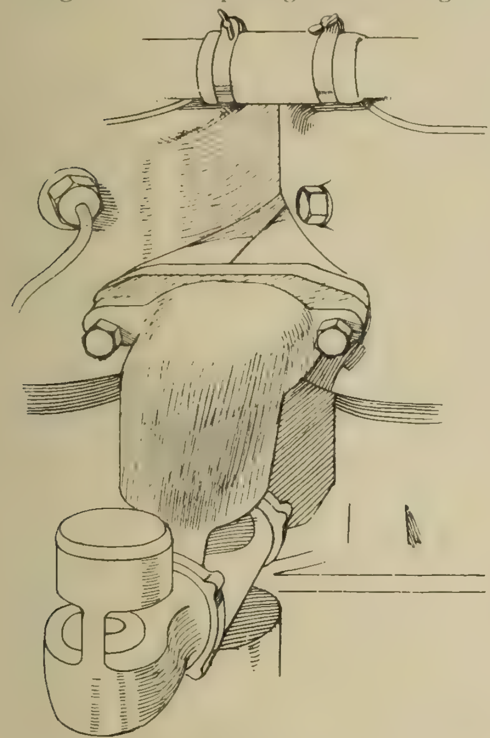


Fig. I. The divided inlet pipe on the 22 h.p. S.C.A.T.

to an end on the off and near sides of the cylinders respectively, and are D-shaped, the two D's being arranged with their flat sides practically touching. The external pipes have single flanges, and the customary two studs, one in each casting, are used to hold them in place, there being a rib or bar across the face of each flange provided with a very fine V-shaped rib projecting above the level of the face of the flange. Thus when the flange is bolted up, this small rib is forced between

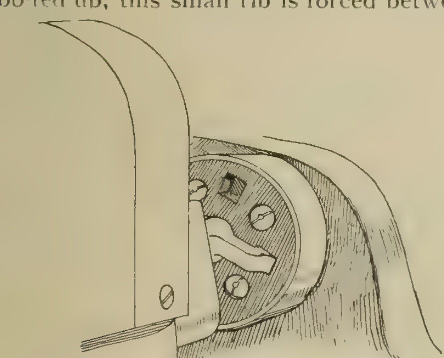


Fig. II. Rubber-block magneto drive on the 30 h.p. Vauxhall.

the two castings, making a gas-tight or water-tight joint. A three-bearing crankshaft is used, the webs being arranged to take a bearing on the ends of the bushes, and the fitting in this respect is performed very carefully, so as to prevent any longitudinal play on the shafts. The same self-starting apparatus is fitted as on the 15 h.p., and the lubrication is likewise forced, by means of a gear pump carried in the usual position low in the sump.

The 20 h.p. Vauxhall we have described already, and the 30 h.p. six-cylinder, with a bore of 90 mm. and a stroke of 120 mm. is very much the same, except that the cylinders are cast in blocks of three. On this engine, also, an air pump is arranged to be driven from the same crank as the oil pump, being situated immediately above the latter. The engine as a whole is striking, it being particularly neat, even for a modern six-cylinder, while the valve arrangement and the cams are an exact reproduction of the highly-successful smaller engine.

The Rochet-Schneider engine is chiefly interesting as being an entirely new design of an old-established Continental firm. Generally speaking, it differs very little from the standard form of design, except that there is a five-bearing crankshaft, and the oil pump, though situated in the sump, is driven from the crankshaft instead of the camshaft. Oil is carried by external pipes to each of the main bearings, passes through the drilled shaft to the big ends, and from thence is carried to the gudgeon pins by pipes external to the connecting rods. The cylinders are cast in single block with a three-flanged loose exhaust pipe and duplex inlet from the Zenith carburettor on the off-side. The engine is mounted rather unusually high in the frame, the 140 mm. stroke, of course, necessitating a rather deep cylinder casting. This permits the magneto to be placed on the crankcase tray on the near side, without in any way obscuring the valves, which are enclosed by a single plate. Cooling is by thermo-syphon, and a neat detail is the lubricator on the fan shaft shown in Fig. III.

So far no mention has been made of either of the Austin engines, but the 18-24 h.p. shown in illustrations Figs. IV. and V. may be taken as typical of the whole of the Austin range, as the construction is precisely the same throughout. This particular engine has a bore of 111 mm. and a stroke of 127 mm., and

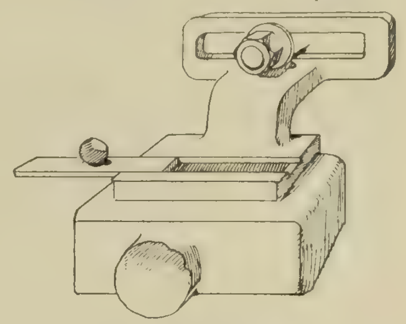


Fig. III. Lubricating device for fan on the Rochet Schneider.

is almost alone in its class, having single separate cylinders, while the manner in which these are spiggotted deeply into the crankcase should be noticed. The lubrication system is fully forced, but a centrifugal or vaned pump is used, and the pressure is therefore very low. Each of the five main bearings are fed by the pump, and the crank webs are provided with scoops or gutters which catch the overflow from the main journals carrying the oil to the hollow crank pins. The pump is driven by the customary skew gear, this being situated in the middle of the crankshaft, and there is a float-controlled needle indicator for showing the level of the oil in the sump, this fitting

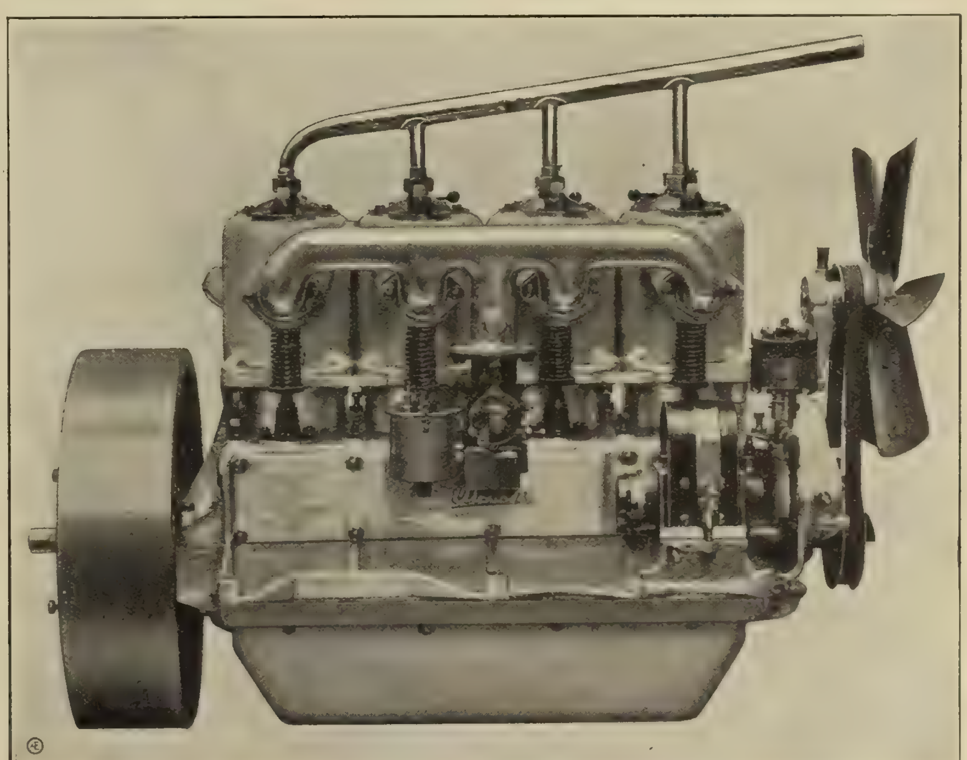


Fig. IV. The 18-24 h.p. Austin engine, showing the method of spigotting the cylinders in the crankcase.

being carried on a branch of the crankcase and arranged so that it can be inspected by a hole cut in the tray which connects the crankcase arms.

CLASS E.—ENGINES WITH A TOTAL CUBIC CAPACITY OVER 6,000 cc.

In this class E the changes in design have been so small and so few new models have been introduced that very little was to be learnt by even the most careful examination of the engines in this section. The very small average stroke/bore ratio of these large engines is very noticeable, and many of them, of course, have six cylinders. One of the entirely new designs which has not yet reached this country is the 25 h.p. Germain, which is announced to have separate external slide valves to each cylinder, with a bore of 106 mm. and a stroke of 180 mm. It is hoped that full and complete details of this engine will shortly be published in *The Automobile Engineer*. The large 35-45 h.p. Maudslay with a bore and stroke of 127 mm. is an exact reproduction of the 17 h.p. already described and illustrated. The 40 h.p. Metallurgique, bore 125 mm., stroke 140 mm., is one of the few remaining engines with overhead rocker-operated inlet valves, while in other respects it is amongst the most modernised of the very large engines. The cylinders are cast in pairs, and the lubrication system is completely forced, while the piping throughout is very large, the engine being arranged more or less for racing work.

The 45 h.p. Sheffield Simplex, again, is an example of an unaltered engine, and the description of the 25 h.p. type of the same make may be taken as applying, with the exception of the fact that the skew gear method of driving the camshaft is not included.

The six-cylinder 50 h.p. Austin, while following the lines of the 18-24 h.p. engine very closely indeed, has a rather interesting inlet pipe, shown in Fig. I. The actual intake is from the bridges which connect the outside portions of the pipe, and it may be presumed that the gas flow is more or less continuous in this outer part, the idea being that as soon as one of the inlet valves close, suction into the particular cross piece stop, and the main stream flows on to the next intake.

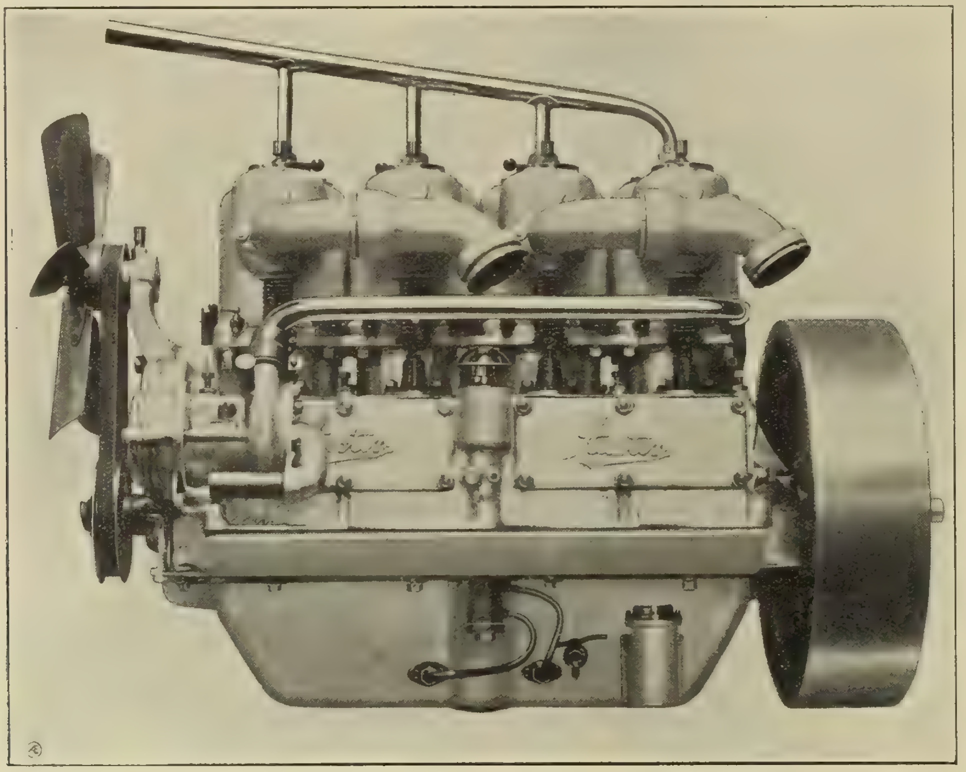


Fig. V. (Class D.) The 18-24 h.p. Austin engine, which is typical of all Austin designs.

The 40 h.p. Rolls-Royce is too well known to need any description, and besides it has not been altered for a considerable time, though it may be remarked that this is one of the extremely small number of engines in which there is a connection between the throttle and the oil supply. With a gear pump or a plunger pump, and a completely forced system such as the Rolls-Royce, it is very doubtful whether this is a necessary or even a desirable feature, because it is an additional complication, while the gear or plunger pump acts more or less in the desired way, because it must of its nature pass quantities of oil in almost direct proportion to the speed, the falling off in the rate of increase at high speed not being sufficient to affect results.

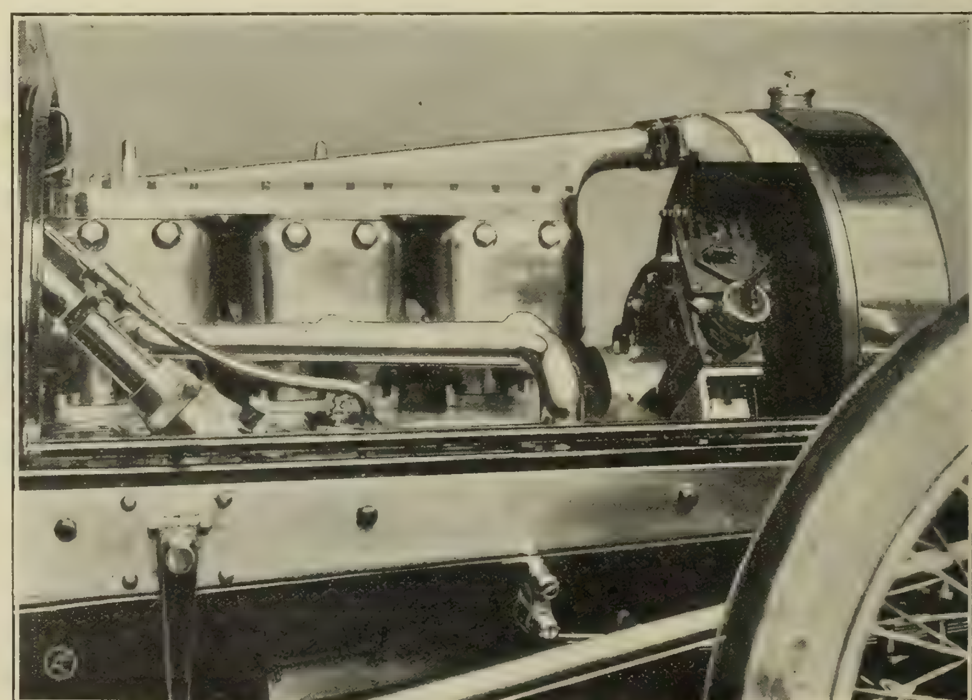


Fig. XV. (Class C.) The off-side of the 25 h.p. Sheffield Simplex.

The 50 h.p. Wolseley is an entirely new engine, and is the largest made by the Wolseley Co. for car work; the six-cylinder which was introduced last year, with the cylinders cast all together and a copper water jacket, having been discontinued. The new 50 h.p. is shown on another page, and at a glance certainly the most striking feature in connection with it is the very shallow crankcase. This complete absence of sump is due to the fact that there is a large cast aluminium external oil chamber made solid with the crankcase, and situated in the position usually occupied by the front near-side crankcase arm. This tank is capable of holding a much larger quantity of oil than the average sump, and the base chamber is exhausted by a gear pump which forces the oil into the reservoir. From the reservoir a second pump supplies the bearing feed. The valves are enclosed and, with all the piping, are arranged on the near side, the usual Wolseley system of studs and dogs being used to retain the pipes in position. The carburettor is a three-jet supplying a very rich gas to a mixing chamber in the middle of the inlet pipe in accordance with the usual Wolseley practice. Steel pistons are used, and considering their size they are not heavy, weighing only 4 lbs. 10 ozs. complete.

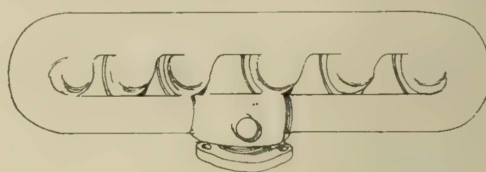
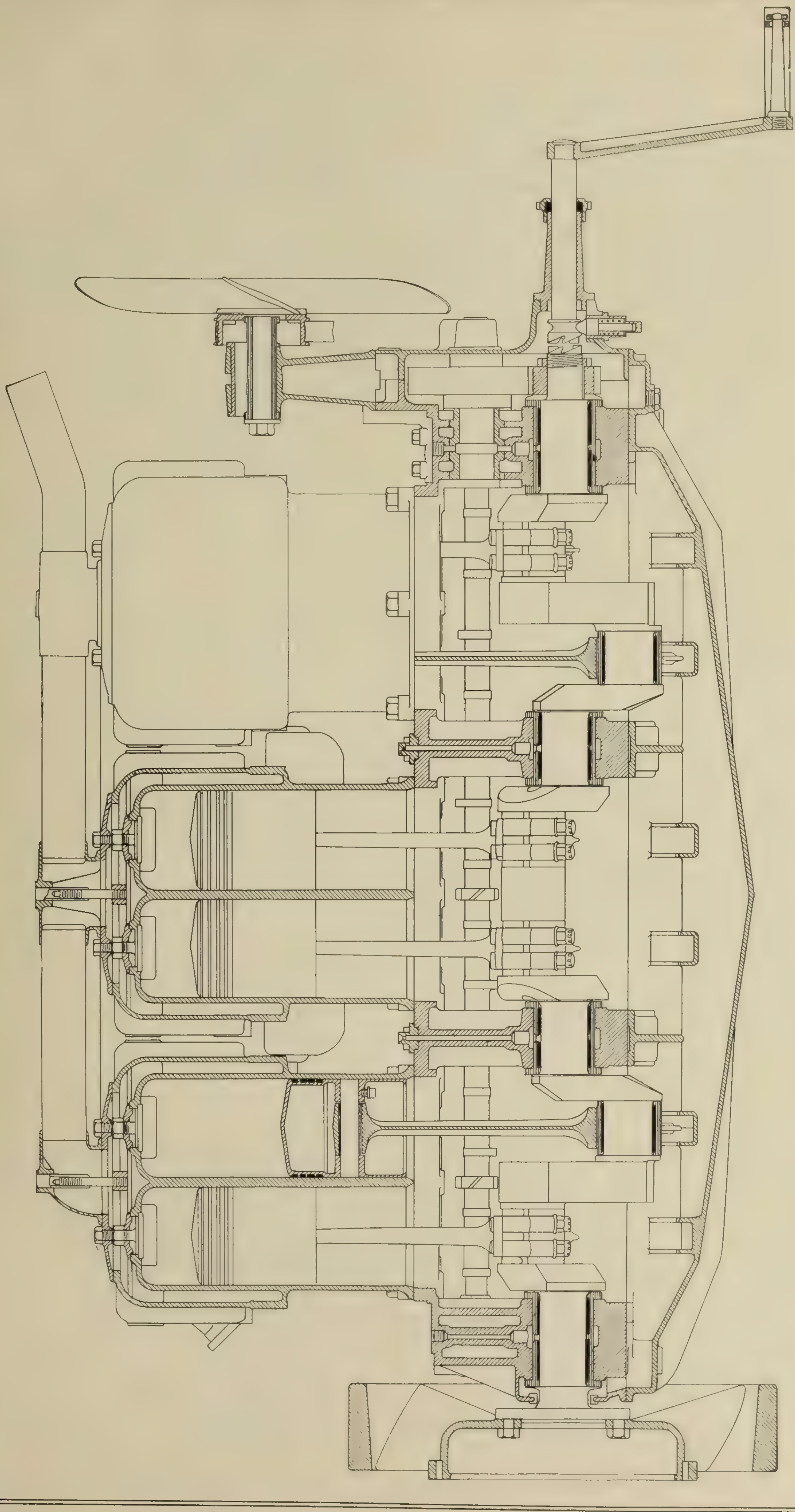
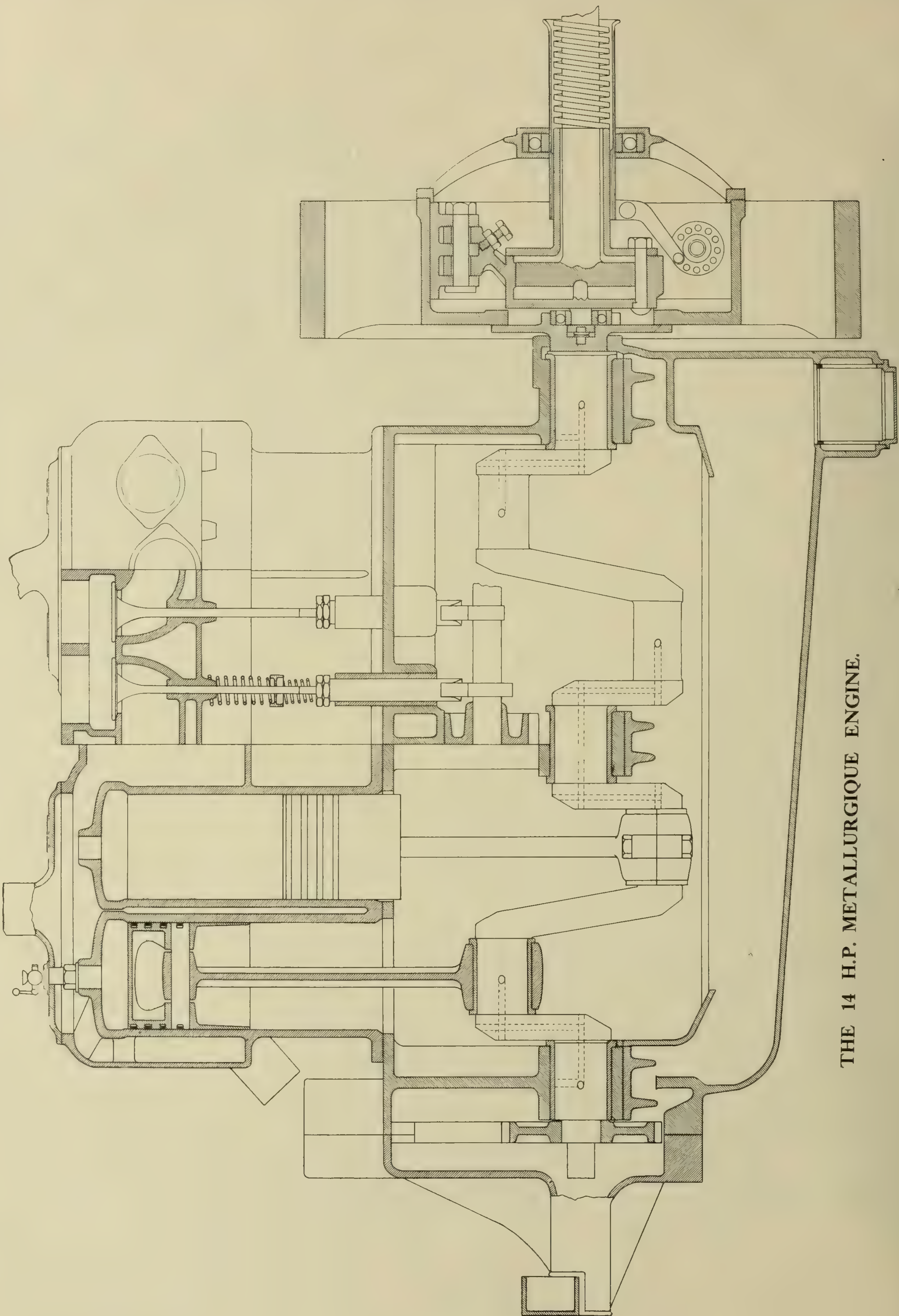


Fig. I. The Six-cylinder Austin inlet pipe.

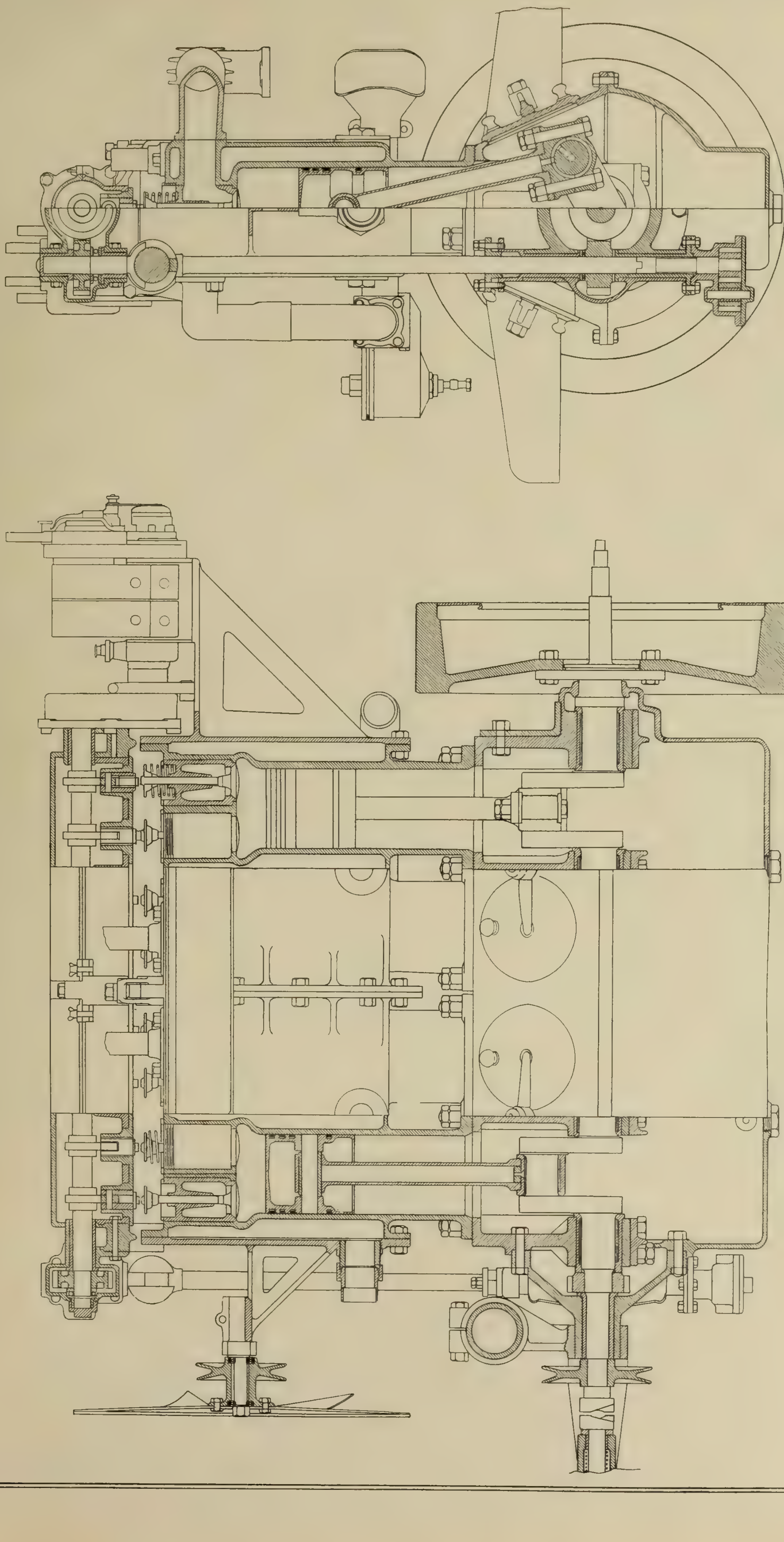
Most of the other cars in this class are both old and well-known, the last (the largest British-made car), being the six-cylinder 90 h.p. Napier, with a bore of 157 mm. and stroke of 127 mm.



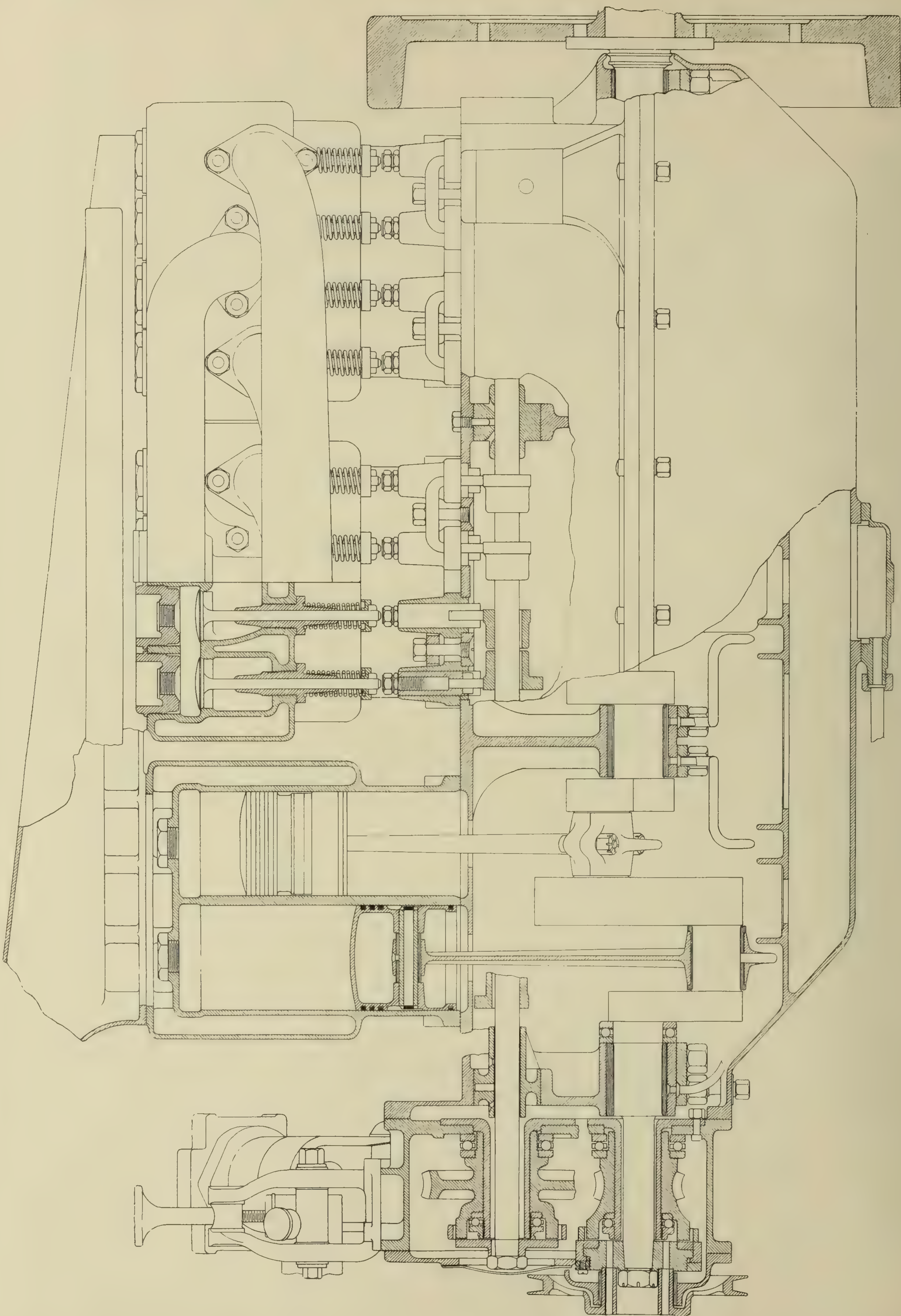
THE 50 H.P. SIX-CYLINDER WOLSELEY ENGINE.



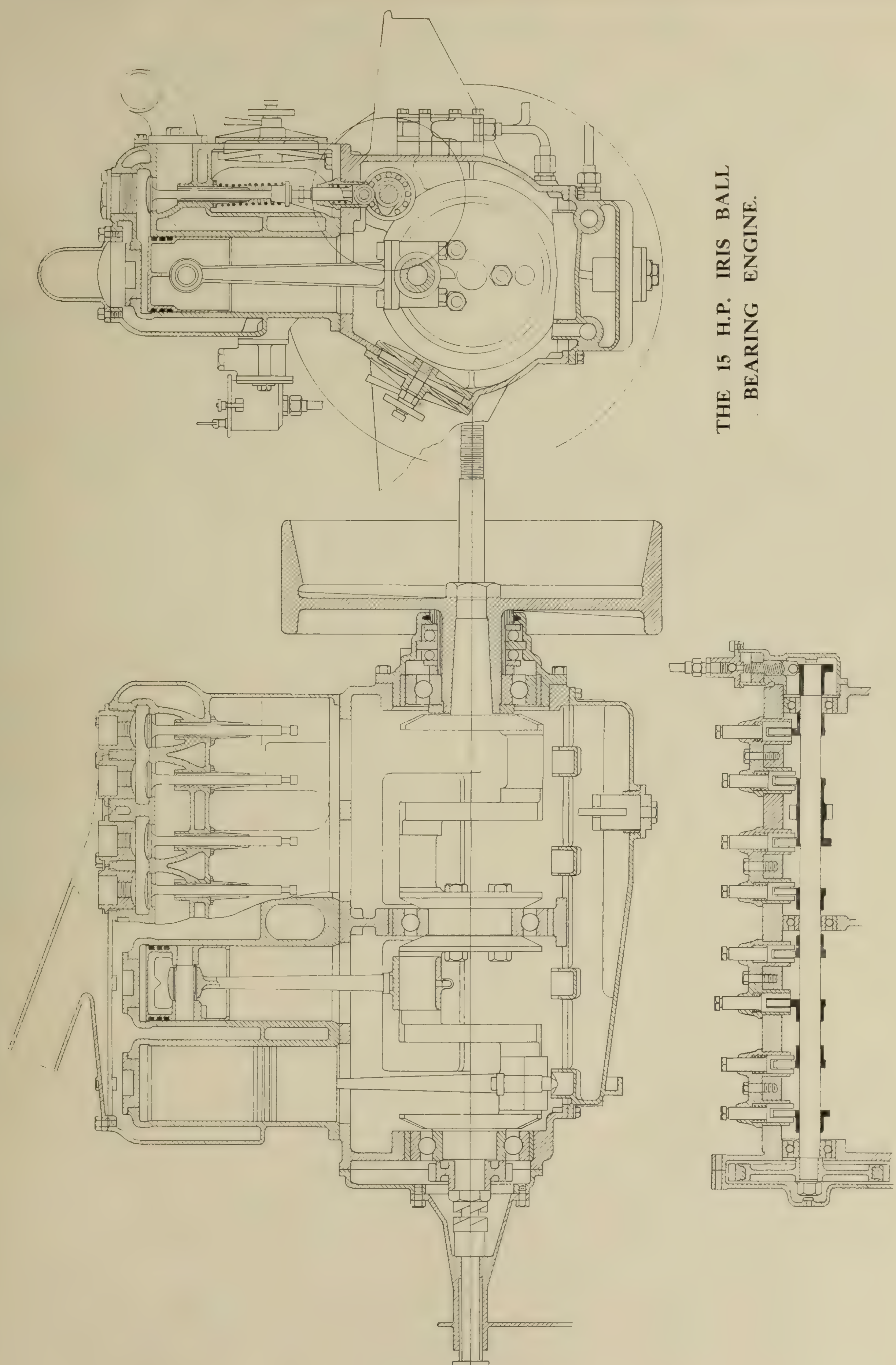
THE 14 H.P. METALLURGIQUE ENGINE.



17 H.P. MAUDSLAY ENGINE.



THE 25-H.P. SHEFFIELD-SIMPLEX ENGINE.



THE 15 H.P. IRIS BALL BEARING ENGINE.

TRANSMISSION DESIGN.

Present practice with regard to clutches, gearboxes and shaft drives.

Clutches.

JUDGING from the numbers which are fitted, the leather-faced cone clutch is most common by quite a considerable amount, taking both the internal and external cones together, while comparing the internal with the external cone, they appear to be in the proportion of two to one.

With the multiple disc variety—and in this we include all that have more than a single plate—the ratio is about 24 per cent., in the single plate about 4 per cent., and with the metal-to-metal expanding type 6 per cent. These figures show clearly that far from being ousted by the multiple disc, as was confidently foretold by certain prophets a year or two ago, the leather-faced cone has actually regained lost ground. The reason for the popularity of this clutch is no doubt in its simplicity and comparative cheapness, coupled with the fact that nowadays engines possess such great controlability in the matter of speed and power as to obviate, to a great extent—if not entirely—the necessity for "slipping" the clutch. The chief fault of the leather-faced type being that it will not stand slipping to any extent.

Although the practice of fitting clutch springs in such a way as to avoid axial load in the shaft is not quite universal, it is nearly so, and another feature which is noticeable on a great many cars is the fitting of springs under the leather for the purpose of obtaining easy, smooth engagement. In some cases these are plate springs; in others they are of the helical type fitted radially. For the connection from clutch to gear-box double universal joints are on the increase. There were twice as many of these as of the single type, but in a good many cases we noticed short shafts with "rounded" square ends, which are no doubt fairly flexible. Many makers, however, still use a rigid coupling, split for ease in dismounting, with square-ended shafts to clutch and gear-box.

As regards multiple disc clutches, flat plates, or plates with a slight amount of buckle to ensure quick release, are in the majority, and it would seem that most firms using this type nowadays make their own, and in some instances have introduced improvements which they have protected.

Gearboxes.

The almost universal practice now is to make these as one-piece castings, with a good large top cover for inspection, a design which readily lends itself to modern methods of machining, and is structurally strong. Considerable diversity of practice obtains in regard to the methods of attachment to the chassis. The number of examples of three-point suspension was comparatively small, especially so in the foreign makes, but it is worthy of note that this method has been adopted by some of the front-rank British makers. Considering, however, the very large number of first-class cars on the road, in which the gear-boxes are *not* three point suspended, but which, in spite of the fact, do run as well and are as free from trouble as anyone could wish, it would be interesting to know whether the makers who are adopting it have done so because of any proved superiority? It certainly seems surprising that so few have adopted it if it is as advantageous as some of its advocates claim.

Most of the cars shown had the gear-boxes bolted down upon either a longitudinal sub-frame or to cross members of the main frame, but in some few cases the boxes were bolted up to the underside of the cross members, a practice which makes dismantling very much easier, and which, in spite of theoretical objections, appears to give no trouble in practical working. Only one or two instances were noted in which gear-boxes were bracketed direct to the side members of the frame, the long arms needed in this type being structurally weak and objectionable.

In regard to gears, it may be said at once that sliding gears have it all their own way, among the exceptions being the Lanchester and Adams, the leading exponents of the epicyclic form of transmission. Four speed boxes are slightly in the majority, and the great advance that has occurred in fitting four speeds to quite small-powered cars was an outstanding feature of the show.

Taking the boxes more in detail, it was noted that the sliding gear shaft—usually the main one—was almost invariably castellated or splined, i.e., with the feathers milled out of the solid, and the great majority had four feathers, a few having six, and one three, but in one case (the Calthorpe) the castellations were very fine, so that the shaft looked like an elongated finely-toothed pinion.

In one case the shaft had two feathers made separately, and pinned in by screws, and in another there was only a single feather. The point made in regard to the separately-made feathers is that the shaft can be ground accurately, and so permit of the sliding gears being made a really accurate fit, which is almost impossible with milled shafts, and this is a point not to be overlooked in striving to make silently working gears.

This last remark raises another reflection, viz., that in only one instance was our attention specifically drawn to special accuracy in gear cutting, although, of course, it by no means follows that this feature was absent in many of the finely-made gear-boxes shown.

Internally-operated selector rods are the rule, but there were a dozen or more instances in which the rods had externally-operated striking gear.

The gate system of change lever was almost universal, and also the conventional form of gate, but there were some interesting exceptions in regard to the latter, which are well illustrated by our sketches.

The vogue of the flush-sided body has caused a number of makers to modify the design of their change-speed gear and brake lever brackets, the brake lever in two or three instances being fitted inside the change lever. A further improvement, which, however, was quite exceptional, was the placing of the brake lever shaft below that of the change lever. But for the most part the latter was made tubular and the brakeshaft put through it.

Reverting to the gears, there is an increasing tendency to make the top speed engagement by sliding the second or third speed wheel into an internally-toothed extension of the constant mesh wheel, or *vice versa*, a practice that doubtless gives an easier and quieter change than the otherwise prevailing dog clutch.

Lubrication.—Save in one or two instances, no special provision is made for gear-box lubrication. Ball bearings may be said to be universal, and the custom is partly to fill the box with fairly liquid grease, and then leave it to its own devices, the only point to safeguard being to prevent escape of the grease. So long as everything goes right, this plan answers well enough, but should a gear get chipped, or in any other way a piece of foreign material find access, it is held in suspension, as it were, in the grease, and probably gets carried between the teeth of some pair of gears, and causes damage. This has led some of the more up-to-date makers to use only oil for lubrication, and a certain maker has gone so far as to provide a self-contained oil pump, which keeps a constant flow of oil over the gears, filters being provided to keep it clean and free from grit, etc.

Brakes.

Only a few cars now have front wheel brakes, these being generally pedal operated. Gear-box brakes were, for the most part, of the external shoe type operated by cranked levers. The shoes were mostly separately hinged, and had renewable liners. Hand adjustments were conspicuous everywhere, and considerable care was manifest in bringing these adjustments within convenient reach.

For rear brakes practically only one description is needed, viz., the internal expanding type, the few exceptions noted being quite insignificant in numbers.

Three or four cars had both sets of brakes in the rear wheel hubs, one or two having them arranged side by side in the same drum, and one or two with two separate concentric drums. Both stranded steel cable and rods are used for operating the side brakes, the latter, in most cases, being compensated by double levers connected to a yoke piece at the divided intermediate cross-shaft. For the final drive bevels still predominate, although worms are steadily increasing in number.

Enclosed propeller and open cardan shafts seemed to divide the honours fairly evenly, though perhaps the former are in the majority.

The propeller shaft casing tube—or torque tube—is now very often fitted at its front end with a fork, which in the majority of cases is spring suspended or spring supported, although in quite a number of cars a rigid link connection was noticed.

Summarising the latest practice as indicated by the show, the specification would be somewhat as follows:—

Clutch.—External cone, leather faced, with springs under front edge of leather and a clutch brake.

Connection to Gear-box.—A double universal coupling allowing for end play.

Gearbox.—One-piece casting with ball bearings throughout, including the spigot end of mainshaft. Sliding gears on a castellated shaft providing four forward speeds and reverse, gate operated. All selector rods and levers enclosed in the box, and fitted with a positive lock, so that only one gear can be engaged at a time. Top speed on the fourth, obtained by sliding the third speed wheel into an internally-toothed extension of the constant-mesh gear. Lubrication recommended to be by oil or thin grease, with glands or other devices to prevent escape of the oil.

Brakes.—Front wheel brakes can hardly be said to have established themselves. A gear-box brake, consisting of a steel drum and a pair of external shoes lined renewably with cast iron, separately hinged at the side, top, or bottom, and preferably built up in one with the gear-box. The outer ends of the shoes operated by a crank lever and screw-ended pull rod with hand adjusting fly nut. Springs provided to keep in the off position, and adjustable stop screws or lugs to prevent rattle.

Rear brakes of the internal expanding type with cast-iron solid shoes, or shoes of cast malleable with cast-iron facing segments easily renewable. Pull rods to hand lever to be compensated.

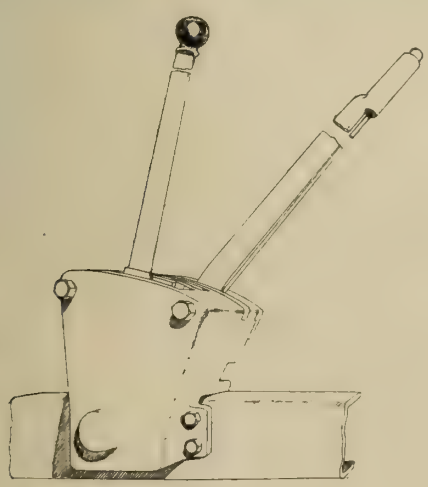
The shaft of the hand lever not to be combined with a tubular shaft of the change lever, but to be mounted on a separate and independent centre.

SELECTED EXAMPLES OF TRANSMISSION DESIGN.

The 16 h.p. Adams is one of the few cars in the show with planetary gear transmission. It is fitted with an internal type leather-faced clutch connected through a double universal joint to the gear box, one being of the double-fork type, and one of the sliding block type. The clutch withdrawal levers have two small ball bearing wheels acting on the back end of the sliding universal joint box, while a small bracket and catch to hold the clutch pedal out of engagement when desired is fitted to the dashboard. The gear box is suspended at three points, and encloses, as already stated, planetary type gears, providing three forward speeds and a reverse. As an alternative, a conventional sliding gear type of box can be fitted, giving four forward speeds and a reverse. The planetary gear box may be controlled by a hand lever and a gate, or by a pedal also working in a gate.

A foot pedal operates front wheel brakes through Bowden wire mechanism, the former being of the internal expanding type acting on pressed steel drums. The rear brakes are, as usual, of the internal expanding pattern. The cardan shaft is open, and has a universal joint at the back of the gear box enclosed in a dustproof cover, which is secured to a cross member of the frame.

The Argyll clutches are of the multiple disc type, connection to the gear box being through a split coupling. The gear box is very neat and compact, giving four speeds and reverse with direct on fourth. The mainshaft is castellated, and the



The enclosed gate of the 12 h.p. Argyll.

layshaft driving gears are at the front. The change-speed mechanism is reminiscent of racing practice, for it is entirely enclosed by a metal covering formed in one part with the gate, see Fig. 1. The gate itself has its corners bevelled off to allow the lever to slide through with unusual ease, and there is no doubt that the device is most effective.

The 12 h.p. Argyll is fitted with front wheel brakes, these being foot-pedal operated, and there is, of course, no gear box brake. The rear brakes have pressed steel drums, and are of the internal expanding type with a pull-on hand lever. The propeller-shaft is enclosed, the forward end of the tube being held in a ball and socket joint split vertically, the base portion of which is bolted to a cross member.

The Arrol-Johnston engine and gear-box are built together on the unit system. The clutch being of the metal disc type, and having three plates held in contact with each other by spring pressure and working enclosed in an oil bath. It can be removed very simply without disturbing any other part of the engine or gear box. The latter gives four speeds, with direct on fourth, all the selector levers being inside the box, and the rods fitted with notches and locking plungers. The constant mesh gears are at the forward end, and are of the single helical type, as seen in Fig. II., where the peculiar top speed clutch can also be seen. The front wheel brakes of the expanding ring type (Fig. III.), are interchangeable with those on the back wheels, which latter are operated by the usual pull-on hand lever. The propeller-shaft is enclosed, with one universal joint only at the front end. A fork attachment is used, bracketed to the end plate of the gear box.

The Austin clutch, standardised for the 18-24 h.p., 40 h.p., and 50 h.p., is of the multiple disc type, but a leather-faced clutch with segmental leather can be fitted if desired, and is made standard for the 15 h.p. model. This leather-faced clutch is built up in segments, any one of which can be easily and quickly unshipped when desired, for cleaning, oiling, or re-leathering, which latter operation is much simplified by this method of construction, as only short pieces of leather are required. The gear-box, which is four point suspended, provides four forward speeds and a reverse, with direct drive on fourth. It is connected to the clutch by double universal joints, that at the back end being of the sliding block type, and that at the front of the ring type. The mainshaft is castellated, and has the constant mesh gears at the front end, behind which there is an extra ball bearing on the mainshaft.

The gear-box brake has external shoes, made very broad and substantial. The

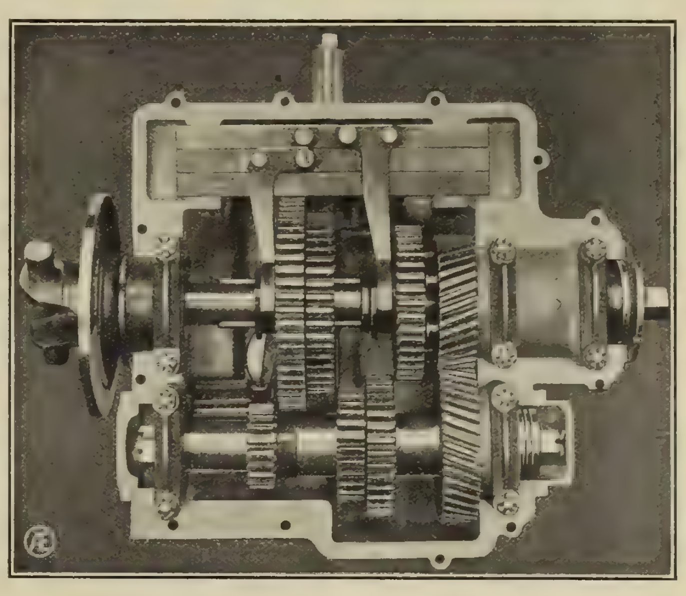


Fig. II. The 15.9 Arrol-Johnston gear box.

shoes are supported separately and are operated by an overhead crank lever. The side brakes are of the usual internal expanding type with compensated action, and they have a push-on hand lever, which has its shaft below that of the change-speed lever —one of the few instances in the show in which this practice is adopted.

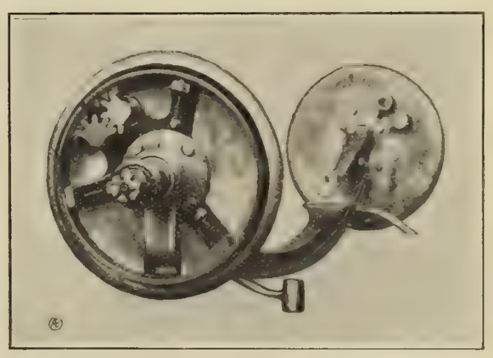


Fig. III. Arrol-Johnston brake mechanism.

An open type of cardan shaft is fitted, with a universal joint at the front end, and a sliding block coupling at the back, the final drive being by bevels. There is a tubular torque member fitted, spring suspended from a cross member of the frame.

A peculiarity on the Bentall is the clutch, which is a cone with a fibre face. It is self-contained, runs in oil, and is easily detachable without dismantling the engine or gear, the connection to the gear-box being through a short shaft with two sliding block ends held apart by a spring, while by compressing the spring the shaft and its end can be taken out instantly. The gear-box is built up in one with the propeller-shaft tube and stayed to the back axle diagonally, thus the transmission and back axle form a single unit of the three-point suspension type. There are three forward speeds, with direct on top, the mainshaft being castellated, with the constant mesh gears at the back end, while the top speed engagement is by jaw clutch. The change lever is on the ratchet principle, with two ratchet plates held together by springs, the change-speed lever sliding through one speed at a time, a slight pressure on the lever to left or right being necessary for changing up or down respectively. Both foot and hand brakes are contained in the rear axle hubs, and are of the internal expanding type, but acting on two separate drums, the outer drum being connected to the foot pedal.

The Crossley clutch is of the metal-to-metal type with two expanding segments, the gear-box and engine being built together on the unit system, and the coupling to the gearbox is rigid.

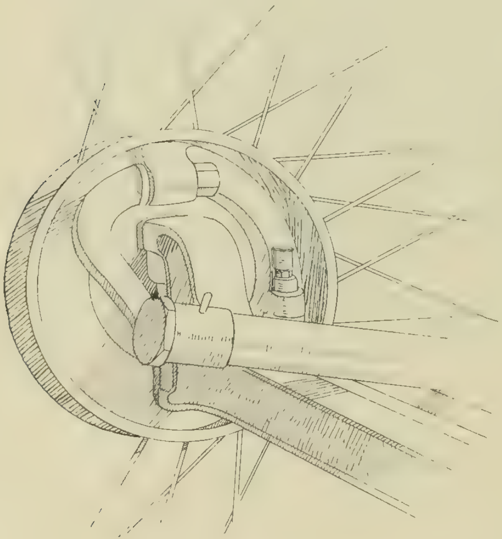


Fig. IV.

The gearbox is very neat and compact, one of the few which has its lay-shaft placed directly underneath the mainshaft, and gives four forward speeds. The top speed engages by sliding the third speed pinion into an internally-toothed wheel keyed to the constant-mesh gear. The back end of the gear-box forms the third point of suspension of the unit, and is slung from a tubular cross member of the main frame, through which is passed the shaft of the change-speed lever, a very clean bit of designing. Ball bearings are used throughout, and there is an extra bearing on the inside of the constant mesh pinion on the mainshaft.

The usual gearbox brake is dispensed with, and instead we find a pair of internal expanding brakes, pedal operated, and pushed well into the hubs of the front wire wheels. These brakes are curious, insomuch as the covers are so formed that they act in place of the more usual swivel. Both the tie rod and steering arm are fixed thereto by a method made clear in Fig. IV. The rear brakes are also of the internal expanding type, well enclosed in the back wheels.

The cardan shaft is enclosed in a taper steel tube, the forward end of which is supported by a ball and socket joint built into the back end of the gearbox.

In regard to the transmission of the Daimler cars, the only striking alteration is the substitution of a rear axle, com-

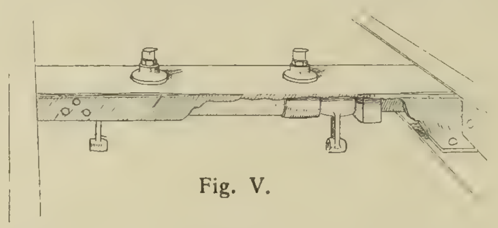


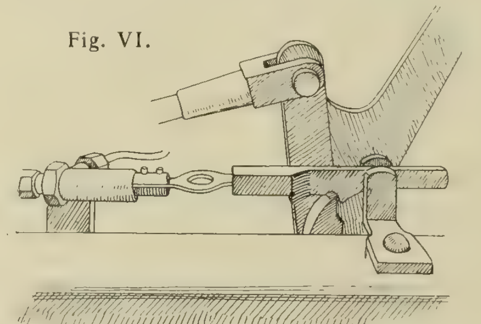
Fig. V.

posed of an aluminium worm-and-worm-wheel case, flanked with taper castings and tubes, for the old pressed steel back axle now used only on the 38 h.p. We need only therefore give briefly the leading features of the design. The clutch is of the external leather-faced cone type, with an external spring giving a constant thrust on the crankshaft through a ball washer, and has a double sliding block coupling connecting to the gear-box. Moreover, a leather-faced clutch stop is fitted, which is held up to its work by a small spiral spring. The latter provides three forward speeds with gate change. The constant mesh gears are at the back end, and the top speed engagement is obtained by sliding the third speed wheel into an internally-toothed wheel.

The propeller shaft brake and the rear brakes are both of the external band type, with metallic packing liners. The cardan shaft is open, has a universal joint of the double-fork cross-pin type at the gearbox end, and a sliding block at the back. The springs are still used to take both radius and torque action.

The clutch of the six-cylinder Delaunay-Belville is a leather faced cone, with a flexible joint connection to the gear box, the clutch operating shaft being placed below the engine shaft for convenience in dismounting. There are three forward speeds, with direct on top, engaging by a dog clutch with a gate change. The main shaft is castellated, and the selector rods are operated from the outside of the gear-box. The gearbox brake is of the external type, and has two shoes of pressed steel, with renewable liners, these shoes being supported by two hinge pins in the same horizontal line, at the right-hand

Fig. VI.



side, operated by a crank lever. The hand brake follows usual lines, but the cross-shaft is most neatly housed inside the middle cross-member, where it is protected completely, and it takes its bearings from the two hangers (both of which are shown in Fig. V.), so that grease can easily be supplied directly to the surfaces.

A torque member is fitted from the back axle on the right-hand side of the car, and a long radius rod on the left hand, both of these having ball ends at the front with springs and adjustment. The cardan shaft is of the open type, having a ring pattern of universal joint at the front, and sliding block at the back end.

The 15-20 h.p. F.I.A.T. may be taken as an example of Italian work, though this differs but little from the French. It is fitted with a multiple disc clutch, and the connection to the gear box is through a divided square sleeve, secured by six bolts. The gear box, which is gate operated, gives four forward speeds, the constant mesh wheels being at the back, while a commendable point is that there is an extra ball bearing on the main shaft inside the constant mesh gear. The main shaft is castellated, and the top speed engagement is by a four-jaw clutch.

The gearbox brake is of the external shoe type, fitted with renewable liners, and has the hinge pin below, applied by a cross shaft and helices on top, while the side brakes are of the internal expanding type, operated by wire cable. One point in the brake-gear worthy of note is the method adopted for water-cooling the brakes, and this will be readily understood by reference to Fig. VI. The front end of the gear box is supported by a flanged cross member of the frame, and the back by a dropped cross member, the propeller shaft being enclosed in a pressed steel casing, formed in one with the back axle case, while the front end is supported by a forked end on two rigid brackets.

One of the most interesting clutches now being made is the Germain internal expanding metal-to-metal design. The internal member of this clutch is a complete steel ring, expanded primarily by pushing in a cone on which rides a small lever, with roller end, and this lever is mounted eccentrically on a pin which gives movement to the end of the expanding member. In order to provide adjustment for wear, the face of this small lever is finely serrated, so that by undoing a nut it can be partly rotated, as required. The connection from the clutch to the gearbox is through a double universal joint, which, with the clutch spring, are enclosed in an oil-tight brass casing. The method of attachment of the gearbox to the frame is somewhat uncommon, being by means of two vertical end flanges bolted up to two flanged cross members of the main frame. It is wide, but very short, the gears being of large diameter, though narrow. Four forward speeds are provided, with a gate change, and the constant mesh gears are placed at the back. The main shaft has four castellations, and the driect drive is obtained by sliding one of the gears into an internal toothed wheel. The gearbox brake is of the internal expanding type, and very broad, with seven cooling ribs turned on the outside of the cast steel drums. It is operated from the foot pedal by a shaft passing through the side of the gearbox. The side brakes are also of the internal expanding type, and are operated by wire cable. An open type cardan shaft is fitted, with double cross pin universal joints, but no torque or radius rods are fitted, though otherwise the design is re-

markably clean and carefully thought out.

The Hotchkiss clutch is of the leather-faced cone type, and is totally enclosed, connection to the gear box being through a double universal joint of special design,

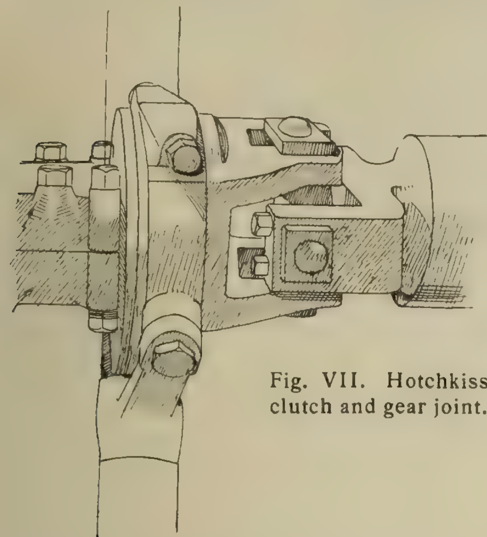


Fig. VII. Hotchkiss clutch and gear joint.

that on the gearbox shaft being a jaw pattern with block trunnion, and on the clutch shaft an open jaw with two sliding blocks, allowing for end movement (see Fig. VII.). There are four forward speeds, with a gate change. The main shaft is castellated, with constant mesh gears at the front end, and the top engagement is by jaw clutch. The reverse is one of the very few in which the pinion is normally at rest, it being brought into

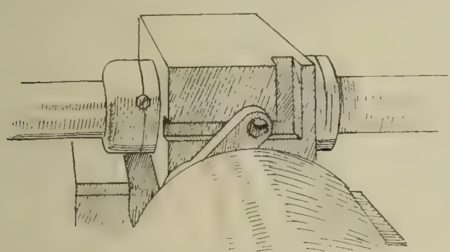


Fig. VIII. The reverse actuat'ng lever to Hotchkiss cars.

engagement eccentrically by the device shown in Fig. VIII. The first movement moves the first speed wheel back, and the pin then enters the cross slot, thereby circling the reverse into mesh. The gear box brake is of the external shoe type,

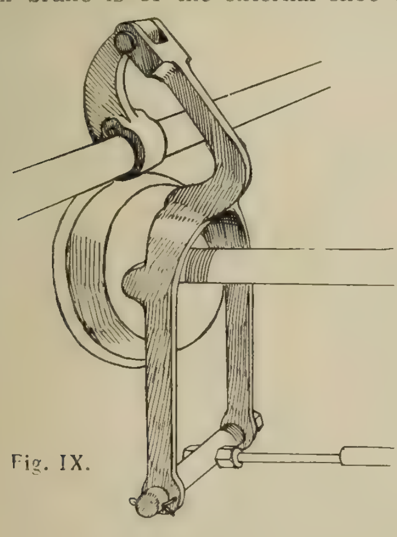


Fig. IX.

supported below and fitted with renewable liners, while the side brakes are of the internal expanding type, with cable pull compensation through a hollow cross shaft. The cardan shaft is open, with a long sliding block coupling at the back, and a trunnion type universal joint at the forward ends. No torque or radius rods are fitted.

The Rochet-Schneider has a leather cone clutch, connecting to the gear box through a short shaft, with a split coupling at the gear box end secured by six bolts. There is a very neat setting for the clutch striking lever, the latter being under-hung, as shown in Fig. IX., this making the withdrawal of the clutch a simple and easy affair. The gearbox, which is of the four point suspended type, and divided, has four speeds, the selector rods having external jaws. The gearbox brake is of the external shoe type, the shoes being fitted with renewable liners, hinged to a stout pin projecting from the gear box casting at the back, and operated at the other end by a fore and aft crank lever from the foot pedal. The propeller shaft is enclosed, and the casing tube depends for its front end support on a ball bearing on the propeller shaft. There is the usual universal coupling behind the gear box, protected by a neat brass cover, and radius rods are fitted, with hinged connection to the axle, and ball joints at the front ends.

The clutch on the Sizaire is metal-to-metal, with a single disc, through the bushed boss of which passes the rear end of the crankshaft, affording a good central support. On the cross-shaft of the clutch pedal is slung the two ends of a horse-shoe bracket supporting the forward end of the propeller shaft casing, which forms the only torque member. There is no gearbox, in the ordinary acceptance of the term, the change speed mechanism, by which are provided three forward speeds and a reverse, is combined within the rear axle casing, and comprises, instead of the usual bevel crown wheel, a lantern type wheel, the teeth of which are designed to mesh with any one of four pinions that can be presented to it by means of an angularly rocking bracket, in which they are supported. Two sets of brakes are provided, foot and hand-operated respectively, both of which are arranged side by side in drums on the rear wheels, both pairs being of the usual internal expanding type, cam-operated, and with stranded cable pulls. The Sizaire transmission and change speed arrangements are quite unique, and it must have required considerable courage to introduce such unorthodox ideas, but the continuance of their use by the original makers argues that the life is reasonable and that the efficiency is not too low.

The Sunbeam clutch is an internal leather-faced cone, connecting to the gear box through a double universal joint. The box is of the three-point suspended type, being very neatly slung to the frame by hinge bolts, two at the front and one at the rear end, as shown in Fig. X. There are four speeds, and the main shaft, which has a ball bearing spigot end, is castellated, while the constant mesh gears are at the front end, the top speed engagement being by four-jaw clutch. Both the clutch and brake pedals are provided with a neat form of height adjustment, which is explained fully by Fig. XI. The gear box brake is of the internal expanding type, with external ribs to the drum, operated by ball-jointed levers, and the side brakes are of the usual expanding type, with compensated rods. The cardan shaft is open, having a ring and trunnion encased joint at the front end, and a sliding block coupling behind (Fig. XII.). Neither torque nor radius rods are fitted. The engine gearbox and cardan shaft are set at an angle to give straight line transmission, and the hand brake lever of the side brake is arranged inside the change speed lever.

The Thornycroft clutch is of the multiple disc type connecting to the gear box through a short piece of shaft with cas-

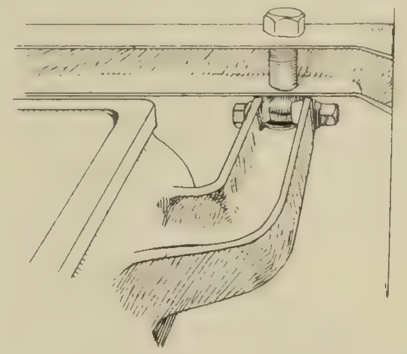


Fig. X. Showing method of attaching gear box arms to the frame.

tellated ends, but the withdrawing mechanism is operated through two horizontal bars hinged close to the left-hand side member of the frame and actuated by the pedal at the other extremity. The gear box is three-point suspended, and the change lever bracket, which also

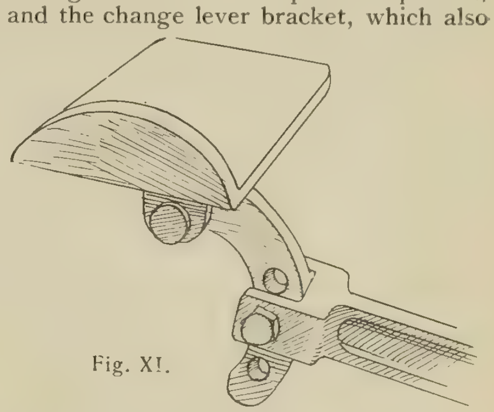


Fig. XI.

supports the side brake hand lever, is set below the level of the frame, where it is out of the way for body fitting. Both of the gear shafts are square, and the constant mesh gears are at the front end. The gearbox brake consists of a pair of external steel shoes with cast iron liners on two hinge pins at the left-hand side, the outer ends of the shoes being splayed right and left, in order to

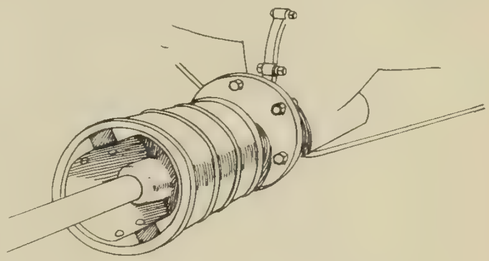


Fig. XII. The Sunbeam rear universal joint.

permit the use of vertical rods and a crank lever working fore and aft, with a ball joint pull rod to foot pedal. The cardan shaft is open, connecting to a worm drive underneath the back axle, and fitted with a sliding block type of universal joint at both ends.

The Thames clutch is of the rare, but usually excellent, metal to metal internal cone type, and connection to the gear box is by a very short shaft, with a square end coupling. There are four speeds, and a

special type of change lever swinging inwardly and outwardly at the neutral point, in order to engage with one or other of the two selector levers within the side bracket on the sector. The entirely plain quadrant used is shown in Fig. XIII.,

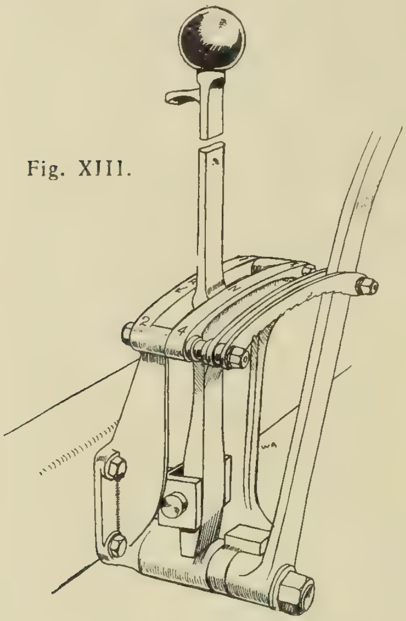


Fig. XIII.

and the mechanism really amounts to allowing the selectors in the gearbox to act as the gate, for the striking lever can be slid into engagement with one or other of the selectors by pushing the change-lever handle outwards (transversely to its normal direction of motion), or pulling it inwards. As soon as either selector is thus picked up the act of sliding the gears locks the change lever. Thus two speeds are obtained simply by holding the lever inwards while changing, and the other two by the contrary action. The main shaft is castellated with four keys, and the constant mesh gears are at the front end, the top speed engagement being obtained by a jaw clutch. This chassis is fitted with front wheel brakes, foot operated through cross shafts and rods, with compensating action, and we noticed that the dust cover of these brakes and the steering arm are cast in steel in one piece, making a very

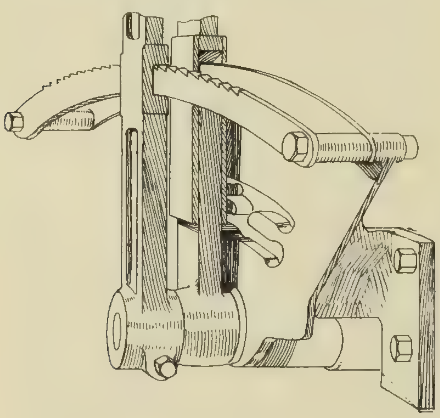


Fig. XIV. Vinot gate.

neat and serviceable arrangement. The side brakes are of the usual internal expanding type, with compensating rods. The propeller shaft is enclosed, the casing tube ending at the forward end in a substantial spherical joint.

The Vinot gearbox is cylindrical in shape and of cast iron. There are three forward speeds and reverse operated by a particularly effective-looking vertical type of gate, Fig. XIV., and two external end selector rods. The main shaft is hexagonal, the constant mesh gears at front end, and top speed engagement is obtained by a jaw clutch.

The new Valveless car contains several improvements on last year's model, although it is similar in its general lines. The clutch is of the external cone leather-faced type, with springs under the leather, the compression spring being self-contained in a long sleeve, the outer end of which is partly cut away to give access to the adjusting nuts. The clutch withdrawal levers are provided with ball bearing ends, and the connection to the gearbox is through a double universal joint, the forward one of the cross pin type and the rear one fitted with sliding blocks in an oil-retaining sleeve lubricated from the inside of the gearbox. The gearbox, which is three-point suspended, is very short, being only $7\frac{3}{4}$ ins. from centre to centre of the ball bearings, and provides four forward speeds, the reverse being idle when not in use. The main shaft is castellated with three keys, and the top speed engagement is obtained by sliding the front gear wheel into the internal toothed constant mesh wheel. The selector rod ends are external, and provided with a neat locking device. The main shaft has an oil-retaining gland at the front and a felt washer at the back.

The propeller shaft brake is of the internal expanding type, with the shoes hinged to the projecting end of the reverse gear shaft, and is pedal operated. An open cardan shaft is fitted, with cross pin universal joint at front, the pins being of different sizes, the smaller one passing through the larger and both being lubricated from the inside of the gearbox. There is a sliding block coupling at the back end. No torque or radius rods are fitted, the rear springs being made specially strong on this account, and with very little camber.

The Deasy models all follow the same lines exactly, the clutch being a single-plate type, but a considerable improvement has been effected in this year's model by fitting the clutchshaft with a ball and groove type of universal joint in the boss of the plate, a similar type of universal joint being fitted at the back end of the clutchshaft in front of the gearbox. The latter, which is of the one-piece type, with a large inspection cover on top, provides four forward speeds and a reverse, while it has a conventional type of gate change. The layshaft is so arranged as to run at a low speed, and the reverse pinion is idle except when it or the first speed are in use. The constant mesh gears are at the front end, and the top speed engagement is obtained by sliding an internally-toothed wheel over part of the constant mesh pinion.

The shaft brake is of the external shoe type suspended from two studs on top, and operated by a crank lever underneath, the hand adjustment being brought well to the side by means of a tubular shank. The side brakes are of the usual internal expanding type, but, contrary to the almost universal practice, the hand lever pushes forward, and it is brought inside the change-speed lever. The cardan shaft is open, has a ring type universal joint at the back and a sliding block type in front. The Lanchester type of rear spring suspension is used, including the system of parallel torque rods.

As regards the smaller Wolseley models these are still on the lines of the 16-20 h.p. recently described in our columns, but the 50 h.p. chassis embraces a number of improvements. The clutch is a multiple disc connecting to the gear box through a single universal joint of the ring and trunnion type. The gear box is three-point suspended to a tubular cross member in front, and a flanged member at the back, it giving four forward speeds with a plain pattern gate change. The main shaft is castellated, and has six keys, the layshaft drive gears being at the front end. There is an extra ball bearing inside the constant mesh pinion, and the top speed engagement is obtained by sliding the fourth speed wheel into the centre.

Continuous lubrication of this box is provided by means of an oil pump attached to the outer end of the layshaft, con-

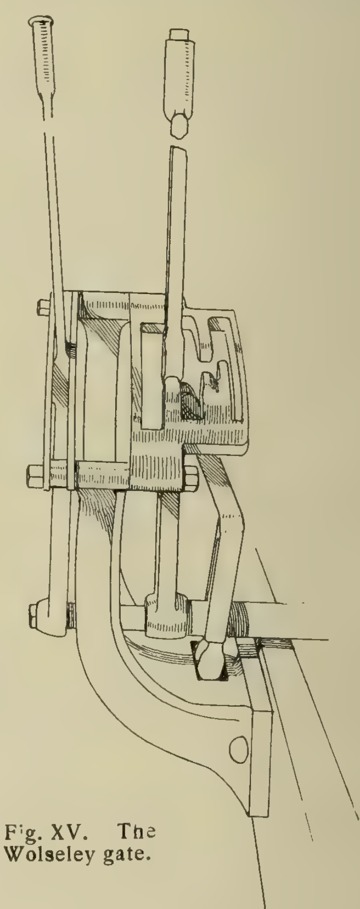


Fig. XV. The Wolseley gate.

veniently accessible filters being provided.

An ingenious pattern of gate is used on this and all other Wolseley chassis, and it is fully explained by the sketch, Fig. XV., the ball-ended fulcrum lever giving a very free changing action and removing all danger of binding.

The shaft brake has external shoes with cast iron liners supported on a single hinge pin at the top, having a crank lever with hand adjustment underneath, adjustable stop screws being fitted to prevent rattle. The side brakes are of the usual internal expanding type with rods and compensating motion, but with horseshoe-shaped flat springs to effect the contraction instead of the usual coil type. The cardan shaft is open, having a universal joint at the front end and a sliding block coupling at the back, and there is a tubular A-shaped torque member with spring and ball end supported from a cross member of the frame.

FRAME, SPRING, AXLE & STEERING GEAR DESIGN.

Frames.

THIS portion of the chassis has been modified but very little. The general design with inswept front and upswept rear appears to have settled down, and the straight frames which the coachbuilders liked so much a year or two ago are now only met with in a few rare instances. The upswept rear portion fits in very well with three-quarter elliptic springs where the front portion of the lower springs acts as a radius rod, for it enables the front spring bracket to be carried out from the frame in a direct line and does not incommode the coachbuilder. The slight tendency which at one time prevailed to set the frame down very low at the doors for the rear seats and carry it upwards in two steps is not so prevalent, and simplifies the scantlings for the body, whilst the increased height of the steps is little more than negligable.

In cars of the 15 h.p. class the side members have been considerably strengthened, often having a depth of five inches between the flanges. The stiffness thus obtained cannot fail to be beneficial to the bodies, for the springing of the frame probably strains the woodwork, especially the door pillars, more than is generally appreciated.

The ash frame reinforced with flitchplates has now practically disappeared. The only notable examples of the use of a wood frame are the Sizaire and the Panhard. In the latter case the steel frame is normal in front of the dashboard but just behind it becomes much deeper, whilst the flanges narrow until at the upward set over the rear axle they widen out and the web again narrows down. Between the dash and the rear axle the frame is stiffened with an ash beam placed on the outside and bolted up, on which the body is generally mounted. In connection with this subject it may be remarked that the latest motor omnibuses are being built with ash frames reinforced by means of flush flitch-plates. In order to accommodate the flat front springs which have come more into prominence the front dumb irons have been set down further, giving a more blunt appearance to a number of cars, as the frames have been kept flat up to the radiators and all the set obtained between the radiators and the front dumb irons.

The elaborate pressed frame employed on the Darracq is not used on other cars, but this firm still employs it. The shape gives considerable lateral stiffness as well as great strength in a vertical direction.

A noticeable feature of the French cars generally is the absence of the large gusset plates that were used at one time; so much is this in evidence that the frames appear to lack longitudinal stiffness, and some form of oblique bracing would seem desirable. The Arrol-Johnston, on the other hand, has great stiffness in this direction. The side members are swept in to meet at a point at the extreme rear of the car. This form of the frame allows free scope to the design of the boat-shaped bodies which this firm is supplying as their standard pattern for open touring cars.

The tubular cross member which has always been used to a greater or less extent seems to be coming more into favour. There are considerable claims for its adoption. It is undoubtedly stiffer than the pressed channel section in every direction other than vertical, and it has much greater torsional rigidity. Probably it is not used more on account of the cost of welding on the necessary lugs and brackets, but it would greatly improve several makes of car, especially those which at present have the gear boxes mounted on rectangular pressed steel U section members, which pass beneath them and are secured to the side frames. As a rule, these cross frames have to be supported to prevent the gear boxes moving when the pedal brakes are applied.

A noticeable case of the use of tubular cross members is the Crossley car. In this case the power unit is slung from the centre of a tube at the front of the engine and from two points at

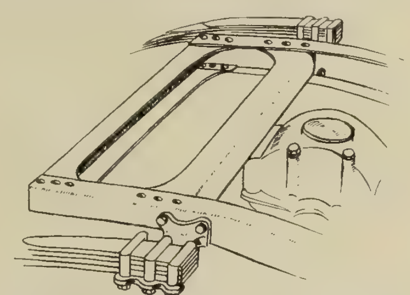


Fig. I.—The rear of the Pilain Frame showing extra cross member.

the rear of the gear box on another tube. The rear cross member is also a tube. A novel point is the method of carrying the change gear actuation through the centre of the tube from which the gear box is hung. This enables the change gear bracket to be rigidly bolted to the frame, and there is no "lining up" to be done. The propeller shaft in this case acts as a radius rod so that the drive is transmitted to the rear suspension tube, which is braced with two oblique tubes to the side frames. The torque from the propeller shaft brake is taken on this tube and it thus has to perform many duties. The reaction from the flywheel when the engine is accelerating is transmitted to the side frame through small tie rods which are placed between the lower portion of the crankcase and the frame, one on each side. A tubular propeller shaft is employed on the Sheffield-Simplex, which is swaged at each end to give considerably greater thickness to the tube at these points and enables a taper to be turned on the end without weakening the tube, and at a very small cost. It would appear that a tube which has been so treated could be flanged at each end, leaving a reasonable thickness of flange without much expense. This would have the effect of obtaining a tube with a circular flange at each end without it being necessary to braze brackets on—the chief objection to the tubular cross member.

The cross frames at the rear of the chassis are of very varied design, especially those which are fitted with three-quarter elliptic springs. Quite a large number have no support near the bracket carrying the upper spring and allow the main frame to carry the weight of the rear portion of the car in torsion even when the springs are considerably overhung. Others have a generous cross member at this point, one of which, the Pilain, is illustrated in Fig. I. The channel section frame is rivetted through the main frame by the same rivets which carry the spring bracket and makes a strong rear support, whilst the side member is merely called upon to withstand the stresses in a vertical direction.

In practically every case the cross frames are pressed with the gussets in one piece.

Underframes.

Amongst the smaller cars there are not a large number with underframes, and in the larger types the number of underframes shows signs of diminishing. It appears to be becoming recognised more and more that a really good universal joint between engine and gear box is indispensable, and when this is accepted the underframe ceases to be of much value. On a number of cars an underframe is formed in one piece with the main frame, the lower flange of which is carried towards the centre of the chassis and is rivetted to or itself pressed into a channel section on which the engine and gear is carried. This method, whilst saving something in the erecting of the frame, also makes it very stiff. It appears to be open to the disadvantage that any spring or whip there may be in the main frame is communicated to the underframe and destroys the alignment.

Front Suspension.

The springs at the front of chassis have been modified but very little. They are practically all carried beneath the frame and are fixed to the dumb irons at the front channel end by means of a bolt, and at the rear end have a shackle, the top bolt of which passes through the frame. The slide which several makers fitted to the rear end of the front spring a few years ago has disappeared. There is a distinct tendency to employ springs which under normal load are quite flat, and this remark applies to rear springs as well as front ones. They are also made much longer than formerly, front springs 40 ins. long being common. This feature, combined with a large number of relatively thin plates, enables the springs to be made very soft, a flexibility of 11 ins. and 12 ins. per ton being common.

Rear Suspension.

The most popular type of rear springing is the three-quarter elliptic. The advantage of the flexibility possessed by this type, coupled with the lateral rigidity and absence of "swaying" has induced a number of makers to adopt it, and discontinue the use of the transverse spring, which is now only very seldom seen.

A distinct feature is the number of

firms which are departing from the use of the springs to act as radius rods. Especially in the case of the larger cars is this the case. One reason given for this is the unsuitability of the central spring bolt to act in shear, and there have been numerous cases where the heads of these bolts, through which the drive is

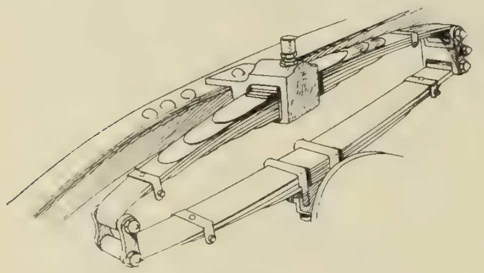


Fig. II. The Austin suspension.

obtained, have sheared off. Possibly this is due to some extent to the spring makers employing mild steel bolts for this purpose, although alloy steel has been used in a great many cases. The stress on these bolts is undoubtedly very severe.

A modification of the three-quarter elliptic is the suspension employed by the Daimler and Standard. In these cases the half elliptic is mounted on two spiral springs at its rear end, which in turn are carried by the rear dumb iron. A further modification is that of the lever spring suspension, which consists of a flat coiled spring carrying the load on the rear end of the half elliptic through a pivotted lever. It is claimed for these systems that whilst the carriage springs absorb the large vibrations, the auxiliary springs respond to the small ones, which the less flexible plate springs would not be affected by. It may be mentioned that the lever spring suspension is now being fitted to heavy cars with double springs, by means of which the load on each spring is reduced, and the flexibility of the springs increased.

Full elliptic springs are fitted to a few cars. On the Austin the front end of

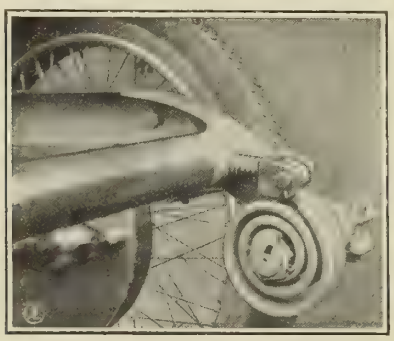


Fig. III.—The lever spring suspension.

the lower spring is anchored to the frame, and the drive taken through it, whilst the front end of the upper spring is free to swing on shackles. This system enables the full benefit of the flexible full elliptic spring to be employed without sacrificing the lateral rigidity of the frame on the axle.

Rigidity is obtained on the Arrol-Johnston (also fitted with full elliptic springs) by means of a simple parallel motion, which anchors the centre of the axle casing to the frame, whilst allowing freedom of movement to the springs.

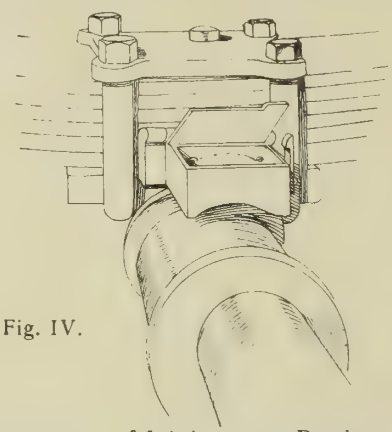
Cantilever springs are still employed on the Lanchester and New Engine cars, and the Deasy has also adopted this system of suspension.

By the use of double cantilever springs the torque on the Marlborough car is taken on the springs, in addition to the driving and braking effort. This system of rear springing was used with considerable success on the Napier cars, which were built two years ago for the Grand Prix.

The lubrication of spring bolts is a matter which does not appear to have received much attention by a large number of firms. In some cases small oil holes have been drilled in the springs, and no further provision made for greasing the bolts. The cheeseparing policy adopted by these firms in this respect is greatly to be deplored.

Amongst the firms which have adopted an efficient means of lubrication for the spring bolts, the balance of opinion for oil and grease seems to be fairly level. In connection with spring-bolt oilers, a small "Enots" lubricator, which can be used either vertically or horizontally, has

Fig. IV.



Arrangement of lubricator on Brasier axle.

been introduced, and whilst holding a fair amount of oil, can be readily filled, and is very successful.

One of the most difficult parts of a car to lubricate properly is that type of rear spring seating which is not fixed to the axle tube. The small amount of rotation prevents the efficient use of grease, as the grease ways are apt to become clogged, and it is not easy to put an oiler immediately above the bearing on account of the spring being in the way. A unique solution of the difficulty has been devised by the Crossley, which consists of a wick syphon oiler screwed into the spring centre bolt, which is hollow. The oil passes through the centre of the bolts, down through a hole in the spring seating on to the axle tube just in the centre of the bearing. Another good arrangement is the oil cup or box on the Brasier, shown in Fig. IV.

Rear Axles.

The popularity of the worm drive is steadily increasing, and the number of cars, particularly British-built cars with worm drives, is increasing by leaps and bounds. To a great extent this is due to the demand for silent running, for the worm drive is not only more silent than the bevel after it has been on the road for some time, but it is easier, quicker, and cheaper to get quiet in the first place, which is a point for the manufacturer to consider. With regard to the actual cost of manufacturing the two types, there is not a great deal either way, although the expense of the worm drive axle mounts up on account of the large ball thrusts which are fitted. With regard to efficiency, the bevel has the advantage, for the surface friction between the worm and the worm wheel accounts for a considerable amount of power, whereas in the case of the bevel the contact is of a rolling nature, and the friction much less. Experiments made with two similar cars, one a worm drive, the other bevel driven, which have been run for some distance at the same time, has shown that the rise of temperature in the bevel gear casing is very much less than in the case of the worm drive. It has also been shown that the petrol consumption suffered somewhat in the case of the worm-driven car.

The Dennis still has the worm above the worm wheel, and gets the advantage of a considerable amount of clearance from the road, and this advantage will be still more apparent should there be a demand for colonial worm-driven cars. The thrust races fitted in most of the worm-driven cars to take the worm thrust are of ample dimensions, $\frac{3}{4}$in. balls being used in several cases. It is stated that unless large thrust races are employed the races are ruined, owing to the heavy starting thrust partly bedding the balls into the casing of the thrust race, and ruining the surface. The amount the balls are bedded in is only microscopic, but the effect is to crack off pieces of the hardened rim of the races, which quickly damages the thrusts.

Bevel drive axles have been modified very considerably. The chief feature in the design of axles throughout has been in the nature of making them more accessible. In a great many cases it is possible now to dismantle an axle completely without taking this casing apart, except to remove the bevel wheel, and even this may sometimes be removed through the inspection cover.

The pitch of the bevel teeth is generally between 4 and 5, but in some of the small cars the teeth are as small as 7 pitch.

The worms and worm wheels are practically all of hardened steel and phosphor bronze respectively, while the bevels are of hardened steel without exception.

Differentials.

Both the bevel and the spur type of differential pinions have their supporters, and although the spur type has gained a few adherents, the bevel type is far from obsolete. From the point of view of manufacturing, the spur wheel is cheaper, especially as some of the bevel wheels are cut on bevel planing machines, which adds considerably to their cost. They lend themselves more readily to design, however, and are more easily packed into a relatively small cage.

There is a marked tendency to adopt aluminium for the centre casing portion in place of malleable iron, the design being as nearly spherical as possible, in order to get it as strong as possible. The desire to make the casings accessible from the rear has resulted, however, in the fitting of a cover of large diameter, which has the effect of impairing the strength of an otherwise very strong centre casting.

In connection with accessibility, the Armstrong-Whitworth axle allows for the bevels to be adjusted when the axle is being tested, which must result in a considerable saving in time in getting the bevels quiet.

Cast steel casings are employed on the Metallurgique, and these large castings, which are tube and central casing in one piece with a joint at the centre of the axle, are giving great satisfaction.

A design which has rapidly come to the front is a pressed steel casing made in two halves longitudinally, and welded together. The shape is not unlike a banjo with two finger boards at opposite sides of the centre. The differential gear is inserted through a large hole at the rear, and the bevel pinion casing is mounted in the corresponding hole at the front.

Several firms have adopted this casing chiefly on account of its lightness and accessibility. The Thorneycroft worm drive is mounted on this type of casing, which in this case is turned into a horizontal position with the inspection cover at the top. This design is open to the criticism that by turning the casing horizontal the strength is in the wrong plane to resist the weight of the car.

The centre casing of the 12-15 h.p. Panhard is a die casting in aluminium alloy. The two halves are not machined in any way before they are bolted together. These are very large die castings to be manufactured in commercial alloy.

The spherical aluminium centre casing has brought into popularity the conical steel axle tubes which are flanged to a large diameter where they are bolted on to the casing, and taper down to a

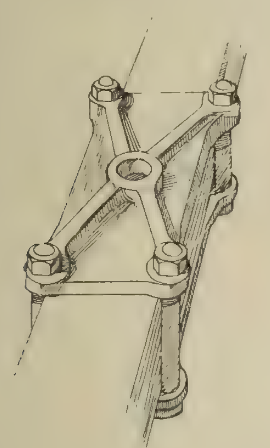


Fig V. –Straker-Squire spring pad.

small diameter, where the road wheels are mounted. These tubes are stampings forged from the solid, and give a very strong but light appearance to the aluminium casings. Where this type of axle casing is employed the torque rod is arranged to encircle the tube, and is bolted to the casing by the same bolts which are used to secure the tube and casing. The torque rod is made flexible, so that it will respond readily by bending in a horizontal plane, when the axle is thrown out of its position at right angles to the length of the car when one spring becomes depressed more than the other. The Mors axle is typical of this arrangement, the pressed steel torque rod being bolted to the axle casing by the same bolts which secure the tube to the differential casing.

A neat arrangement for securing a plain axle tube to the centre casing is that of Straker-Squire, in which the sockets in the centre case are split, and are clamped to the tubes by four bolts in each. The spring pad on this axle is also neat, the usual U clips being dispensed with in favour of bolts, as shown in Fig. V. In the case of the Panhard axle, which has been previously mentioned, the axle tube is riveted to the casing with rivets inserted from the centre of the tube, and a collar is shrunk on the outside of the casing.

Front Wheel Brakes.

Signs are not wanting to show that brakes operating on the front wheels are becoming more popular. Not only did more makers exhibit this form of brake at the Show, but others are experimenting with it. Possibly the chief reasons for fitting them are the novelty of the device —which is purely a show-room reason— and the intention of manufacturers not to be behind the times in case it comes into great popularity. There is another reason which may have induced some firms to adopt it. The type of torque tube which is secured to the gear box or frame at the universal joint and does not include a forked end, makes it a somewhat difficult matter to arrange the brake drum and brake gear satisfactorily, and the elimination of the brake at this point enables a cleaner design to be introduced. Again, this type of brake has been more or less forced upon those makers who suspend the gear box partly or wholly upon the rear axle. Some of them have both brakes in the rear hubs, but this increase of unsprung weight has tempted others to fit front wheel brakes, instead of piling on weight on the already heavy rear axle.

With the exception of the Thames, which has torque rods fitted to the front axle, all makers are content to take the torque of the front brakes on the springs, and this design has the merit of simplicity, although it is not more correct in theory than mounting the rear brakes in a similar manner. The front axles are considerably stiffened between the brake drums and the spring seatings, as indeed they should be, for the torque is occasionally very severe, and the shock may also be by no means inconsiderable if the brakes are applied suddenly, as they often must be.

Regarding the methods of actuating the front brakes, these are unsatisfactory as a whole, for in every case they are controlled by rods swinging from the frame at some point at the rear of the axle. This can only result in the brake being applied unevenly as the car rebounds on the springs. The effect can best be studied by applying the brake with the car stationary, and causing the springs to rebound by jumping on the front dumb irons. It will be found that the pedal moves quite a considerable amount in some cases. The front ends of the front springs are supported by bolts and the rear ends by shackles, so that the spring seatings oscillate in what is approximately the arc of a circle. Owing to the variation in the camber as the springs rebound the centre of this arc is not coincident with the front spring bolt, but is at a point somewhat higher and in general, somewhere on the front dumb irons. This would appear to be the correct point for suspending the brake actuating links, but it would complicate the control from the pedal and appear unsightly except in those cases where a tube is carried across the dumb irons to which a support for the starting handle is attached. This could be used to carry the actuating levers, but the compensation device would require to be placed on the pedal shaft, where it is now found on most models.

Probably the most correctly actuated front brake is the Thames, which, as previously stated, has torque rods to the front axle. The front springs are shackled at the rear, whilst the front slides in the dumb irons, which are well arranged for lubrication. These slides are necessary on account of the rear ends of the torque rods being fixed and have a less unsightly appearance than shackles would give.

Front brakes are all internal expanding and are actuated by a cam or toggle, the cam often being controlled by shafts placed directly below the axle pins and carried on bearings on the front axle near the spring seats.

If it may be assumed that skidding takes place only when the wheels cease to rotate at a speed corresponding to the speed of the car, then there is less danger of a skid with front wheel brakes, for the application of the brakes puts more weight on the front wheels and reduces the weight at the rear. This argument is in favour of front brakes as they may be made more powerful, but front skids are so much more dangerous than rear ones that probably they will not be used to their fullest extent.

Rear Brakes.

The internal expanding rear brake is now universal, and little modification has appeared in its construction. The actuation also has been changed very little with the exception that the long compensating bars which were placed across the frames on some French cars have now fallen into disuse.

The torque of the rear brakes is still taken on the springs in the majority of cases, and although the springs are not particularly suited for the strains introduced, yet this method adds security to the car by allowing it to be pulled up by the side brakes if anything happens to the torque rod or differential when the pedal brake becomes useless.

Metal to metal brakes, cast iron and steel, or malleable cast iron, are the most popular materials for the rubbing surfaces, but composition linings such as Froudes are being used to a considerable extent. This class of material is particularly suitable for brakes which become lubricated from leaky bearings, etc. There is not the same tendency for the brakes to slip at first and then grip suddenly as there is with the metal to metal brakes as the oil is burnt off.

A novel brake which was fitted to a Metallurgique shown by Hill and Boll is the Weight hydraulic brake. The name is a misnomer, as it is really only hydraulically operated. The pedal actuates a small plunger pump and the pressure, operating through the medium of oil contained in flexible pipes, actuates the brakes, which are placed on all four wheels. The application of the brake is particularly sweet, and it operates as rapidly as though the pedal were positively connected. The action of the fluid gives perfect compensation and there is no sign of oil leaking at the plungers in the brake drums. The hand brake in this case still remains on the rear wheels for use in emergency.

Wheels.

Little can be said in connection with this subject with regard to wood wheels, except that they are still used to a very large extent. There is not the tendency to dish them as much as formerly, and for rear wheels they are seldom dished at all.

The Rudge-Whitworth wire wheels have triple spokes and a double locking device in the case of the detachable wheels.

The pressed steel wheel made by Sankey's is coming into considerable favour, and was exhibited on a large number of cars. Its appearance is exactly that of a wood wheel, and it has the advantage of being very quickly and thoroughly washed. The success of the wheel depends on the class of work which is put into the welding, and so far as can be judged this is excellent. It is necessary also to keep the water from getting to the inside of the hollow spokes and rusting them, and this point appears to have been appreciated.

Front Axles.

The most novel front axle shown is the Lancia, which is of channel section pressed steel. It is of very generous proportions, and despite its section appears to be sufficiently strong, while it is un-

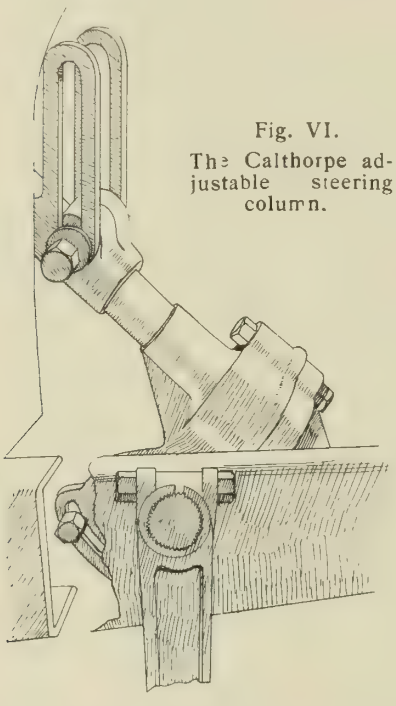


Fig. VI. The Calthorpe adjustable steering column.

doubtedly a cheap job to manufacture. The end pieces are riveted into the pressing, the spring seatings being bolted to the axle, the bolts passing through the flanges the spaces between being taken up by small tubes acting as distance pieces. It will be distinctly interesting to see if this axle is copied by other makers. The section does not seem to be suitable for use in conjunction with front wheel brakes.

The tubular front axle has no new users, and the I section axle maintains its popularity.

Steering.

In connection with the steering box the most noticeable feature is the adoption of a complete worm wheel in place of the sector. The tendency appears to be in this direction without giving any adjustment to the sector shaft with relation to the worm shaft. Some adjustment would appear to be desirable in order to provide for wear of the worm, which generally is being made of harder material. A case-hardened worm used in conjunction with a soft worm wheel would probably last as long as the car. Several makers are using steering columns with adjustable rake, and a neat fitting of this kind is shown in Fig. VI.

Taper roller bearings are being used in steering boxes for both the worm and the sector shafts, and as they are suitable to take both end thrust and journal pressure, should be very successful.

The steering connections generally are somewhat heavier than previously, and more wearing surface is being allowed on the pin joints. The axle pins are being set out of the vertical in a good many cases, but opinions differ as to whether they should be inclined forwards or backwards. Some firms appear to believe that either way gives the steering a sense of direction. In any case the inclination should only be slight, as the wheels become unsteady when the pins are inclined more than a few degrees.

Torque Rods.

Probably no feature is more noticeable than the growth of the number of firms which use a casing or tube surrounding the propeller shaft for the purpose of taking up the torque. This method, which has been standard practice on F.I.A.T. cars for several years, is now being very generally adopted. This F.I.A.T. arrangement has been modified in several ways. One of the most interesting is the Panhard device. On these cars—the same arrangement is fitted to several models—the torque tube is relatively short and the forked end, which is anchored in line with the universal joint, is not rigidly bolted to the cross frame but is supported by bolts with spiral springs which give a cushion action. The propeller shaft runs in bearings in the torque tube, and owing to the centre of the support for the fork varying as the springs are compressed by the driving effort, a short universal shaft is necessary between the front end of the propeller shaft and the rear of the gear box. The two joints are of the block and trunnion type, and take up but little room.

An advantage claimed for this device is that the fork need not be capable of rotating on the torque tube. This is necessary where the points to which the fork are attached are fixed in order to compensate for the swaying of the frame on the axle when both springs are not compressed to the same camber. In the Panhard type the springs on the fork supports allow the fork to follow the movement of the axle in this respect.

Several firms have taken advantage of this type of torque rod to make it do the duties of the radius rods. The thrust is taken, generally speaking, on the universal joints, although some firms have fitted spherical thrust blocks to relieve the joint of what must be considerable stress. In order to transmit this thrust to the main frame it is the general practice to fit massive and deep cross frames behind the gearboxes and to support the rear end of the gearboxes from these frames.

The triangular torque rod introduced some years ago still has its adherents, although the Renault, which had it first, has been modified in this respect. The same type of rod is still employed, but it is now in the form a stiff tube mounted in the axle casing. This arrangement would appear to be preferable to the triangular torque rods, met with in a few cases, where the tubes have been set five or six inches out of straight to clear brake actuations, etc.

Radius Rods.

The number of firms fitting a radius rod on each side of the axle is steadily increasing. Since they no longer drive through the rear springs the radius rods have become necessary, and these are generally ball-ended to allow freedom of motion in any direction.

The Delaunay-Belleville has a novel method of taking the torque and drive. A triangular torque rod is mounted on the axle casing at the right side about midway between the centre of the casing and the frame. The front end of the rod is not mounted in a swivelling bracket as is common practice with this device, but is secured firmly to the chassis. The rod therefore acts as a radius rod. The second radius rod is provided on the other side of the axle casing in a corresponding position to that occupied by the torque rod. The portion which encircles the axle tube is mounted on a spherical bearing to allow free movement to the rear axle with respect to the frame. This device obviates the necessity of using three members by making the torque rod do double duty, and appears to be a sound design.

As a general rule, this type of torque rod is secured as nearly as possible in a line with the universal joint at the rear of the gearbox. The effect is to give the universal joint on the axle little to do other than to allow the propeller shaft to telescope into the joint, whilst the front universal joint takes up all the motion due to the angularity of the gear box shaft and the propeller shaft. This

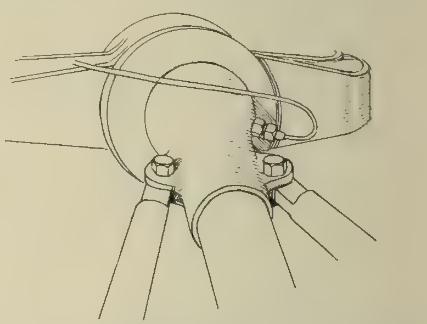
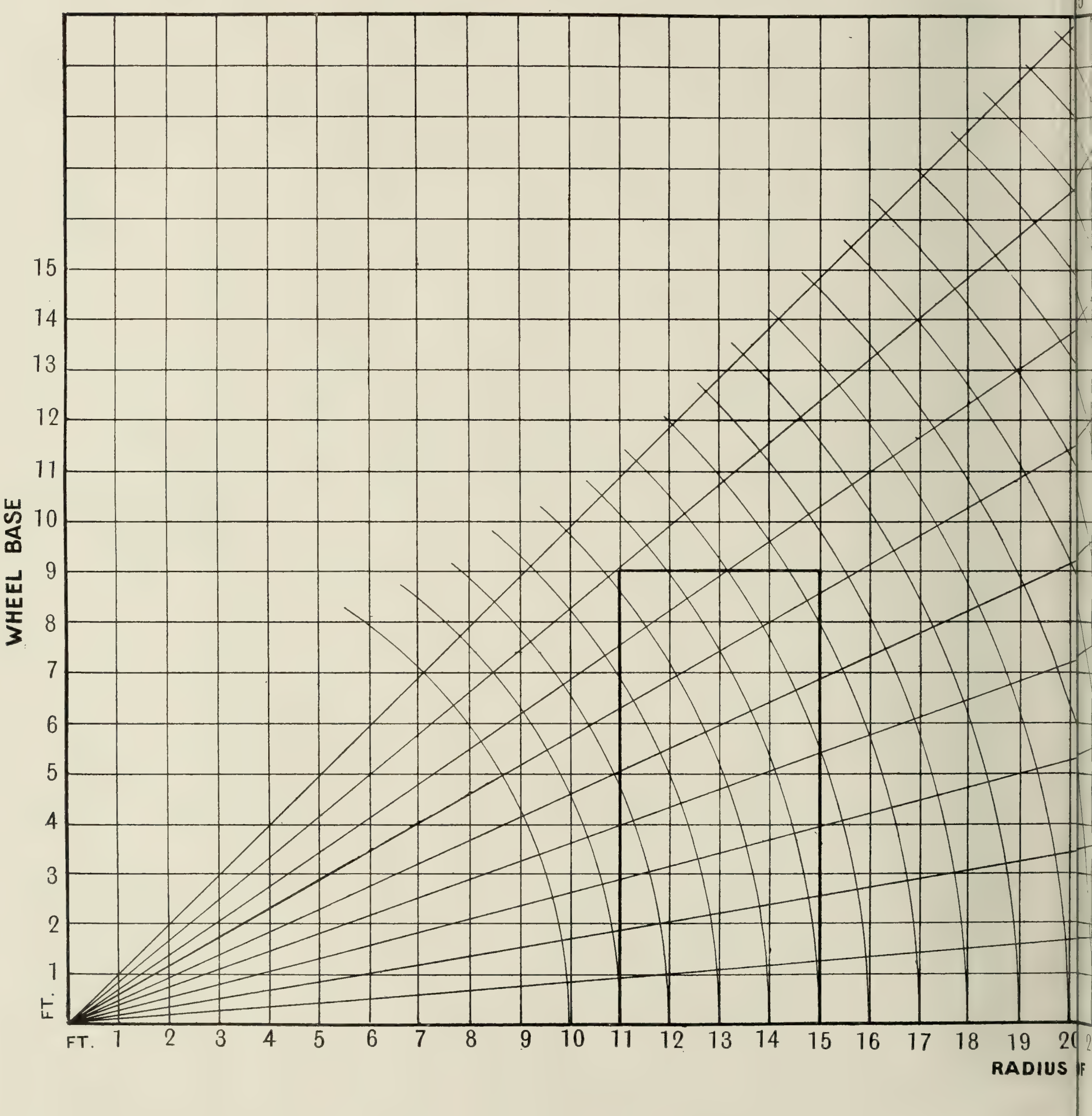


Fig. VII.—The ball ended torque rod on Mercedes chassis.

is not a correct theoretical motion, for a single universal joint transforms a constant angular velocity to an irregular one in a shaft inclined at a small angle to it. It is certain that this effect is not evident in the case of the small angle at which the shafts are inclined, although several French firms went to considerable trouble a few years ago to constrain the axes of the bevel pinion and the rear shaft of the gearbox to remain parallel and thus obtain the same velocities in both. This has now disappeared, but the same result is obtained in a much simpler manner on the Austin. The torque rod in this case is anchored at a point midway between the two universal joints, and when the springs rebound there is always the same angle between the gear box shaft and the propeller shaft as there is between this latter and the bevel pinion shaft. The small power Napier cars had torque rods mounted in this way last year, but this year's model has the front of the torque rod in line with the gear box universal joint. That this would appear to be the best arrangement for the constant velocity of the shafts is merely a theoretical point, whilst the short torque rod does undoubtedly tend to lift the rear of the chassis when the clutch is let in too suddenly.

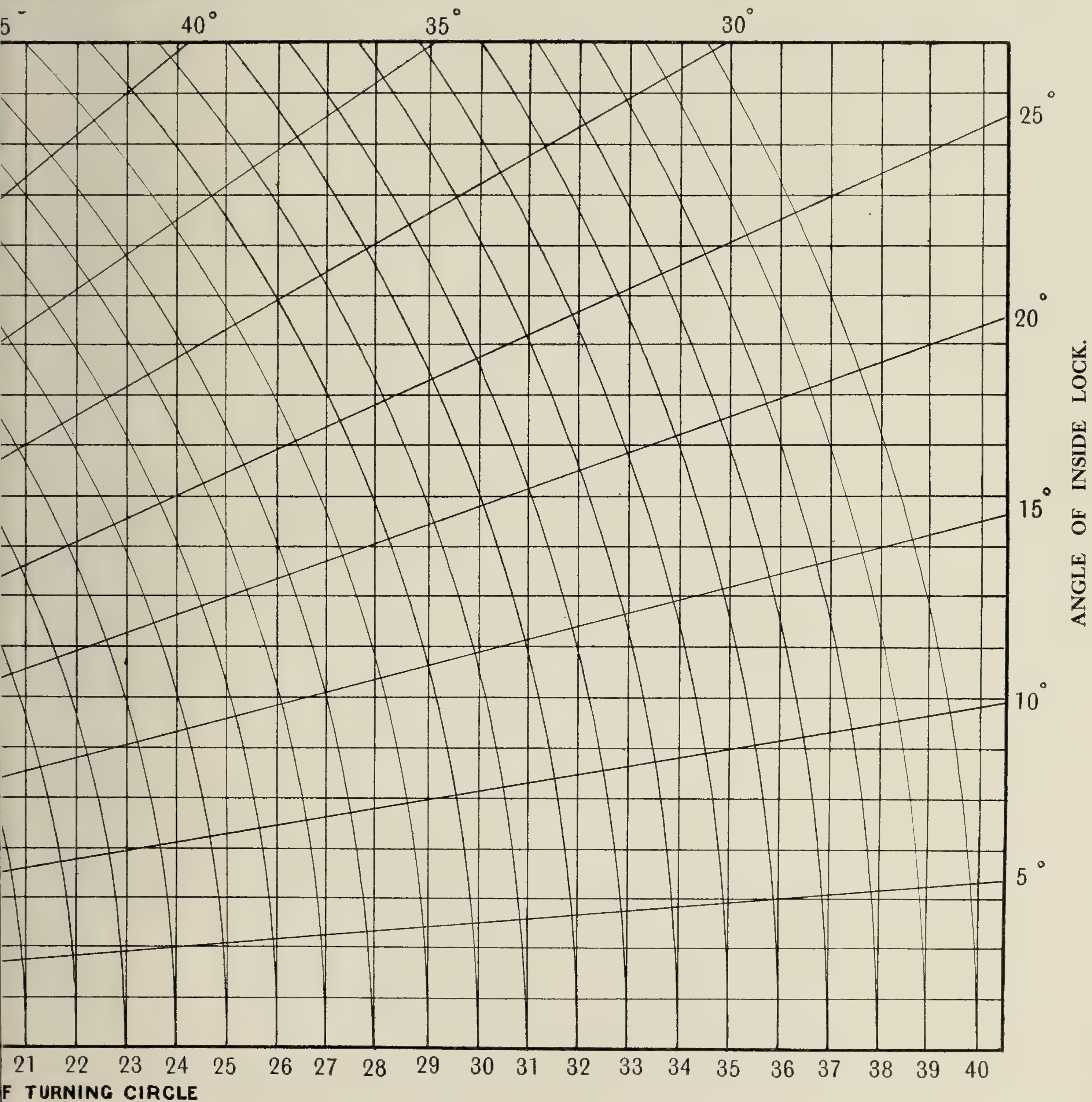
GRAPHIC METHOD OF DETERMINING THE TURN

To find the turning circle first follow the horizontal line giving the wheelbase till the line corresponding to back along the horizontal a distance equal to the track and mark this point. From the two points found t diameter of the circle in which the chassis should turn. In the example shown by thick lines the wheelbase the necessary angle of lock for any required turning c



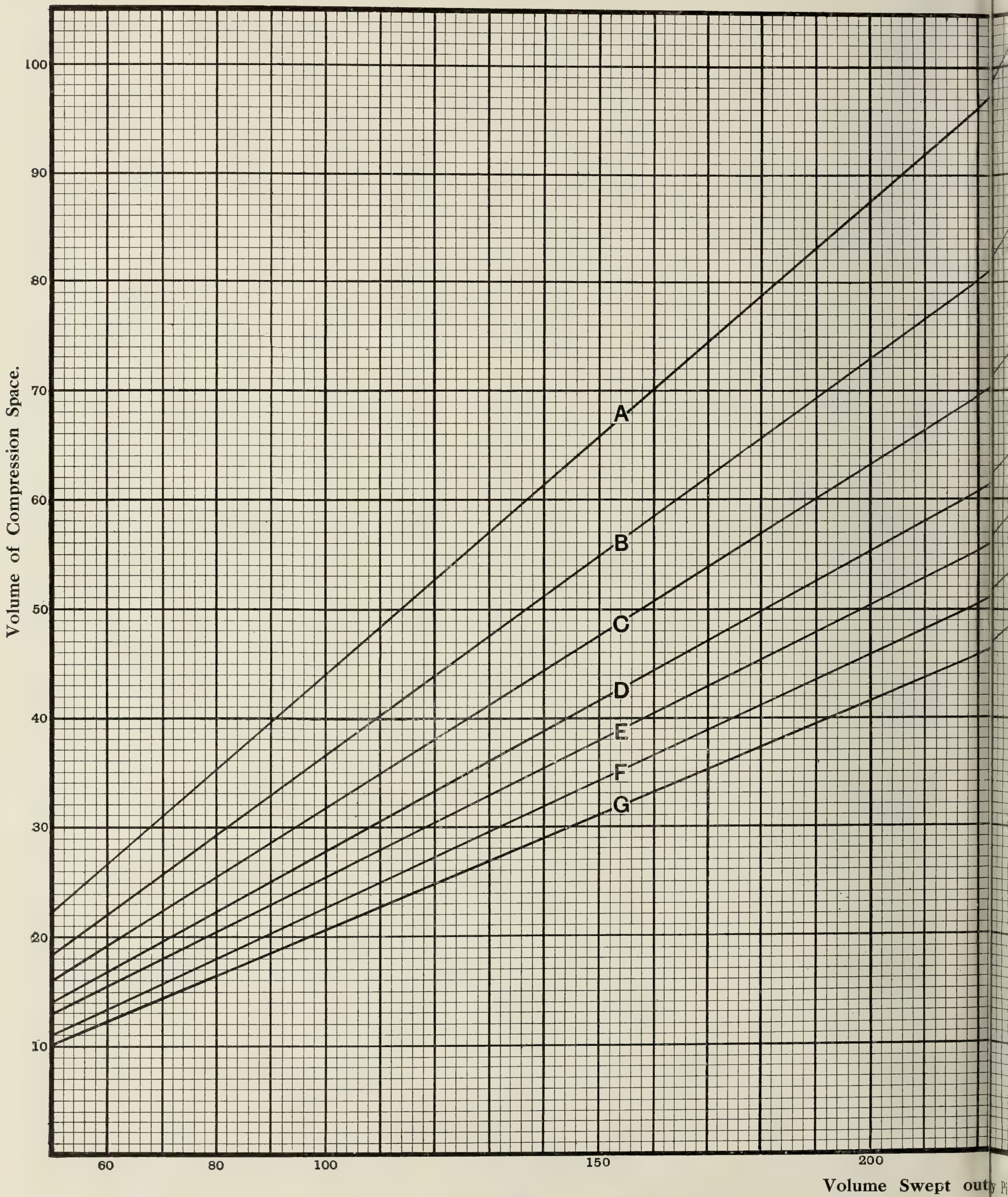
[NG CIRCLE FROM ANY CHASSIS PROPORTIONS.

he angle of inside lock is reached. Mark the point and let fall a perpendicular to the base line. Also set
s, follow the arcs of circles to the base line, and the sum of the radii found by this means is equal to the
: nine feet, the track four feet, and the turning circle 17 ft. 6 ins. plus 14 ft. 3 ins., or 31 ft. 9 ins. Conversely,
cle may be obtained with any given wheelbase and track.



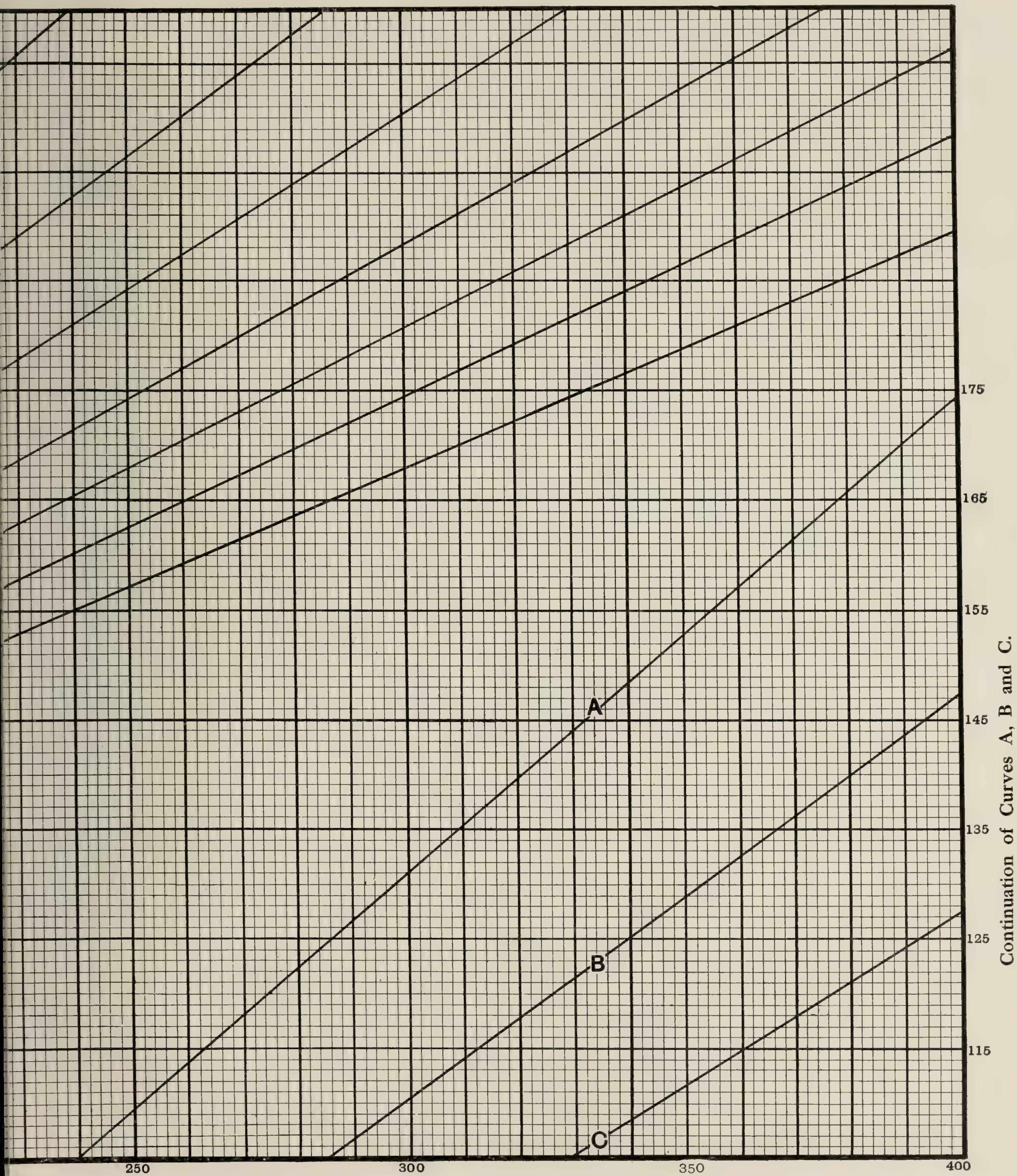
COMPARISON OF CYLINDER VOLUMES, COMPRESSI

To find the volume of compression space to give any desired compression follow the vertical line from
The curves correspond to the following pressures, and are calculated to be intermediate between the ad



N SPACE VOLUMES, AND COMPRESSION PRESSURE.

cylinder volume to the requisite curve and then read off compression chamber volume on vertical scale.
atic and isothermal values:—A=60, B=70, C=80, D=90, E=100, F=110 and G=120, pounds per square inch.



Piston per Stroke.

THE PRINCIPAL DETAILS OF TWO HUNDRED AND SIXTY FIVE 1911 PATTERN PLEASURE CAR CHASSIS.

The classification is by order of the total cylinder volume, and the chassis are divided into five classes: Class A, consisting of cars with engines with a total cubic capacity of less than 2,000 cc. Class B, 2,000—3,000 cc. Class C, 3,000—4,500 cc. Class D, 4,500—6,000 cc. and Class E, over 6,000 cc.

Abbreviations used in the tables are as follows:—

Under the heading "Lubrication," "Splash" means all systems in which the oil level is not controlled automatically. "Forced" means a system with no splash, unless stated otherwise. "Trough" means that the oil is fed to gutters into which scoops on the big ends dip, so distributing the oil to all parts.

Under "Control," "T." means Throttle and "I." Ignition.

Under "Type of Clutch," "P." means all varieties of plate or disc. "L.C." Leather Cone, "M.C." Metal Cone. * Epicyclic gear. "E." Expanding metal to metal. and † Contracting band brake type.

Gear Ratios are given in terms of the number of engine revolutions to one of the road wheels.

Under "Springs," "½" means half elliptic, "¾" three-quarter elliptic, "¼" quarter elliptic, and "T.R.S." Transverse Rear Spring.

CLASS A.—Cars with engines in which the total volume swept out by the pistons per rev. does not exceed 2,000 c.c.

Name and Nominal Horse-power of Car.	Bore.	Stroke.	Volume Swept out by Pistons.	Ratio of Stroke to Bore.	Volume of Compression Space.	No. of Cylinders	No. of Cylinders Cast Together	Valve Arrangement.	Cooling.	Lubrication.	Control.	Weight of Piston.	Petrol Feed.	Type of Clutch.	Gear Ratios. 1st.	2nd.	3rd.	4th.	Size of Wheels.	Wheel-base.	Track.	Turning Circle.	Type of Final Drive.	Type of Rear Springs.
	mm.	mm.	c.c.		c.c.							lb. oz.							mm.	ft. in.	ft. in.	ft.		
Rover 6	97	110	812	1.1	184	1	1	Same side	Pump	Splash	Throttle	3 15	Gravity	P.	16	9	5		700	6 0	4 0	33	Bevel	½
Renault 8	75	120	1058	1.6	—	2	2	Same side	Thermo	Splash	Throttle	—	Gravity	L.C.	15	7.5	1.37		700	7 0	3 9	27	Bevel	½
Rover 8	101	130	1062	1.3	272	1	1	Knight	Pump	Trough	Throttle	3 9	Gravity	P.	17.4	9.6	5.3		750	7 0	4 1	—	Bevel	½
N.S.U. 10	75	125	1102	1.6	—	4	4	Same side	Thermo	Forced	Throttle	—	Gravity	L.C.	—	—	—		750	7 0	3 9	—	Bevel	½
Austin 10	63	89	1107	1.4		4	1	Opposite	Pump	Forced	T. and I.	—	Gravity	L.C.	16	7.6	4.5		700	8 0	4 0	—	Bevel	½
N.S.U. 10	60	100	1128	1.6	—	2	2	Same side	Thermo	Forced	T. and I.	—	Gravity	L.C.	—	—	—	—	750	7 0	3 9	—	Bevel	½
Opel 10	65	90	1191	1.4	—	4	4	Same side	Thermo	Splash	T. and I.	—	Gravity	L.C.	—	—	—	—	700	6 11	3 9	28	Bevel	½
Charron 8	80	120	1204	1.5	—	2	2	Same side	Thermo	Forced	Throttle	—	Gravity	L.C.	—	—	—	—	810	8 8	4 4	—	Bevel	¾
Delahaye 10	80	120	1204	1.5	—	2	2	Same side	Pump	Forced	Throttle	—	Gravity	L.C.	18	10	5	—	760	8 7	4 2	—	Bevel	¾
Phœnix 8-10	90	100	1272	1.1	—	2	2	Same side	Thermo	Splash	T. and I.	—	Gravity	P.	11.7	7	4.2	—	700	7 0	4 0	25	Chain	½
Thames 8	114	127	1295	1.1	—	1	1	Same side	Thermo	Splash	Throttle	—	Gravity	M.C.	—	—	—		—	—	—	—	Worm	½
Rover 8	114	130	1326	1.1	370	1	1	Same side	Pump	Splash	T. and I.	5 10	Gravity	P.	13.3	7.4	4.1	—	750	7 0	4 1	34	Bevel	½
Martini 12	70	90	1382	1.3	—	4	4	Same side	Thermo	Splash	Throttle	1 4	Gravity	P.	14.8	8.2	4.4	—	750	7 9	3 11	26	Bevel	¾
N.A.G. 12-14	75	85	1499	1.1	—	4	4	Same side	Thermo	Forced	Throttle	—	Gravity	L.C.	—	—	—	—	760	8 2	3 10	26	Bevel	¾
Brasier 10-12	90	120	1526	1.3	—	2	2	Same side	Thermo	Splash	Throttle	—	Gravity	L.C.	—	—	—	—	760	8 3	3 11	—	Bevel	¾
Hurtu 10	70	100	1536	1.4	455	4	4	Same side	Thermo	Splash	Throttle	1 9	Gravity	L.C.	12.8	7.14	4.4	—	700	7 11	4 0	34	Bevel	¾
A.G.R. 10	70	100	1536	1.4	455	4	4	Same side	Thermo	Splash	Throttle	1 9	Gravity	L.C.	—	—	—		750	8 3	3 10	32	Bevel	¾
Opel 14	64	120	1540	1.9	-	4	4	Same side	Thermo	Splash	Throttle	—	Gravity	L.C.	—	—	—		750	8 6	4 6	32	Bevel	¾
Brasier 11-15	67	110	1548	1.6	—	4	4	Same side	Thermo	Splash	Throttle	—	Gravity	L.C.	—	—	—		760	8 3	3 11	—	Bevel	¾
N.S.U. 14	75	88	1552	1.2	—	4	4	Same side	Thermo	Forced	Throttle	—	Gravity	L.C.	—	—	—		760	8 9	3 9	—	Bevel	½
D.F.P. 10-12	65	120	1588	1.8	—	4	4	Same side	Thermo	Splash	Throttle	—	Gravity	L.C.	—	—	—	—	750	7 11	4 1	—	Bevel	¾
Alldays 10-12	95	114	1614	1.2	416	2	2	Opposite	Thermo	Forced	T. and I.	2 6	Gravity	L.C.	—	—	—	—	760	8 0	4 0	—	Bevel	½
Standard 12	68	114	1633	1.7	—	4	4	Same side	Thermo	Forced	T. and I.	—	Pressure	L.C.	19	9.5	5		800	9 0	4 6	—	Worm	½
De Dion 12	66	120	1641	1.8	—	4	4	Same side	Thermo	Forced	Throttle	—	Gravity	P.	—	—	—	—	810	{8 9 / 9 6}	4 2	—	Bevel	¾
N.B. 10	90	130	1653	1.4	—	2	2	Same side	Thermo	Forced	—	—	Gravity	P.	—	—	—	—	810	8 0	4 2	—	Bevel	¾
Renault 10	70	110	1689	1.6	—	4	4	Same side	Thermo	S. & F.	Throttle	—	Gravity	L.C.	17	9	4.9	—	800	8 3	4 4	28	Bevel	¾
Daimler 12	69	114	1700	1.8	—	4	2	Knight	Pump	Trough	T. and I.	—	Pressure	L.C.	—	—	—	—	870	9 8	4 4	—	Worm	½
Adler 12	75	103	1716	1.3	—	4	2	Same side	Pump	Forced	—	—	Gravity	P.	—	—	-	—	810	8 10	4 3	—	Bevel	—
Le Gui 10	65	130	1721	2	—	4	4	‡	Thermo	Forced	—	—	Gravity	—	—	—	—	-	—	8 6	4 4	—	Bevel	—
Adams 10	122	152	1722	1.2	420	1	1	Same side	Pump	Forced	Throttle	4 3	Gravity	*	14.5	7	3.7	—	760	7 11	4 2	30	Chain	Full
Darracq 10	68	120	1742	1.8	—	4	2	Same side	Pump	Forced	Throttle	—	Gravity	L.C.	12.8	7.4	4.15	—	750	7 8	4 1	—	Bevel	¾
Imperia 12	75	100	1764	1.3	—	4	4	Same side	Thermo	Splash	Throttle	—	Gravity	P.	13	8	4	—	760	9 1	4 2	—	Bevel	—
Mors 10	80	90	1807	1.1	—	4	4	Same side	Pump	Forced	Throttle	—	Gravity	†	—	—	—	—	810	9 7	4 3	—	Bevel	¾
Brasier 12-18	70	120	1843	1.7	—	4	4	Same side	Thermo	Splash	Throttle	—	Gravity	P.	—	—	—	—	765	9 4	4 1	—	Bevel	¾
D.F.P. 15	70	120	1843	1.7	—	4	4	Same side	Thermo	Splash	Throttle	—	Gravity	L.C.	—	—	4	—	750	8 11	4 4	—	Bevel	¾
Mercédès 15	70	120	1843	1.7	—	4	2	‡	Pump	Forced	T. and I.	—	Pressure	M.C.	—	—	—		810	8 11	4 3	—	Bevel	½
Opel 16	70	120	1843	1.7	—	4	4	Same side	Thermo	Splash	Throttle	—	Gravity	L.C.	—	-	—	—	760	9 0	4 2	35	Bevel	¾
Sizaire 12	70	120	1843	1.7	—	4	4	‡	Thermo	Splash	Throttle	—	Gravity	P.	—	—	—	—	750	8 7	4 6	26	Bevel	Full
Rover 12	96	130	1879	1.4	—	2	2	Knight	Pump	Trough	T. and I.	2 8	Gravity	P.	14.6	8.1	4.5	—	810	8 10	4 2	—	Worm	½

CLASS A.—Cars with engines in which the total volume swept out by the pistons per rev. does not exceed 2,000 c.c.—*Continued.*

Name and Nominal Horse-power of Car.	Bore.	Stroke.	Volume Swept out by Pistons.	Ratio of Stroke to Bore.	Volume of Compression Space.	No. of Cylinders	No. of Cylinders Cast Together	Valve Arrangement.	Cooling.	Lubrication.	Control.	Weight of Piston.	Petrol Feed.	Type of Clutch.	Gear Ratios. 1st.	2nd.	3rd.	4th.	Size of Wheels.	Wheel-base.	Track.	Turning Circle.	Type of Final Drive.	Type of Rear Springs.
	mm.	mm.	c.c.		c.c.							lb. oz.							mm.	ft. in.	ft. in.	ft.		
Phœnix 10-12	102	115	1879	1.04	596	2	2	Same side	Thermo	Splash	T. and I.	2 6	Gravity	P.	11	6.6	4		760	8 0	4 0	27	Chain	T.R.S.
Deasy 12	75	110	1940	1.5	—	4	2	Same side	Thermo	Splash	T. and I.	—	Gravity	P.	13.6	7.6	4.2	—	810	9 3	4 2		Worm	3/4
Delahaye 12-16	75	110	1940	1.5	—	4	4	Same side	Pump	Forced	Throttle	—	Gravity	L.C.	17	10	6	—	815	{8 8 / 9 5}	4 5	—	Bevel	3/4
Humber 12	75	110	1940	1.5		4	4	Same side	Thermo	Forced	—	—	Gravity	L.C.	—	—	—	-	810	8 10	4 9	—	Bevel	—
Itala 15	75	110	1940	1.5	—	4	4	Opposite	Pump	Forced	Throttle	—	Pressure	P.	—		—		810	8 6	4 6	—	Bevel	3/4
Métallurgique 14	75	110	1940	1.5	—	4	4	Same side	Thermo	Forced	Throttle	—	Pressure	E.	—	-			810	9 4	4 4	—	Bevel	3/4
Unic 12-14	75	110	1940	1.5	—	4	4	Same side	Pump	Forced	Throttle	—	Gravity	L.C.	—	—		—	810	9 4	4 5	-	Bevel	—
Zedel 14	72	120	1953	1.7	—	4	4	Same side	Pump	Forced	Throttle	—	Gravity	P.	—	—		—	710	7 8	3 9	-	Bevel	3/4
Argyll 12	72	120	1953	1.7	560	4	4	Same side	Thermo	Forced	Throttle	1 7	Gravity	P.	16.1	9.2	6.2	4.1	760	9 0	4 3		Worm	3/4
Austrian-Daimler 12-15	76	110	1993	1.4	—	4	4	Same side	Pump	Forced	T. and I.	—	Gravity	L.C.	14	7	4.1		810	8 8	4 9	28	Bevel	1/2
Peugeot 10-14	70	130	1996	1.8	—	4	4	Same side	Pump	Splash	Throttle		Gravity	L.C.	—	—	—		810	9 1	4 1	—	Bevel	1/2

CLASS B.—Cars with engines in which the total volume swept out by the pistons per rev. is over 2,000 c.c. and under 3,000 c.c.

Name and Nominal Horse-power of Car.	Bore.	Stroke.	Volume Swept out by Pistons.	Ratio of Stroke to Bore.	Volume of Compression Space.	No. of Cylinders	No. of Cylinders Cast Together	Valve Arrangement.	Cooling.	Lubrication.	Control.	Weight of Piston.	Petrol Feed.	Type of Clutch.	Gear Ratios. 1st.	2nd.	3rd.	4th.	Size of Wheels.	Wheel-base.	Track.	Turning Circle.	Type of Final Drive.	Type of Rear Springs.
De Dion 14	75	120	2116	1.6	—	4	2	Same side	Pump	Forced	T. and I.	—	Gravity	P.	—	-	—		810	{8 10 / 9 7}	4 2	—	Bevel	3/4
Lorraine-Dietrich 12-16	75	120	2116	1.6	—	4	4	Same side	Thermo	Splash	Throttle	—	Gravity	L.C.	-	-	—		810	9 3	4 7	—	Bevel	3/4
Mass 10-12	75	120	2116	1.6	—	4	4	Same side	Thermo	Forced	Throttle	—	Gravity	L.C.	14	7	4		750	8 2	4 0	25	Bevel	1/2
Mors 10-12	75	120	2116	1.6	—	4	4	Same side	Pump	Forced	Throttle	—	Gravity	—	—		—		760	8 9	4 0	—	Bevel	3/4
Gladiator or Vinot 12-14	80	110	2209	1.4	—	4	4	Same side	Pump	Forced	Throttle	—	Gravity	L.C.	--	—	---		810	9 2	4 3	35	Bevel	3/4
Austrian-Daimler 16-18	80	110	2209	1.4	—	4	4	Same side	Pump	Forced	T. and I.	—	Pressure	L.C.	15	8	5.4	3.7	815	9 2	4 1	33	Bevel	1/2
Motobloc 12	80	110	2209	1.4	—	4	2	Same side	Pump	Splash	Throttle	—	Gravity	P.	-	—	-		810	9 7	4 1	30	Bevel	3/4
Davy 12-18	89	89	2214	1	—	4	2	Piston	Thermo	Trough	Throttle	—	Gravity	L.C.	—	—	—		810	8 6	4 4	—	Bevel	3/4
Itala 14-16	77	120	2232	1.6	-	4	4	Same side	Pump	Forced	T. and I.		Pressure	P.	-	—	—		810	8 8	4 6	—	Bevel	3/4
Iris 15	80	114	2289	1.4	—	4	4	Same side	Thermo	Trough	T. and I.		Pressure	L.C.	13.9	7.25	4		815	9 4	4 6	30	Bevel	1/2
Gobron 12-16	90	180	2289	1	760	2	2	—	Pump	—	Throttle	2 8	Gravity	L.C.	—	-	—		810	7 0	3 6	42	Bevel	3/4
Calthorpe 12-14	75	130	2293	1.7	508	4	4	Same side	Thermo	Forced	T. and I.	1 12	Gravity	P.	9.9	6	3.5		760	8 2	4 2	27	Bevel	3/4
La Buire 10	75	130	2293	1.7	—	4	4	Same side	Pump	Forced	Throttle	—	Gravity	P.	—	-	-		810	9 1	4 5	24	Bevel	3/4
Lancia 12-16	75	130	2293	1.7	—	4	4	Same side	Pump	Forced	—	-	Pressure	P.	—	—	—		815	—	—	—	Bevel	3/4
Wolseley 12-16	79	114	2309	1.8	657	4	4	Same side	Thermo	F. & T.	T. and I.	1 10½	Gravity	P.	14	7.4	4.3		810	9 0	4 0	—	Worm	1/2
Minerva 16	82	110	2323	1.3	556	4	4	Knight	Pump	Trough	T. and I.	1 12	Gravity	L.C.	11.4	6.7	4.8	3.6	815	9 4	4 2	34	Bevel	3/4
Crossley 12-14	79	120	2352	1.4	705	4	4	Same side	Thermo	Forced	Throttle	1 7	Pressure	†	15.3	9.3	5.5	3.78	810	8 11	4 6	30	Bevel	3/4
Leon Bollée 14-20	83	110	2380	1.3	—	4	4	Same side	Thermo	Forced	T. and I.	2 4	Gravity	L.C.		—	3.7		815	9 3	4 6	40	Bevel	3/4
Hillman 12-15	89	96	2388	1.7		4	2	Opposite	Pump	Trough	Throttle	—	Gravity	L.C.	11.8	6.9	4		760	{9 0 / 9 8}	4 5	25	Bevel	3/4
Vulcan 15.9	80	120	2409	1.5	—	4	2	Same side	Thermo	Forced	Throttle	2 0	Gravity	L.C.	—	—			760	8 5	4 5	28	Worm	3/4
Thames 15.9	80	120	2409	1.5	—	4	4	Same side	Thermo	Trough	Throttle	—	Pressure	P.	—	—			—	—	—		Bevel	1/2
Talbot 12	80	120	2409	1.5	—	4	2	Same side	Pump	Forced	T. and I.	—	Gravity	L.C.	14.6	9	6	4	810	9 0	4 2	—	Bevel	T.R.S.
Sunbeam 12-16	80	120	2409	1.5		4	2	Opposite	Pump	Forced	Throttle	—	Gravity	L.C.	13.2	8.7	5.9	4	810	{9 5 / 10 0}	4 6	29	Worm	3/4
Star 12	80	120	2409	1.5	615	4	2	Same side	Pump	Forced	Throttle	1 13	Gravity	L.C.	12	6	3.6	3	810	9 10	4 3	—	Bevel	3/4
Panhard 12-15	80	120	2409	1.5	-	4	4	Same side	Thermo	Forced	Throttle	—	Gravity	L.C.	16.5	8.5	5		815	9 6	4 8	-	Bevel	3/4
Mors 12-16	80	120	2409	1.5	—	4	2	Same side	Thermo	Forced	Throttle	—	Gravity	†	—	—	—		815	10 2	4 7	—	Bevel	3/4
Martini 15.8	80	120	2409	1.5	—	4	4	Same side	Pump	F. & T.	T. and I.	—	Pressure	P.	15.8	6.9	4.2	3.7	810	8 10	4 3	27	Bevel	3/4
Dodson 12-16	80	120	2409	1.5	—	4	2	Same side	Thermo	Splash	Throttle	—	Gravity	L.C.	—	—	-		810	9 1	4 5	—	Bevel	3/4
Darracq 15	80	120	2409	1.5	—	4	2	Same side	Pump	Forced	Throttle	—	Gravity	L.C.	12.8	7.4	4.15		810	9 2	4 0	—	Bevel	3/4
Charron 15.8	80	120	2409	1.5	—	4	4	Same side	Thermo	Forced	Throttle	—	Gravity	L.C.	—		—		810	9 4	4 4	30	Bevel	3/4
Berliet 15	80	120	2409	1.5	—	4	2	Same side	Thermo	Forced	Throttle	—	Gravity	P.		—	—		810	9 7	4 5	—	Bevel	1/2
Benz 15	80	120	2409	1.5	—	4	2	—	—	—	—	—	—	—		—	—		—	—	—	—	—	—
Arrol-Johnston 15.9	80	120	2409	1.5	720	4	2	Same side	Thermo	Splash	Throttle	1 2	Gravity	P.	14.7	8.9	6.1	4	815	9 6	4 5	30	Bevel	Full
Armstrong-Whitworth 15.9	80	120	2409	1.5	—	4	4	Same side	Thermo	Forced	Throttle	—	Pressure	P.		—	—		810	{8 3 / 9 3}	4 6	{27 / 28}	Bevel	3/4
Argyll 15	80	120	2409	1.5	672	4	2	Same side	Thermo	—	Throttle	1 13	Gravity	P.	15.6	9.3	6.25	9.12	810	9 8	4 6	—	Bevel	3/4
Rochet-Schneider 15.8	80	120	2409	1.5	—	4	4	Same side	Pump	Forced	Throttle	—	Gravity	L.C.	—	-	—		815	9 7	4 5	—	Bevel	—
Piccard-Pictet 15.8	80	120	2409	1.5	—	4	4	Same side	Thermo	Forced	Throttle		Gravity	P.	—	—	—		810	9 4	4 4	—	Bevel	—
Valveless 15	112	127	2462	1.16	732	2	2	None	Thermo	Forced	T. and I.	4 8	Pressure	L.C.	11	7.5	5	3.5	815	9 3	4 8	30	Bevel	1/2
Lanchester 20	101	76	2483	0.75	—	4	1	Opposite	Thermo	Forced	T. and I.	—	Pressure	P.	{15 / 16.5}	7.6 / 8.5	{4.3 / 4.8}	—	875	{9 5 / 10 5}	4 10	45	Worm	1/4
Rover 15	85	110	2486	1.3	650	4	4	Same side	Pump	Forced	T. and I.	2 6	Gravity	P.	13.3	7.4	4.1		810	9 1	4 4	—	Bevel	1/2
Alldays 14	86	108	2505	1.26	630	4	2	Same side	Thermo	Forced	Throttle	2 4	Gravity	L.C.	—	—	—		810	9 0	4 6	—	Bevel	3/4
Rothwell 15	79	127	2510	1.61	875	4	2	Same side	Thermo	Forced	T. and I.	1 13	Gravity	L.C.	9.5	5.07	2.95		810	9 6	4 3	35	Bevel	T.R.S.
Zedel 20	82	120	2534	1.46	—	4	2	Same side	Pump	Forced	Throttle	—	Gravity	P.	—	—	—		815	9 2	4 5	—	Bevel	3/4
N.S.U. 20	85	115	2542	1.35	—	4	2	Opposite	Pump	Splash	T. and I.	—	Pressure	P.	—	—	—		815	10 0	4 3	—	Bevel	1/2
Humber 12	90	100	2544	1.11	—	4	2	Opposite	—	—	T. and I.	—	Gravity	L.C.	—	—	—	—	—	—	—	—	—	—

CLASS B.—Cars with engines in which the total volume swept out by the pistons per rev. is over 2,000 c.c. and under 3,000 c.c.—*Continued.*

Name and Nominal Horse-power of Car.	Bore.	Stroke.	Volume Swept out by Pistons.	Ratio of Stroke to Bore.	Volume of Compression Space.	No. of Cylinders.	No. of Cylinders Cast Together.	Valve Arrangement.	Cooling.	Lubrication.	Control.	Weight of Piston.	Petrol Feed.	Type of Clutch.	Gear Ratios. 1st.	2nd.	3rd.	4th.	Size of Wheels.	Wheel-base.	Track.	Turning Circle.	Type of Final Drive.	Type of Rear Springs.
	mm.	mm.	c.c.		c.c.							lb. oz.							mm.	ft. in.	ft. in.	ft.		
Delage	66	125	2565	1.89	—	6	6	Same side	Thermo	Forced	Throttle	·	Gravity	—	—	—	—		810	9 4	4 5	—	Bevel	½
N.A.G. 20	83	120	2596	1.44	—	4	2	Opposite	Thermo	Splash	Throttle	—	Gravity	L.C.	—	—	—		820	9 10	4 7	26	Bevel	¾
Daimler 15	80	130	2610	1.62	—	4	2	Knight	Pump	Trough	T. and I.	—	Pressure	L.C.	—	—	—		870	9 8	4 4	—	Worm	½
Deasy 14-20	80	130	2610	1.62	—	4	2	Same side	Thermo	Forced	Throttle	—	Gravity	P.	15.08	9.43	5.66	3.77	815	10 2	4 3	—	Worm	¼
F.I.A.T. 15	80	130	2610	1.62	—	4	4	Same side	—	Splash	Throttle	—	Gravity	P.	—	—	—		815	9 0	4 7	—	Bevel	—
Germain 15	80	130	2610	1.62	—	4	4	‡	Thermo	Forced	T. and I.	—	Pressure	E.	—	—	—		810	9 9	4 2	—	Bevel	¾
Singer 15	80	130	2610	1.62	—	4	2	Same side	Thermo	F. & T.	T. and I.	—	Gravity	L.C.	12.1	6.6	4.53	3.5	815	9 6	4 6	—	Bevel	¾
Peugeot 12-15	80	130	2610	1.62	—	4	2	Same side	Pump	Forced	Throttle	—	Gravity	L.C.	—	—	—	—	815	9 10	4 2	—	Bevel	½
Opel 20	80	130	2610	1.62	—	4	4	Same side	Thermo	Splash	Throttle	—	Gravity	L.C.	—	—	—	—	815	9 6	4 3	—	Bevel	¾
Brasier 15-22	80	130	2610	1.62	—	4	4	Same side	Thermo	Splash	Throttle	—	Gravity	P.	—	—	—	—	815	—	—	—	Bevel	¾
Mercédès 20	80	130	2610	1.62	—	4	2	‡	Pump	Forced	T. and I.	—	Pressure	—	14.2	7.5	4.7	3.35	820	9 11	4 5	44	Bevel	½
Calthorpe 15	75	150	2646	2	588	4	2	Same side	Thermo	F. & T.	T. and I.	1 10	Gravity	P.	9.3	5.5	3.3		810	9 0	4 2	33	Bevel	¾
Gobron 15	75	150	2646	2	880	4	—	—	Pump	Forced	Throttle	—	Gravity	L.C.	—	—	—		815	7 0	4 4	42	Bevel	—
Le Gui 15	75	150	2646	2	—	4	4	‡	Thermo	Forced		—	Gravity		—	—	—		750	8 6	4 4	—	Bevel	—
Standard 16	89	108	2674	1.21	—	4	2	Same side	Thermo	Forced	T. and I.	—	Pressure	L.C.	16	8.25	4		810	10 6	4 8	—	Bevel	½
S.C.A.T. 15	85	120	2712	1.41	—	4	4	Same side	Pump	Forced	Throttle	—	Forced	P.	13.3	7.9	6.1	3.2	810	9 6	4 3	—	Bevel	¾
Adams 16	85	120	2712	1.41	—	4	2	Same side	Thermo	Forced	Throttle	1 13	Gravity	L.C.	{17.1 {25.7	10.4 9.6	6.7} 4.0}	4	810	10 0	4 7	36	Bevel	½
Armstrong-Whitworth 17.9	85	120	2712	1.41	—	4	4	Same side	Thermo	Forced	T. and I.	—	Pressure	P.	12	7.8	4.8	3	815	9 8	4 6	30	Bevel	¾
Napier 15	82	127	2759	1.55	—	4	4	Same side	Thermo	Forced	Throttle	—	Gravity	P.	—	—	—		815	8 10	4 8	25	Bevel	½
Dennis 18	90	110	2798	1.22	—	4	2	Opposite	Thermo	Forced	Throttle	—	Gravity	P.	15	8.5	4.6	3.6	810	9 8	4 6	35	Worm	½
Austin 15	89	114	2836	1.28	—	4	1	Opposite	Pump	Forced	T. and I.	—	Pressure	L.C.	11.9	6.35	4.8	3	815	9 3	4 6	—	Bevel	Full
Straker-Squire 15	87	120	2851	1.38	—	4	4	Same side	Thermo	Forced	T. and I.	—	Pressure	L.C.	12.8	6.85	4.06		810	9 3	4 4	37	Bevel	½
N.E.C. 20	127	114	2886	0.91	—	2	1	Same side	Thermo	Forced	T. and I.	—	Gravity	L.C.	—	—	—	—	875	9 6	4 7	—	Worm	¼
Delahaye 16-20	85	130	2938	1.53	—	4	4	Same side	Pump	Forced	Throttle	—	Gravity	L.C.	15.5	8	5	4	875	—	—	—	Bevel	¾
Bentall 16-20	100	95	2983	0.95	—	4	1	Opposite	Pump	Splash	Throttle	—	Gravity	C.	—	—	—		810	{9 3} {10 3}	4 4	40	Bevel	¾
Crowdy 19.6	89	120	2985	1.35	—	4	2	Same side	Thermo	Forced	—	—	Gravity	L.C.	14.6	8.8	4	—	815	9 6	4 6	—	Worm	½

CLASS C.—Cars with engines in which the total volume swept out by the pistons per rev. is over 3,000 c.c. and under 4,500 c.c.

Name and Nominal Horse-power of Car.	Bore.	Stroke.	Volume Swept out by Pistons.	Ratio of Stroke to Bore.	Volume of Compression Space.	No. of Cylinders.	No. of Cylinders Cast Together.	Valve Arrangement.	Cooling.	Lubrication.	Control.	Weight of Piston.	Petrol Feed.	Type of Clutch.	Gear Ratios. 1st.	2nd.	3rd.	4th.	Size of Wheels.	Wheel-base.	Track.	Turning Circle.	Type of Final Drive.	Type of Rear Springs.
Wolseley 16-20	90	120	3052	1.33	983	4	2	Same side	Pump	Trough	T. and I.	2 11	Pressure	P.	15	7.6	5.1	3.78	{815 {820	9 5} 10 1}	4 4	—	Worm	½
Vauxhall 20	90	120	3052	1.33	—	4	4	Same side	Thermo	Forced	T. and I.	2 9	Gravity	P.	13.2	8.4	5.3	3.6	810	10 3	4 6	—	Bevel	½
Unic 16-20	90	120	3052	1.33	—	4	2	Opposite	Pump	Forced	Throttle	—	Gravity	L.C.	—	—	—	—	815	10 1	4 5	—	Bevel	Full
Star 15	90	120	3052	1.33	623	4	2	Same side	Pump	Forced	T. and I.	1 13	Gravity	L.C.	12	6	3.6	3	810	9 10	4 3	—	Bevel	¾
Imperia 18	90	120	3052	1.33	—	4	4	Same side	Thermo	—	Throttle	—	Gravity	P.	10.3	6.6	3.6		815	9 5	4 4	—	Bevel	
De Dion 18	90	120	3052	1.33	—	4	2	Same side	Pump	Forced	Throttle	—	Gravity	P.	—	—	—		875	10 7	4 7	—	—	¾
Bell 16	90	120	3052	1.33	509	4	2	Same side	Pump	F. & T.	Throttle	3 0	Pressure	E.	14.4	7.2	3.6		815	9 6	4 6	35	Bevel	½
Belsize 14-16	90	120	3052	1.33	771	4	2	Opposite	Thermo	Trough	T. and I.	2 9	Gravity	P.	11.2	5.6	3.6		810	8 9	4 3	30	Bevel	¾
Deasy 18-24	90	120	3052	1.33	—	4	2	Same side	Thermo	—	T. and I.	—	Gravity	P.	13.2	8.25	4.6	3.3	820	10 10	4 6	—	Worm	¼
Piccard-Pictet 18-22	90	120	3052	1.33	—	4	2	Same side	Pump	Forced	—	—	—	P.	—	—	—		815	9 6	4 7	—	Bevel	¾
Hotchkiss 16-20	95	110	3115	1.16	—	4	4	Same side	Pump	—	Throttle	—	Gravity	L.C.	—	—	—	—	875	10 0	4 5	—	Bevel	¾
Davy 18-24	89	127	3159	1.43	—	4	2	Same side	Thermo	Forced	Throttle	—	Gravity	P.	—	—	—		815	9 6	4 6	—	Bevel	¾
La Buire 15	85	140	3164	1.64	—	4	4	Same side	Pump	Forced	Throttle	—	Pressure	P.	—	—	—		815	8 10	4 5	23	Bevel	¾
Alldays 20-25	95	114	3228	—	428	4	2	Opposite	Thermo	Forced	Throttle	2 8	Gravity	L.C.	—	—	—	—	815	{9 6} {10 6}	4 6	—	Bevel	T.R.S.
Singer 20	90	130	3307	1.44	—	4	2	Same side	Thermo	F. & T.	T. and I.	—	Gravity	L.C.	{10.3 {12 1	5.7 6.6	3.9 4.5	3 3.5	815 820	9 6} 10 6}	4 6	—	{Bevel {Worm	¾ ¾
Maudslay 17	90	130	3307	1.44	918	4	2	—	Thermo	Forced	T. and I.	2 7	Pressure	L.C.	10.9	6.8	4.5	3.14	875	9 9	4 6	35	Bevel	¾
Martini 16-24	90	130	3307	1.44	—	4	2	Same side	Pump	Splash	T. and I.	1 7½	Pressure	P.	11	6.5	4.3	3.5	815	9 10	4 5	28	Bevel	¾
Lorraine-Dietrich 15-20	90	130	3307	1.44	—	4	4	Same side	Thermo	Splash	Throttle	—	Gravity	L.C.	—	—	—	—	875	10 6	4 8	—	Bevel	¾
Motobloc 16	80	110	3313	1.39	—	6	3	—	Pump	Splash	Throttle	—	Gravity	P.	—	—	—		820	10 6	4 6	—	Bevel	¾
Valveless 25	133	140	3389	1.05	1030	2	2	None	Pump	Forced	Throttle	4 14	Pressure	L.C.	10	6.2	3.2		820	10 0	4 8	36	Bevel	½
Mass 15	95	120	3398	1.26	—	4	1	Opposite	Pump	Forced	T. and I.	—	Gravity	L.C.	9.75	7.25	3.3		810	9 6	4 3	30	Bevel	¾
Clément-Talbot 15	90	140	3561	1.55	—	4	2	Same side	Pump	Forced	T. and I.	—	Gravity	L.C.	12.8	7.3	4.7	3.4	815	9 9	4 8	—	Bevel	T.R.S.
Métallurgique 20	90	140	3561	1.55	—	4	2	Same side	Thermo	Forced	T. and I.	2 4	Pressure	P.	—	—	—	—	820	10 5	4 5	—	Bevel	¾
Brasier 18-30	90	140	3561	1.55	—	4	2	Same side	Thermo	Splash	Throttle	—	Gravity	P.	—	—	—	—	875	{9 8} {10 4}	4 7	—	Bevel	¾
Argyll 20	90	140	3561	1.55	1020	4	2	Same side	Thermo	—	T. and I.	2 13	Gravity	P.	13	7.6	5	3.5	875	10 2	4 7	—	Bevel	¾
Mercédès 30	90	140	3561	1.55	—	4	2	—	Pump	Forced	T. and I.	—	Pressure	—	14.2	7.5	4.7	3.35	820	9 11	4 5	44	Bevel	½
Germain 18	102	110	3594	1.08	—	4	1	Opposite	Pump	Forced	T. and I.	—	Pressure	L.C.	—	—			815	9 9	4 0	—	Bevel	¾
Renault 18-25	80	120	3614	1.5	—	6	3	Same side	Thermo	Forced	Throttle	—	Gravity	L.C.	14	10	6	3.5	880	10 1	4 7	40	Bevel	¾
Vulcan 23.5	80	120	3614	1.5	—	6	2	Same side	Thermo	Forced	T. and I.	2 0	Gravity	L.C.	—	—	—	—	870	9 8	4 6	—	Worm	¾
Clément-Talbot 20	80	120	3614	1.5	—	6	2	Same side	Pump	Forced	T. and I.	—	Gravity	L.C.	12.8	7.3	4.7	3.4	820	10 2	4 7	—	Bevel	T.R.S.
Sunbeam 18-22	80	120	3614	1.5	—	6	3	Opposite	Pump	Forced	Throttle		Gravity	L.C.	15.3	8.6	6.4	3.5	815	{11 2} {10 1}	4 7	—	Worm	¾
Panhard 18-24	80	120	3614	1.5	—	6	6	Same side	Pump	Forced	Throttle		Gravity	P.	—	—	—	--	880	10 3	4 8	—	Bevel	¾

CLASS C.—Cars with engines in which the total volume swept out by the pistons per rev. is over 3,000 c.c. and under 4,500 c.c.—*Continued.*

Name and Nominal Horse-power of Car.	Bore.	Stroke.	Volume Swept out by Pistons.	Ratio of Stroke to Bore.	Volume of Compression Space.	No. of Cylinders	No. of Cylinders Cast Together	Valve Arrangement.	Cooling.	Lubrication.	Control.	Weight of Piston.	Petrol Feed.	Type of Clutch.	Gear Ratios. 1st.	2nd.	3rd.	4th.	Size of Wheels.	Wheel-base.	Track.	Turning Circle.	Type of Final Drive.	Type of Rear Springs.
	mm.	mm.	c.c.		c.c.							lb. oz.							mm.	ft. in.	ft. in.	ft.		
Mors 17-20	80	120	3614	1.5	—	6	2	Same side	Pump	Forced	Throttle	—	Gravity	T.	-	-	-		880	10 5	4 7	-	Bevel	¾
Darracq 20	80	120	3614	1.5	—	6	3	Same side	Pump	Forced	Throttle	—	Gravity	L.C.	12	7.9	5.1	3.2	815	10 0	4 6	-	Bevel	¾
Charron 24	80	120	3614	1.5	—	6	3	Same side	Thermo	Forced	Throttle	—	Gravity	L.C.	—	—	—	—	815	10 5	4 10	33	Bevel	¾
Arrol-Johnston 23.9	80	120	3614	1.5	1080	6	2	Same side	Thermo	Forced	Throttle	1 2	Gravity	P.	14.7	8.9	6.08	4	820	11 1	4 5	—	Bevel	Full
Charron 22-24	95	130	3681	1.36	—	4	4	Same side	Thermo	Forced	T. and I.	—	Gravity	L.C.	—		—	—	815	10 6	4 10	33	Bevel	¾
Delahaye 20-30	95	130	3681	1.36	—	4	4	Same side	Pump	Forced	T. and I.	-	Gravity	L.C.	14.5	8	4.75	3.5	880	10 0	4 6	—	Bevel	¾
Rochet-Schneider 18	95	130	3681	1.36	—	4	4	Same side	Thermo	Forced	T. and I.	—	—	L.C.	—		-	—	815	10 3	4 5	—	Bevel	¾
Lanchester 28	102	76	3725	0.74	—	6	1	Opposite	Thermo	Forced	T. and I.	-	Pressure	P.	12.7 / 15.7	5.8 / 7.6	3.8 / 4.3	—	880	10 5 / 11 5	4 10	45	Worm	¼
Thornycroft 18	102	114	3758	1.12	1232	4	4	Same side	Thermo	Forced	Throttle	2 12	Pressure	P.	—	-			815	9 8	4 5	48	Worm	½
Berliet 22	100	120	3768	1.2	—	4	2	Opposite	Pump	Forced	Throttle	-	Pressure	P.	—	-	-	—	880	10 6	4 9	—	Bevel	½
Armstrong-Whitworth 25	100	120	3768	1.2	—	4	1	Opposite	Pump	Forced	Throttle	-	Pressure	P.	—	—		—	820	10 10	4 6	—	Bevel	¾
Unic 24-30	102	116	3790	1.14	—	4	2	Opposite	Pump	Forced	Throttle	—	Gravity	P.	-	-		—	880	10 5 / 11 0	4 7	—	Bevel	½
Peugeot 15-20	90	150	3816	1.67	—	4	—	Same side	Pump	Forced	T. and I.	—	Gravity	L.C.	—	—	—	—	880	10 5	4 7	—	Bevel	½
Calthorpe 20	90	150	3816	1.67	1696	4	2	Same side	Thermo	Forced	T. and I.	2 0	Gravity	P.	9	5.4	3.3		870	9 6 / 10 0	4 6	38	Bevel	¾
Mors 17-20	95	135	3823	1.42	—	4	2	Opposite	Pump	Forced	Throttle	—	Gravity	†	—	—	—		880	10 5	4 7	—	Bevel	¾
Sunbeam 16-20	95	135	3823	1.42	—	4	2	Opposite	Pump	Forced	Throttle	—	Gravity	L.C.	15.3	8.6	6.4	3.5	815	10 1 / 10 7	4 7	—	W or B	¾
Daimler 23	80	130	3915	1.63	—	6	2	Knight	Pump	Trough	T. and I.	-	Pressure	L.C.	—	—	4.25		880	10 4	4 4		Worm	½
Leon Bollée 18-25	98	130	3920	1.32	—	4	4	Opposite	Pump	Trough	T. and I.	4 12	Gravity	L.C.	—	—	—	3.5	880	10 0	4 8	42	Bevel	½
Standard 20	89	108	4043	1.21	—	6	2	Same side	Pump	Forced	T. and I.	—	Pressure	P.	16	8.25	4		870	10 6	4 8	—	Bevel	½
Adams 30	85	120	4068	1.41	1014	6	2	Same side	Thermo	F. & T.	T. and I.	—	Gravity	L.C.	15 / 22.6	9.3 / 8.5	5.8 / 3.5	3.5	880	12 0	4 7	43	Bevel	½
Panhard 18-30	100	130	4082	1.3	—	4	1	Opposite	Pump	Forced	T. and I.	-	Gravity	P.	13	7	4.5	3.3	880	10 3	4 8	—	Bevel	¾
Opel 25	100	130	4082	1.3	—	4	2	Same side	Thermo	Splash	T. and I.	—	Pressure	L.C.	—	—	—	—	880	10 4	4 3	40	Bevel	¾
Mercédès 40	100	130	4082	1.3	—	4	2	Knight	Pump	Trough	—	—	Pressure	—	—	—	—		820	10 3	4 5	—	—	
Lancia 24	100	130	4082	1.3	—	4	4	Same side	Pump	Forced	—	—	Pressure	P.	13.08	7.94	5.4	3.26	820	9 0 / 10 6	4 4	—	Bevel	¾
De Dion 25	100	130	4082	1.3	—	4	2	Same side	Pump	Forced	Throttle	—	Gravity	P.	—	—	—	—	880	10 11	4 7	—	—	¾
Deasy 24-30	100	130	4082	1.3	—	4	2	Same side	Thermo	Trough	Throttle	—	Gravity	P.	13.2	8.25	4.95	3.3	880	10 10	4 6		Worm	¼
Minerva 26	102	125	4085	1.22	956	4	2	Knight	Pump	Forced	T. and I.	2 10	Gravity	L.C.	10	5.9	4.3	3.1	880	10 0	4 7	—	Bevel	¾
Napier 30	82	127	4122	1.55	—	6	2	Same side	Pump	Forced	Throttle	—	Pressure	P.	—	—	—		880	10 0	4 8	41	Bevel	¾
Rothwell 20	102	127	4150	1.24	1816	4	2	Same side	Thermo	Forced	Throttle	2 14	Gravity	L.C.	9.5	5.07	2.95		810	9 6	4 3	36	Bevel	T.R.S.
Vinot 25-30	101	130	4165	1.28	—	4	2	Same side	Pump	Forced	Throttle	—	Gravity	L.C.	—		3.5		820	10 0	4 8	35	Bevel	¾
Daimler 25	101	130	4165	1.28	—	4	2	Knight	Pump	Trough	T. and I.	—	Pressure	L.C.	—	—	4.2 / 3.3		920	10 4	4 7	—	Worm	½
Wolseley 20-28	102	130	4248	1.27	1344	4	2	Same side	Pump	T. & F.	T. and I.	—	Pressure	P.	12.5	7	4.8	3.2	880	10 0 / 10 6	4 6	—	Bevel	T.R.S.
Hotchkiss 20-30	95	101	4248	1.07	—	6	2	Same side	Pump	Splash	Throttle	—	Gravity	L.C.	—	—	—	—	880	11 2	4 7	—	Bevel	½
Sheffield-Simplex 25	85	127	4294	1.5	976	6	2	Same side	Thermo	Trough	T. and I.	2 0½	Pressure	P.	12.5 / 13.4	6.6 / 7	3.95 / 4		880	10 6 / 11 3	4 8	41	Bevel	½
Dodson 20-30	100	140	4396	1.4	—	4	2	Same side	Thermo	Splash	Throttle		Gravity	L.C.	—	—	—	—	880	10 2	4 7	-	Bevel	¾
Renault 20-30	100	140	4396	1.4	—	4	2	Same side	Thermo	Splash	Throttle		Gravity	L.C.	13	7	5	3.07	880	10 2	4 7	40	Bevel	¾
Panhard 25	100	140	4396	1.4	—	4	1	Knight	Pump	Forced	T. and I.	—	Pressure	P.	10.5	6	4	3	880	10 3	4 8	—	Bevel	¾
Piccard-Pictet 22-30	100	140	4396	1.4	—	4	2	—	Pump	Forced	—	—	—	P.	—	—	—	—	880	8 8 / 10 0	4 7	—	Bevel	—
Mors 20-30	100	140	4396	1.4	—	4	2	Same side	Pump	Forced	Throttle	—	—	†	—	—	—		880	10 4	4 7	—	Bevel	¾
Darracq 22	100	140	4396	1.4	—	4	2	Same side	Pump	Forced	Throttle	—	Gravity	L.C.	12	7.9	5.1	3.2	815	10	4 6	—	Bevel	¾
Berliet 20-25	100	140	4396	1.4	—	4	2	Same side	Thermo	Forced	Throttle	—	Pressure	P.	—	—	—	—	880	10 4	4 9	—	Bevel	½
Austrian-Daimler 25-30	105	130	4498	1.24	—	4	2	Opposite	Pump	Forced	T. and I.	—	Pressure	P.	10.6	8	3.4	3	880	10 3	4 4	35	Bevel	½

CLASS D.—Cars with engines in which the total volume swept out by the pistons per rev. is over 4,500 c.c. and under 6,000 c.c.

Name and Nominal Horse-power of Car.	Bore.	Stroke.	Volume Swept out by Pistons.	Ratio of Stroke to Bore.	Volume of Compression Space.	No. of Cylinders	No. of Cylinders Cast Together	Valve Arrangement.	Cooling.	Lubrication.	Control.	Weight of Piston.	Petrol Feed.	Type of Clutch.	Gear Ratios. 1st.	2nd.	3rd.	4th.	Size of Wheels.	Wheel-base.	Track.	Turning Circle.	Type of Final Drive.	Type of Rear Springs.
Crowdy 30	110	120	4560	1.1	—	4	2	Opposite	Pump	Splash	—	—	Pressure	P.	6.75	6	4.5	3	875	9 6	4 7	—	Bevel	½
Crossley 20	102	140	4575	1.37	—	4	2	Same side	Thermo	Forced	Throttle	—	—	P.	13.5	7.5	4.1	3.4	875	9 9	4 6	33	Bevel	—
Bell 20	102	140	4575	1.37	747	4	2	Opposite	Pump	Trough	Throttle	4 2	Pressure	P.	12.4	6.2	3		820	9 6	4 6	35	Bevel	½
Talbot 25	102	140	4575	1.37	—	4	2	Same side	Pump	Forced	T. and I.	—	Gravity	L.C.	11.3	6.5	4.2	3.05	820	9 10	4 7	—	Bevel	T.R.S.
S.C.A.T. 22	102	140	4575	1.37	—	4	2	Same side	Pump	Forced	Throttle	—	Pressure	P.	9.9	5.5	3.5	2.8	875	10 6	4 5	—	Bevel	¾
Gobron 20-30	90	180	4579	2	1520	4	2	Same side	Pump	Forced	Throttle	—	Gravity	—	—	—	—	—	880	10 9	4 7	—	Bevel	—
Imperia 28	106	130	4586	1.22	—	4	—	—	—	Splash		—	Pressure	—	9	5.9	3.25	—	880	10 1	4 4		Bevel	½
Leon Bollée 24-30	106	130	4586	1.22	—	4	2	Opposite	Pump	Trough	T. and I.	5 4	Gravity	L.C.	—	—	—	3	880	10 0	4 8	42	Bevel	½
N.E.C. 30	114	114	4651	1	—	4	1	Same side	Thermo	Forced	T. and I.	—	Gravity	L.C.	—	—	—	—	920	10 7	4 11	—	Worm	¼
Sunbeam 25-30	105	135	4666	1.29	—	4	2	Opposite	Pump	Forced	T. and I.	—	Gravity	L.C.	15.3	8.6	6.4	3.5	820	10 1 / 10 7	4 7	—	Worm	¾

CLASS D.—Cars with engines in which the total volume swept out by the pistons per rev. is over 4,500 c.c. and under 6,000 c.c.—*Continued.*

Name and Nominal Horse-power of Car.	Bore.	Stroke.	Volume Swept out by Pistons.	Ratio of Stroke to Bore.	Volume of Compression Space.	No. of Cylinders	No. of Cylinders Cast Together	Valve Arrangement.	Cooling.	Lubrication.	Control.	Weight of Piston.	Petrol Feed.	Type of Clutch.	Gear Ratios. 1st.	2nd.	3rd.	4th.	Size of Wheels.	Wheel-base.	Track.	Turning Circle.	Type of Final Drive.	Type of Rear. Springs
	mm.	mm.	c.c.		c.c.							lb. oz.							mm.	ft. in.	ft. in.	ft.		
Dennis 24	100	150	4710	1.5	—	4	2	Opposite	Thermo	Forced	Throttle	—	Pressure	L.C.	9.6	5.6	3.57		815	10 4	4 8	40	Worm	½
La Buire 24	85	140	4746	1.65	—	6	6	Same side	Pump	Forced	Throttle	—	Pressure	P.	—	—	—	—	875	11 6	4 7	35	Bevel	¾
Brasier 24-40	85	140	4746	1.65	—	6	3	Same side	Thermo	F. & S.	Throttle	—	Gravity	P.		—	—	—	880	10 3	4 7	—	Bevel	¾
Brooke 25	92	120	4780	1.32	—	6	—	Opposite	Pump	Forced	Throttle	—	Pressure	P.	—	—	—	—	875	10 6	4 8	—	Bevel	½
Alldays 30	95	114	4842	1.47	—	6	2	Opposite	Thermo	Forced	Throttle	2 7	Gravity	L.C.	—		—	—	880	{10 3 / 11 3}	4 9	—	Bevel	T.R.S.
Rochet-Schneider 25	105	140	4844	1.33	—	4	4	Same side	Pump	Forced	—	—	—	L.C.	—	—	—	—	880	10 10	4 7	—	—	—
Iris 25	108	133	4873	1.23	—	4	2	Same side	Pump	Forced	T. and I.	3 2	Pressure	P.	9.7	5.3	2.9		880	10 6	4 8	—	Bevel	½
Vauxhall 30	90	120	4884	1.33	—	6	3	Same side	Thermo	Forced	T. and I.	—	—	P.	9	5.9	3.25		880	10 1	4 4	—	Bevel	—
Belsize 18-22	90	120	4884	1.33	1935	6	2	Opposite	Thermo	Trough	T. and I.	2 9	Gravity	—	11 2	5.6	3 6		815	10 0	4 8	34	Bevel	¾
Métallurgique 26	102	150	4902	1.47	—	4	2	Same side	Thermo	Forced	T. and I.	3 12	Pressure	E.	—			—	880	11 6	4 8	—	Bevel	¾
Austin 18-24	102	127	4912	1.24	—	4	1	Opposite	Pump	Forced	T. and I.	—	Pressure	L.C. or P.	8.6	5.15	3.6	2.6	880	9 11	4 7	—	Bevel	Full
Charron 30	110	130	4940	1.18	—	4	1	Opposite	Pump	Forced	Throttle	—	Gravity	L.C.	—		—	—	880	9 4	4 5	—	Bevel	T.R.S.
Hotchkiss 20-30	110	130	4940	1.18	—	4	2	Same side	Pump	Splash	Throttle	—	Gravity	L.C.	—		—	—	880	10 6	4 7	—	Bevel	½
Imperia 24-30	110	130	4940	1.18	—	4	—	—	—	F. & S.	—	—	—	P.	11.5	5.5	4	3	880	9 7	4 8	—	—	—
Mass 20	110	130	4940	1.18	—	4	1	Opposite	Pump	Forced	T. and I.	—	Gravity	L.C.	7.5	4	2.5		875	9 6	4 3	32	Bevel	¾
Lanchester 38	102	102	4958	1	—	6	1	Opposite	Thermo	Forced	T. and I.	—	Pressure	P.	12.7	5.8	3.8		895	{10 7 / 11 7}	4 10	45	Worm	¼
Wolseley 24-30	90	120	4960	1.33	1572	6	2	Same side	Pump	T. & F.	T. and I.	2 11½	Pressure	P.	12.57	7.03	4.8	3.2	880	10 9	4 6	—	Bevel	T.R.S.
Crowdy 29-34	89	127	4979	1.42	—	6	2	Same side	Thermo	Forced	—	—	Gravity	L.C.	12.8	7.6	3.5		820	10 9	4 6	—	Worm	½
Peugeot 22-30	100	150	5024	1.5	—	4	—	Same side	Pump	Forced	Throttle	—	Gravity	L.C.	—	—	—	—	880	10 10	4 9	—	Bevel	½
Maudslay 25-30	114	127	5181	1.11	1676	4	2	—	Thermo	Forced	T. and I.	4 0	Pressure	L.C.	10.23	6.18	4.14	3.3	880	10 6	4 9	46	Bevel	¾
N.A.G. 40	115	125	5190	1.09	—	4	2	Opposite	Pump	Splash	T. and I.	—	Pressure	L.C.	—	—	—	—	880	10 4	4 7	26	Bevel	¾
Thornycroft 30	114	127	5293	1.11	1860	4	2	Opposite	Pump	Forced	T. and I.	4 0	Pressure	P.	—	—	—		880	10 7	4 8	41	Bevel	¾
Argyll 30	90	140	5342	1.1	1020	6	2	Same side	Pump	Trough	T. and I.	2 13	Gravity	P.	13	7.6	5	3.5	875	10 11	4 7	—	Bevel	¾
Wolseley 30-34	117	127	5574	1.08	1700	4	2	Same side	Pump	F. & T.	T. and I.	6 14	Pressure	P.	9.3	4.9	3.1	2.3	880	10 6	4 6	—	Bevel	T.R.S.
N.E.C. 40	127	114	5772	0.9	—	4	1	Same side	Thermo	Forced	T. and I.	—	Gravity	L.C.	—	—	—	—	920	{10 7 / 11 6}	4 11	—	Worm	¼
Gladiator 35-40	115	140	5812	1.22	—	4	1	Opposite	Pump	Splash	T. and I.	—	Pressure	P.	—	—	—	3.25	880	10 6	4 8	—	Bevel	¾
Brasier 35-40	120	130	5881	1.08	—	4	2	Same side	Thermo	Splash	Throttle	—	Gravity	P.	—	—	—	—	880	{9 4 / 10 2}	4 7	—	Chain	¾
Germain 28	120	130	5881	1.08	—	4	1	Opposite	Pump	Forced	T. and I.	—	Pressure	E.	—	—	—	—	820	10 3	4 0	—	Bevel	¾

CLASS E.—Cars with engines in which the total volume swept out by the pistons per rev. is over 6,000 c.c.

Name and Nominal Horse-power of Car.	Bore.	Stroke.	Volume Swept out by Pistons.	Ratio of Stroke to Bore.	Volume of Compression Space.	No. of Cylinders	No. of Cylinders Cast Together	Valve Arrangement.	Cooling.	Lubrication.	Control.	Weight of Piston.	Petrol Feed.	Type of Clutch.	Gear Ratios. 1st.	2nd.	3rd.	4th.	Size of Wheels.	Wheel-base.	Track.	Turning Circle.	Type of Final Drive.	Type of Rear. Springs
Mors 28-35	114	150	6120	1.32	—	4	2	Opposite	Pump	Forced	Throttle	—	Gravity	†	—	—	—	—	880	10 5	4 9	—	Chain	¾
Standard 40	101	127	6225	1.25	—	6	2	Same side	Pump	Forced	T. and I.	—	Pressure	P.	14	7.25	3.5		875	11 0	4 10	—	Bevel	½
Napier 45	101	127	6225	1.25	—	6	2	Same side	Pump	Forced	Throttle	—	Pressure	P.	—	—	—		895	11 2	4 8	46	Bevel	T.R.S.
Daimler 38	101	130	6247	1.28	—	6	2	Knight	Pump	Trough	T. and I.	—	Pressure	L.C.	—	—	{4.25 / 3.3}		920	{11 6 / 10 2}	4 7	—	Bevel	½
Minerva 38	124	130	6276	1.05	1712	4	2	Knight	Pump	Trough	T. and I.	—	Gravity	L.C.	7.8	4.6	3.3	2.4	880	10 3	4 7	39	Bevel	¾
Germain 25	106	180	6350	1.7	—	4	1	—	Pump	Forced	T. and I.	—	Pressure	—	—	—		—	820	10 0	4 7	—	Bevel	¾
Hillman 25	127	127	6431	1	—	4	1	Opposite	Pump	—	T. and I.	—	Gravity	L.C.	7.8	3.9	2.5		820	10 6	4 7	—	Bevel	¾
Maudslay 35-45	127	127	6431	1	2012	4	2	—	Thermo	Forced	T. and I.	5 0	Pressure	L.C.	10.23	6.18	4.14	3.3	935	10 6	4 9	46	Bevel	¾
Dennis 40	127	130	6583	1.03	—	4	1	Opposite	Pump	Forced	Throttle	—	Pressure	L.C.	8.6	4.6	3.1	2	880	10 6	4 6	—	Worm	¾
Brooke 40	108	120	6595	1.12	—	6	—	Same side	Pump	Forced	Throttle	—	Pressure	P.	—		—	—	880	10 6	4 8	—	Bevel	½
Iris 35	127	123	6735	1.05	—	4	2	Same side	Pump	Forced	T. and I.	4 4	Pressure	L.C.	8.2	4.5	2.4		880	11 0	4 8	42	Bevel	½
Métallurgique 40	125	140	6871	1.12	—	4	2	Same side	Thermo	Forced	T. and I.	4 12	Pressure	E.	—	—	—	—	880	11 6	4 8	—	Bevel	¾
Austrian-Daimler 50-60	120	154	6966	1.28	—	4	2	Same side	Pump	Forced	Throttle	—	Pressure	P.	11.3	5.9	3.8	2.8	935	11 0	4 7	38	—	½
Sheffield-Simplex 45	114	114	7038	1	1560	6	2	Same side	Pump	F. & T.	T. and I.	3 5	Pressure	P.	{10 / 6.7}	{5.3 / 3.3}	3	—	880	{11 3 / 12 0}	4 8	44	Bevel	T.R.S.
Wolseley 40	127	140	7089	1.1	2225	4	2	Same side	Pump	F. & T.	T. and I.	5 15	Pressure	P.	9.2	4.7	3.16	2.4	895	11 0	4 8	—	Bevel	T.R.S.
De Dion 35	90	140	7123	1.55	—	8	2	Same side	Pump	Forced	Throttle		Gravity	P.	—	—	—		—	—	—	—	Bevel	¾
Crowdy 39-45	127	140	7149	1.1	—	4	2	Opposite	Pump	Splash	—	—	Pressure	P.	6	5.5	4	2.5	880	9 11	4 9	—	Bevel	¾
Austin 50	111	127	7368	1.14	—	6	1	Opposite	Pump	Forced	T. and I.	—	Pressure	P.	7.3	4.3	3	2.2	880	11 6	4 6	—	Bevel	Full
Rolls-Royce 40-50	114	120	7344	1.05	—	6	3	Same side	Pump	Forced	T. and I.	—	Pressure	L.C.	—	—	—		895	11 10	4 8	—	Bevel	¾
Brasier 50-70	112	130	7683	1.16	—	6	3	Same side	Thermo	Forced	T. and I.	—	Gravity	P.	—	—	—		920	{10 10 / 11 6}	4 7	—	Chain	¾
Thornycroft 45	114	127	7909	1.11	2620	6	2	Opposite	Pump	Forced	Throttle	4 0	Pressure	P.	—	—	—		880	11 9	4 8	47	Bevel	¾
Gobron 40-60	114	220	8160	1.75	2620	4	2	Same side	Pump	Forced	Throttle	—	Gravity	M.C.	—	—	—	—	920	10 9	4 7	—	Chain	—
N.A.G. 60	130	160	8492	1.23	—	4	2	Opposite	Pump	Splash	T. and I.	—	Pressure	L.C.	—	—	—	—	895	10 4	4 7	26	Bevel	½
Wolseley 50	114	146	8935	1.28	2620	6	2	Same side	Pump	F. & T.	T. and I.	4 10	Pressure	P.	9.21	4.7	3.16	2.44	935	11 7	4 8	—	Bevel	T.R.S.
Mors 50-60	114	150	9180	1.32	—	6	2	Opposite	Pump	Splash	Throttle	—	Gravity	†	—	—	—	—	880	11 7	4 9	—	Chain	¾
Daimler 57	124	130	9414	1.05	—	6	2	Knight	Pump	Trough	T. and I.	—	Pressure	L.C.	—	—	—		920	11 6	4 4	—	Bevel	½
Gobron 50	110	250	9500	2.26	3160	4	—	—	Pump	Forced	Throttle	—	Gravity	M.C.	—	—	—	—	935	10 9	4 7	54	Chain	—
Napier 65	127	127	9646	1	—	6	2	Same side	Pump	Forced	Throttle	—	Pressure	P.	—	—	—		895	11 2	4 8	48	Bevel	—
Dennis 60	127	130	9874	1	—	6	1	Opposite	Pump	Forced	Throttle	—	Pressure	L.C.	8.6	4.6	3.1	2.1	895	11 6	4 6	50	Worm	¾
Napier 90	155	127	16561	0.82	—	6	1	Same side	Pump	Forced	Throttle	—	Pressure	P.	—	—	—		935	11 11	4 8	50	Bevel	½

INVENTION IN 1910.

Being progress as represented chiefly by recent automobile patents.

By Eric W. Walford, F.C.I.P.A.

IT is often urged that patenting of features is of little value, especially as patterns change from year to year. It should be noted that a man who gives sufficient study to his work to bring out a new design altogether is surely in a rank above his fellows. Therefore, a man who gives much study to any one feature, as may be recognised from patents, may be considered somewhat of a specialist in this direction. Patented features therefore represent the greatest improvements that are made in any industry from time to time, and that progress is represented very largely by features patented from time to time cannot be doubted, but it is few who have the facilities to examine even a moderate number of the patent specifications which are published by the Patent Office. As I endeavour to examine every patent specification relating to automobiles I frequently come across inventions which are of great interest, but which do not come before the public or students of automobile engineering. It is not the intention of this article to call attention to all the most promising of recently patented inventions nor in any sense to prophesy. It is merely intended to take the different parts of the modern motor car in sequence and to call attention to such special features as have been the subject recently of important motor car patents.

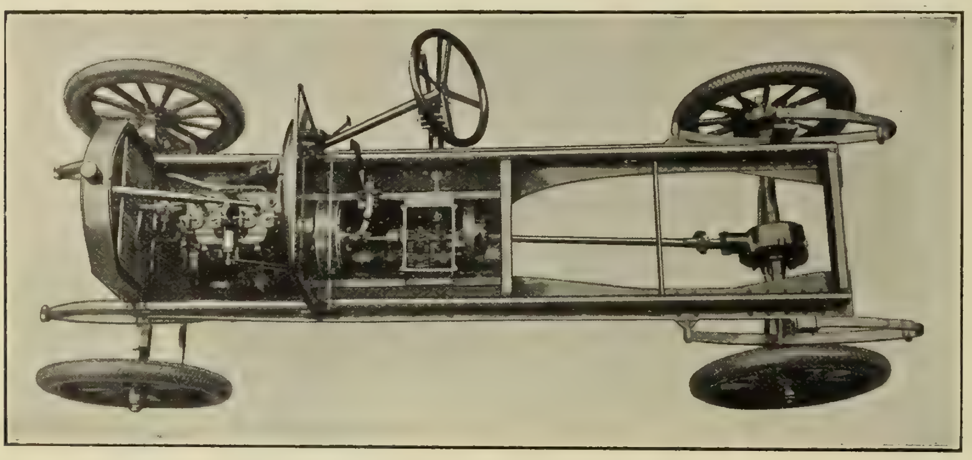


Fig. I. The 15 h.p. Darracq frame construction.

Commencing first with the frame, modern constructions tend to simplify and to strengthen this; simplification of construction has been carried on a considerable distance by the type of frame which is well-known through having been adopted on the 15 h.p. Darracq. This car demonstrates the results which can, and are, being obtained not only in France but also in America by simplification of the different units of its construction. As is shown in Fig. I. practically the whole frame is stamped, and no subsequent assembling or machining is required.

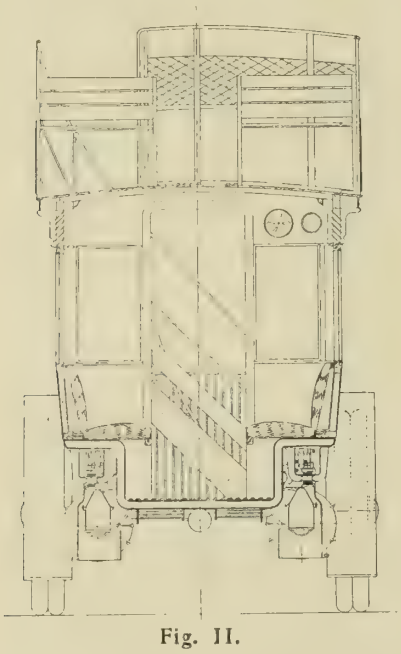


Fig. II.

The frame used on the Daimler omnibus, and shown in Fig. II., is also of a very interesting type, but being unique in its application it needs but passing mention.

Other frame improvements deal with the stiffening of the rear members of the frame, and it is well known that such improvement is necessary. One has but to lift up one corner of a pressed steel frame to find that it can be raised a considerable distance from the ground before the remainder of the frame rises, and this whip of the frame is very detrimental not only to the transmission system but also to the coach work.

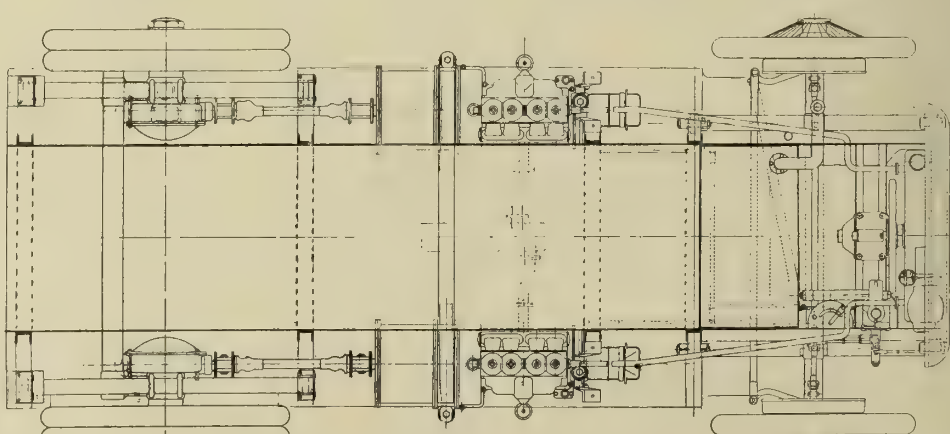


Fig. IIa.

In front axles little seems to have been done recently. The Kirkstall Forge type, in which the steering jaws are separate from the beam portions, is used perhaps more largely than used to be the case, and certain modifications have been necessitated by the employment of front wheel brakes. For the latter a number of different constructions of brake mechanism have been employed. In the majority of cases the actual brake-operating member moves in the axis about which the wheel swivels for steering. It is believed that these brakes are not altogether satisfactory, as that in spite of the employment of what is called compensating mechanism it is merely the effort transmitted to the brake mechanism which is compensated, and the result obtained from the two brakes is dependent amongst other things on the condition of the brake surfaces.

In rear axles attention is being given to worm gearing, and, also, it is regrettable to notice, to arching the axles, so as to enable splayed wheels to be used at the rear as well as the front. One or two constructions in which the differential gear is placed on the propeller shaft have recently appeared, but they do not seem to have attracted much interest.

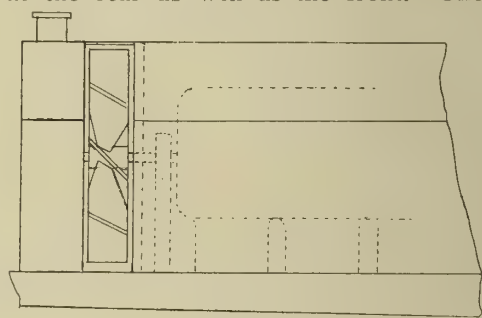


Fig. III.

Concerning road wheels, attention has chiefly been given to devising new means for complete easy attachment, or for rim dismounting, but there is no very notable invention in this category to be chronicled for the past year.

In radiators there seems to be a slight tendency, which it is to be hoped will be fully developed very shortly, to exclude the air passing through the radiator from the bonnet. By closing in the front of the bonnet the dashboard can be carried right down to the frame, cutting off the engine noises from the occupants of

the car. This is to a large extent what is obtained in the Renault type of bonnet and radiator, and a patent has been taken out by De Dion-Bouton whereby a similar effect is obtained with a radiator placed at the front. This is illustrated somewhat crudely in Fig. III., from which it will be seen that a space is left between the radiator and the bonnet and a centrifugal fan is located in the space.

Naturally the greatest improvements have been made in the engine, and in the direction of silence and efficiency, hitherto considered opposites. In the Silent Knight sleeve-valve engine one of the most important features is the high power obtainable without losing the virtue of silence, but some manufacturers now obtain very similar results from the ordinary poppet valve engine. It is now no difficult matter to design valve mechanism which is quiet over a fair range of speed and yet permits the engine to run at very high revolutions, developing high powers, and though at higher speeds it is true the engine is not very quiet, still other noises occur which probably prevent any individual noise being particularly audible. Very large valves are used with special cams and stiff springs, particular attention being paid to the drop of the cams to prevent noisy action.

It is impossible to draw attention to all the different types of silent valve systems which have been produced. There have been sleeves, single and double, which reciprocate or oscillate or combine two of these movements, the two sleeves being either in contact with the pistons or separated by a stationary guide member. Many inventors have turned their attention to disc valves, located flat against the cylinder head, these valves being either single or in pairs and adapted to rotate or oscillate; others have devised piston valves arranged at each side of the cylinder, piston valves in the combustion head, ring valves surrounding the combustion chamber, flat plate valves arranged at each side, and rotary overhead valves.

Where the new engines may claim advantage over an engine with ordinary poppet valves is in that they require practically no attention for the maintenance of maximum efficiency, and for this reason it appears that manufacturers of poppet valve engines will need to give special attention to the means for rendering their valves easy to remove and replace. At present many designers either assume that the valves will remain in place for ever, or that purchasers of their cars will supply themselves with those valve-spring compressing tools which suit their parti-

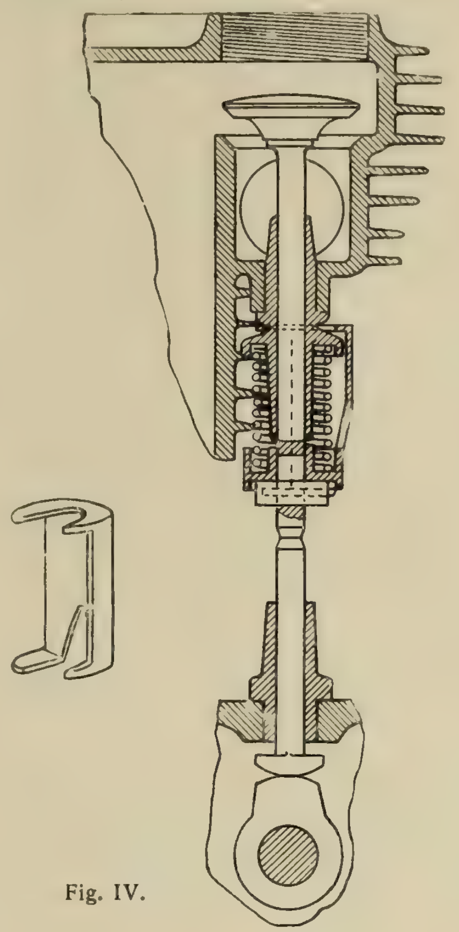


Fig. IV.

cular engine. Often for each valve spring a different tool is required. A valve remover should form part of every engine.

Illustrated in Fig. IV. is a simple little device which is supplied by the Triumph Cycle Co. with their motor-bicycles. It consists of a cage which is slipped beneath the washer, and into a groove in the valve guide, after the engine has been turned to compress the spring. This cage holds the spring compressed, so that when the engine is turned to allow the valve to drop, the cotter can be easily withdrawn. With this tool valves can be removed and replaced without the slightest difficulty, but it will be observed that for the replacement of a spring the tool cannot be used. In the matter of accessibility of the valves a very great deal can and should be done. It would seem that the present type of helical spring is not the best in this connection, and it is surprising that there have not been a number of patents issued lately for inventions dealing with the improved accessibility of the valves.

In engine lubrication controlled splash seems to be holding its own in spite of a tendency towards forced lubrication. The trough system, by no means a new idea, is very widely adopted, and some circulating system seems to be almost universal. In one or two cases the tank is separate from the crank chamber, which is an advantage in that the oil removed has a chance of becoming cooled down before being admitted again to the engine.

In water circulation systems thermo-syphon cooling is on the increase. With small cars, in which the engines are of comparatively small height, the radiator can be placed in front, but in larger powers it has to be placed at the back to get sufficient head. It should be borne in mind that with the radiator at the back, the steeper the hill the car is climbing the smaller is the head, but with the radiator in front the reverse is the case.

In carburettors the tendency seems to be towards abolishing automatically moving parts and the employment of carburettors of the "progressive" type, that is to say, in which the various port openings are opened up simultaneously by the movement of a single lever. This type is represented by the White and Poppe, Trier and Martin, Claudel-Hobson, the Polyrhöe, and other well-known makes, and, in competitions where efficiency is the main test, it is usually found that a carburettor of this type is used by the winning cars. Tendency is towards ensuring the complete filling of the cylinders at high engine speeds, one or two patents having been published recently in which the air or gas is compressed in the crank chamber so as to be forced into the cylinder.

The almost universal adoption of magneto ignition has turned attention to improved methods of starting, and a number of mechanisms have been recently patented for enabling a starting spark to

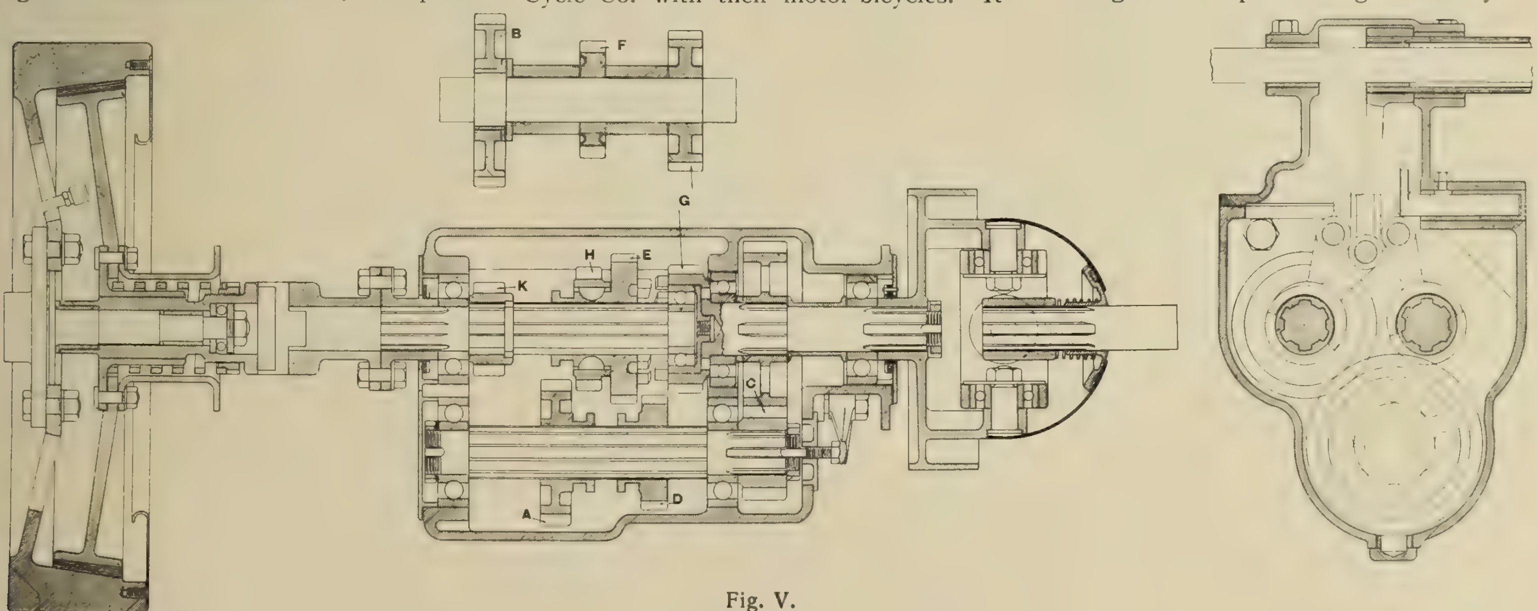


Fig. V.

be obtained from the magneto with the engine stationary. In other devices the magneto distributor is used for the accumulator ignition. Attention is also directed towards filling the cylinders with gas for starting. There seems to be a tendency towards eliminating the advance and retard mechanism or of rendering it automatic. In this connection a patent was recently published in the name of Bosch for a simple automatic device, whilst some Eisemann magnetos have been so fitted for some little time.

In clutches, but slight modification has been made, though in one or two constructions, which have been the subject of recent patents, a special form of spring device is used to cause gradual engagement.

There is little to record in connection with speed gears, but one of the most notable construction is that of the Maudslay, which is illustrated in Fig. V.

This gearbox is a particularly interesting design, as it provides four forward speeds and still has extremely short shafts The highest speed is geared up, the third being the straight-through drive, and in order to understand the operation it must first be realised that the striking arm engages more than one of the strikers at the same time, except for the reverse movement when it engages with one only. Thus, movement of the first and second speed layshaft pinions is simultaneous, and on coming through the gate the first speed pinion is released, leaving the second speed striker still engaged. The second speed pinion then moves together with the third speed clutch, this being necessary in order to move it backwards when engaging the third speed. In Fig. V. the second layshaft which lies below the other two is shown separately in order that the operation may be followed more readily. Starting with the reverse gear it will be seen that the drive passes from K to B, from B to A, and then through the pair C to the propeller shaft. The next motion of the control lever disengages the reverse and picks up the striking lever controlling pinion D. Forward motion then gives the first speed, and backward motion the second speed, using the final drive C as before. Passage across the gate does not release A, which moves idly, and backward motion will move the dog clutch into engagement, pushing pinion D still further backwards where it is out of the way. Next, forward motion, without coming back through the gate, brings wheels E and F into mesh and the drive is then transmitted through the pair G, giving the geared up fourth. The disadvantage of this form of construction is that the two layshafts are always running, and the reverse wheel B is also always running at a different speed to the layshaft which carries it.

In connection with gears generally, a finishing system recently produced by Mr. Poppe and illustrated in Fig. VI. is particularly interesting, because amongst other virtues, the cut is taken in the direction of movement of the engaging faces.

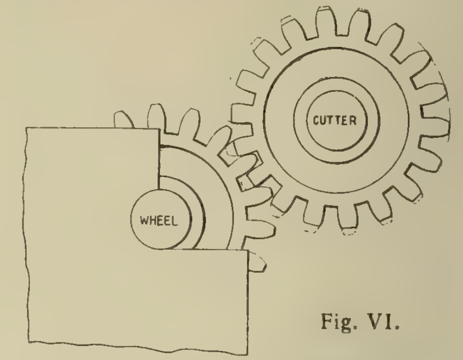


Fig. VI.

This system consists broadly of a cutter with mutilated teeth meshing with a partly finished wheel, the cutter being mounted slightly eccentrically and having an odd number of teeth to provide a hunting tooth effect when, if left with tool marks before hardening or fed with an abrasive, it is said to eliminate any errors of spacing, etc. I am not aware that this system has been applied to change speed gearing, but I know it is a success in connection with timing gears.

DETAIL DEVELOPMENTS.

An illustrated description of some small improvements seen at Olympia.

NOW that the motor car, or rather that particular type of motor car which has become standard automobile practice, can be claimed to be efficient and reliable, more attention is naturally being paid to the comfort of the passenger and driver, and to convenience, or rather lack of inconvenience, in attending to the various details which control the running of the machine. It has been said, and with a good deal of truth, that this year's Olympia Show proved that attention is now being paid to small details more than ever before, the general lines of the car having proved themselves sufficiently tenable to warrant them being left in practically the same condition as they found themselves in last year and the year before. And this detail improvement is a very natural thing, for already the large majority of manufacturers have associated themselves with a design which they see no good reason for changing in any radically elemental manner; hence they have restricted their outlets of originality and development to quite small and almost trivial details. It might be thought that such development was simply undertaken for development's sake, in order that it might be claimed that next year's models are an improvement upon those of 1910, but it must not be forgotten that it is only by great care for such details that the running of the present-day car can be improved greatly, or the comfort of its passengers enhanced.

The purchaser's point of view in this connection is one, however, which ought to be considered by every shrewd manufacturer. For instance, he should endeavour, as far as possible, so to design his detail improvements that with a minimum of trouble they can be fitted to his models of the previous year, which will thus be brought entirely up-to-date. He may well say that if he does so owners of his previous cars will be offered no inducement to purchase his latest productions, but we believe this view to be a mistaken one, for it is probable that the very reverse is the case. Users of his former cars will have all the more incentive to enroll their names in the next year's order book, if they know that by being able to bring their present cars thoroughly up-to-date they will thereby be enabled to obtain a better price for them in the second-hand market.

In the following article we propose to deal with a few detail improvements disclosed at Olympia. It would, of course,

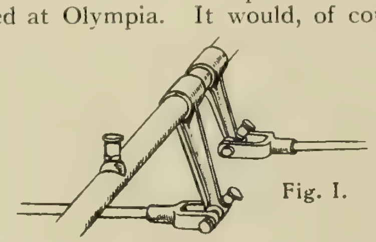


Fig. I.

be quite impossible to deal with them all; hence we have selected a few representative ones. It is quite probable, in fact it is more than probable, that many of the fittings we shall discuss are to be found on many cars other than those from which we obtained them, but the ones which we deal with have been chosen primarily with regard to clearness of description and illustration, and more to represent the trend of design than for any other purpose.

One of the most noticeable features in the Show was that designers are at last beginning to realise that to eliminate any possibility of squeak, adequate means of lubrication must be furnished for all working parts, especially those which come into use very seldom, and also those parts which, from their very situation are apt to be overlooked. We refer more particularly to articulations in the brake mechanism, spring shackles, pedals, etc. These have frequently escaped the attention of the designer, because he has looked upon them as parts which sustain little or no wear, since they are so seldom brought into use, but many have forgotten that it is because of these very facts that dirt is given an opportunity of accumulating, hence squeaks and other noises can very frequently be traced to under-lubrication of small parts. Brake joints, from their situation under the body of the car, are especially prone to become encased in mud, and we noticed a decided improvement in this respect on the Singer cars, in which every articulation in the brake rods had its own lubricator (see Fig. I.). We fancy that with a filling up of a good thick oil every month or two, any possibility of harshness or noisiness in working could be avoided by this simple means, which, if only on the score of its very small expense, and apart from its undoubted efficacy, is worthy of installation in any chassis which pretends to be thoroughly up-to-date. Steering pivots and joints are by now generally thoroughly well lubricated, but several firms, we are sorry to see, still persist in allowing their spring shackles to run perfectly dry, since no adequate provision is made for their lubrication. In the Mors chassis there is a very neat and effective means of lubrication furnished to all the spring shackles. The shackle pins are hollow, and internally screwed through their whole length, and the walls are also drilled. The cap,

which closes them at their outer end, is furnished with the small screw plunger, which is prevented from rotating, but is free to move in and out in relation to this cap. The hollow shackle pins are filled with grease, the cap is inserted, and the small piston screwed in, this having the

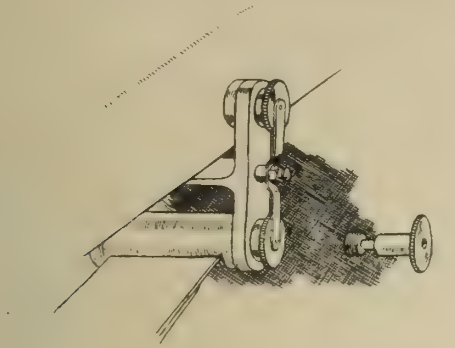


Fig. II.

effect of forcing the grease into every part of the shackle. The caps of each shackle are prevented from falling out with a spring yoke, as shown in the sketch, Fig. II. and in consequence all that it is necessary to do is from time to time to give the milled cap a turn or so until the small piston reaches the limit of its stroke. We believe that this method is considerably to be preferred to the ordinary arrangement of lubricator, as in order to get the oil well into the shackle, a thin lubricant has to be used, which very easily works itself out, whereas a thick grease can be made to do service for a very considerably longer time, and, as in the present arrangement, only requires replenishment at fairly long intervals.

The question of brake adjustment has also very properly received attention, and, as an instance, the manner in which adjustment is allowed for in the brakes on the Rolls-Royce car is, as might be expected, both ingenious and thorough. This is shown in Fig. III., and consists in the employment of a locking joint, somewhat similar to that used on the Auster wind shields, but in this case applied to the operating arm on the camshaft, which expands the brake shoes inside the drum on the back wheels. The teeth of the locking joint are sufficiently small to allow an adjustment to be made which does not seriously effect the "feel"

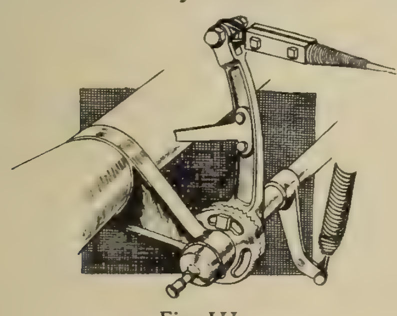
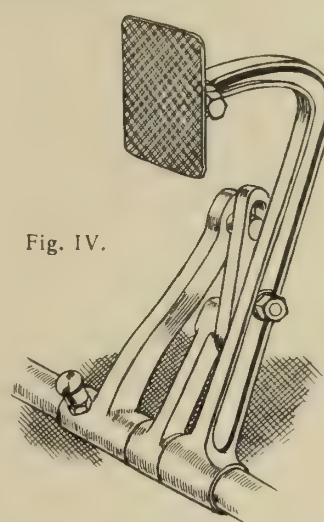


Fig. III.

of the brake pedal, and, at the same time, is one that cannot work loose or come adrift in case of emergency. The sketch also shows the simple way in which a brake stop is provided, the brake arm carrying the small bracket which, when the brake is completely off, is held by the spring against the shell of the back axle, all noise from looseness in this part of the mechanism being thereby eliminated in any and every position of the brake adjustment.

Most of the cars at the Show were fitted with the brake mechanism controlled by rods, but a number still adhere to the stranded steel cable. Whilst admitting that we have seldom heard (at all events within the past few years) of accidents caused by failure of these brake cables, we cannot help thinking that of the two methods it is the less mechanical, and the one which is more likely to feel the effects of neglect, that is to say, those parts of the cable which are subjected to sharp bending strain are liable to weaken and chafe. Furthermore, there are points at which the water is liable to accumulate, especially in the hollow cross shaft through which the cable passes. On the other hand its mode of employment certainly gives a very simple and effective form of compensation. Whilst on the subject of brake compensation it is fitting to deal with the arrangement provided for this purpose on the 12-16 h.p. Argyll chassis, which is exceptionally neat, particularly as it takes up practically no space at all, and does not involve the use of a cross

Fig. IV.



shaft other than the one on which brake and clutch pedals are pivoted. It will be seen from Fig. IV. that it consists of what may be called a small lever differential, the difference in length of the two operating levers being balanced by the position of the fulcrum of the toggle lever which joins them. The two brake levers are fixed to the halves of the long sleeve which encloses the cross shaft, the brake pedal being pivotted loosely on the sleeve attached to the small lever. It will be noticed also that the pedal itself is connected to the pedal lever with a knuckle joint, so that the former can be adjusted to the most convenient position for the driver. On the Rover cars a very ingenious method of housing the ratchet of the hand brake lever has been adopted. This is illustrated in Fig. V., from which it will be seen that the rather untidy ratchet quadrant has been, as far as outward appearances go, entirely dispensed with. The ratchet in this case is contained inside the circular drum, the usual trip pawl being employed, and, in this case, carried inside the hand lever. From every point of view this idea is a decided improvement.

With regard to the operation of the foot brake on the countershaft, many and varied have been the means of obtaining the necessary cross motion between the pedal and the brake shoes. In many of these this action has involved the use of a rather large number of parts, or else sufficient care has not been taken to ensure that only a direct radial and not a side thrust is thrown on any part of the shoes. The mode of actuating the foot brake on the Delahaye chassis is notable for its extreme simplicity, and a sketch of it is given in Fig. VI. The brake pedal lever

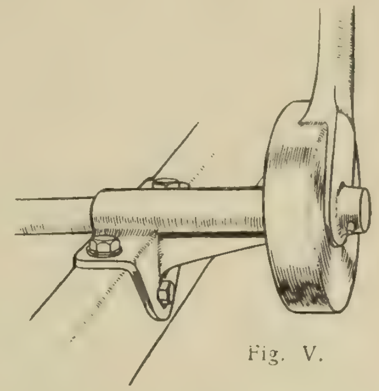


Fig. V.

and the crank which operates the shoes are furnished each with a few teeth representing one quadrant of a bevel pinion. As shown in Fig. VI., a stop is provided to the brake pedal, and also a means of adjusting the brake actuation by hand, while it will be seen that should any wear occur in the bevel gear it can very readily be taken up by the insertion of a thin washer on either of the two pins, thus forcing the teeth further into mesh.

On the whole no very striking development has taken place in brakes themselves, except that the idea of relieving the propeller shaft and differential of all braking stresses by mounting both the foot operated and the hand brake control on the rear wheels is gaining favour. In some cases the brakes are mounted side by side, but the more general plan is to use two concentric brake drums and enclose them in a single casing, while the third method is to use one drum only and apply one brake internally and the other externally. It must, however, be admitted that hub brakes, whether front or rear, certainly require more effort on the part of the driver to pull the car up in a given space than is necessary with a brake on the countershaft, and further, it also seems as though the latter type could be applied with rather less harshness than the former. There is one reason for the

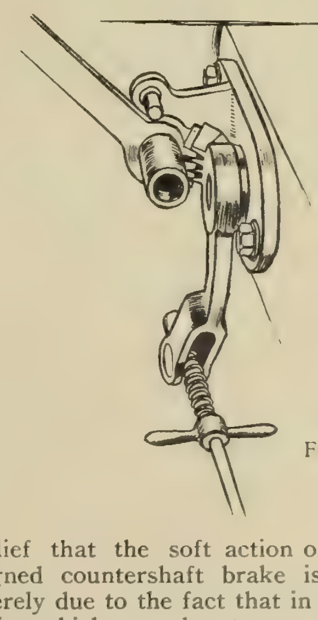


Fig. VI.

belief that the soft action of a well-designed countershaft brake is, however, merely due to the fact that in spite of the pains which are taken to prevent oil gaining access to the surfaces, some slight amount of lubricant does actually find its way there, and because the brake is geared up the presence of this lubricant is not

access to the surfaces, some slight amount of lubricant does actually find its way there, and because the brake is geared up the presence of this lubricant is not actually a disadvantage. It will be understood that in making the above statement we are speaking from our own immediate experience, but if the present type of rear wheel brake does actually suffer from the slight disadvantage we have suggested, in any case it is one which should speedily yield to treatment, apart from the fact that it is very considerably outweighed by the practical advantages of the system.

Turning now to the question of control, we find the most notable feature to be the considerable simplification which has been made in this connection, so that not only clear dashboards, but also clear steering wheels are the rule rather than the exception; indeed, we think that in some cases simplification has been obtained at the expense of efficiency, though when one considers the ill-tutored hands into which its very reliability and simplicity direct the modern car, something must be said for this point. There is not the slightest doubt that better results can be obtained with any given engine in which the ignition point is alterable than when the magneto ignition is fixed. Many users, however, do undoubtedly prefer to have their hand control reduced merely to the throttle lever, and we must confess that in the ordinary running of a car, fixed ignition in which the timing has been well selected does not appear to have any noticeable disadvantages when compared to alterable ignition, except when it comes to a burst of speed, a crawl behind traffic on the high gear, or a timed hill climb. As far as the throttle is concerned, the best makers adhere to the policy of so fitting the accelerator pedal that the amount of gas it allows to pass to the engine when in its fully-off position can be adjusted so as to permit the engine to turn round quite slowly. Examples of this idea are shown in Figs. VII. and VIII., the former illustrating the method adopted on the 15 h.p. Napier car, where the accelerator pedal is linked up to a lever on a small cross shaft, which, under the action of the spring on the pedal, is brought up against a stop, the end of which can be lowered by a small ratchet hand lever on a dashboard, so that the minimum opening to the throttle can be set with extreme nicety.

The method used on the 15 h.p. Germain is illustrated in Fig. VIII., and is rather more simple in arrangement. A small plate is screwed on to the dashboard immediately above the accelerator pedal pivot, and against this plate there is a small pointed screw, provided with a lock nut, so that the position in which the pedal comes up against the stop can be regulated and set. On this car the throttle control is notable for the fact that it gives a perfectly even distribution of opening throughout its entire length, that is to say, the throttle opening is at all times directly proportional to the length of the arc through which the accelerator pedal is moving. In the ordinary case the pedal operates the throttle through plain levers, so that generally towards the end of the stroke, when the throttle is being opened wide, a very large motion of the pedal is necessary to obtain quite a small additional throttle opening. This fault is inherent in the common type of control illustrated in Fig. VII., for it will be seen that as the pedal is pressed down the turning moment of the slide pin about the cross shaft increases, hence, towards the end of the stroke much larger movements from the pedal are required to get a proportional increase of throttle opening than at the beginning. In the Germain arrangement, however, the bent crank which bears against the roller lever operated by the accelerator pedal is so shaped that this differential throttle opening effort is

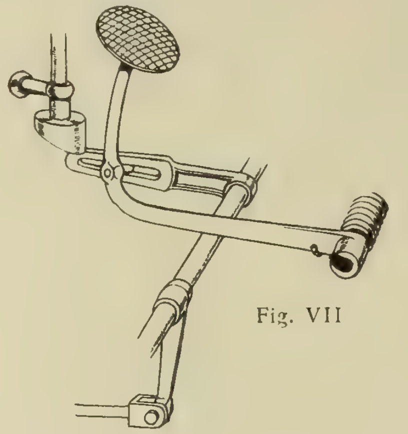


Fig. VII

avoided without the introduction of any additional working parts; indeed, the whole arrangement of the control in the Germain is simplicity itself.

Thanks to the excellence of the modern carburetter—i.e., compared to the instruments of a year or two ago—the extra air lever, which at one time was a common enough fitting on the steering wheel, has now practically disappeared—at all events when it is fitted it is to be found generally on the dashboard rather than on the steering column.

A distinctly neat and novel method of limiting the minimum opening of the throttle is fitted on the 20 h.p. Mors chassis. As shown in Fig. IX. it consists of a milled head attached to the outer sleeve of the column, and by rotating a stop, which the accelerator pedal lever rests against in the "off" position, can be altered in position. The beauty of this device lies in the fact that it takes up no appreciable room, and has no projecting points to catch either in the driver's glove or coat sleeves. In our opinion every effort should be made to introduce small controls in this manner, so that untidy projecting details can be avoided.

Another contrivance which certainly makes for the drivers' and passengers' convenience, and is therefore well worthy of attention, is the folding steering wheel which is fitted to the Lanchester cars, and allows the driver easily to enter the car on the gear lever side. The wheel is held in its normal position by a latch, so fitted

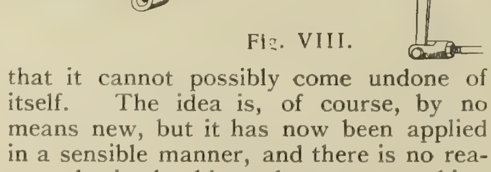


Fig. VIII.

that it cannot possibly come undone of itself. The idea is, of course, by no means new, but it has now been applied in a sensible manner, and there is no reason why it should not become something approximating to standard practice.

At last manufacturers are beginning to realise that all men are not the same stature, and for the first time adjustable brake and clutch pedals are now becoming comparatively common. The majority of designers are content merely to provide the pedal with a knuckle joint, but this is far from being sufficient. What is wanted is an arrangement which will give an adjustment over at least six inches—the normal position being midway between the limits, and not, as is generally the case, one of the limits themselves. If possible, too, the arrangement fitted should allow the necessary adjustment to be made with as little trouble as possible, preferably without lifting up the footboards.

There can scarcely be any doubt that magneto ignition has, during the past year, undergone more progressive development than perhaps any other portion of the car's economy. The magneto has, in fact, been brought to a wonderfully high state of efficiency and reliability, and remarkable steps have been made towards automaticity in working and general convenience. First of all, there is the Eisemann automatic-advance machine, in which a centrifugal governor, mounted on the driving shaft, and enclosed in a casing forming a part of the frame of the magneto, acts upon a quick-pitch helical keyway, and advances or retards the armature, and with it the contact breaker, according to the speed of the engine. A similar device has for a long

Fig. IX.

time been fitted to the Germain cars, and was also at one time used on the F.I.A.T., but its incorporation in the magneto itself is undoubtedly a step in the right direction, for the automatic-advance magneto, besides having the advantage that no ignition control lever is required, has also the great additional one that the "break" occurs always when the magnetic field is at its maximum pressure, and hence the spark produced is at all times of the greatest possible strength.

Then there is the two-spark machine, in which sparks are delivered synchronously at two ordinary single pole plugs. There are two systems used for this purpose. In the Unterburg and Halmite machine two entirely independent armatures, employing a double contact breaker, are mounted in tandem on the same shaft, and serve a double distribution, so that the effect is the same as two distinct and separate magnetos. In the Bosch system a single armature is used, but has a double secondary winding. Only a single low tension contract breaker is used, hence the synchronisation of the two sparks is quite perfect. This system, however, has the disadvantage that when starting up, the current generated in the high tension winding is not of sufficient strength adequately to pass across two gaps, hence one set of plugs has to be cut out with a switch. It will be seen that in this case the effect obtained is analogous to that got with the use of a double pole and a

single pole plug in series, and an ordinary single spark magneto.

The dual-ignition magneto has also been considerably improved during the past season. The principal difficulty lies in the differential advance and retard mechanism, which is required by the fact that the lag in the coil system is constant, whereas that in the magneto system becomes less and less as the speed is increased to a certain limited extent, therefore the magneto automatically times itself, and consequently only requires the contact breaker to be moved through a small arc to give full advance and full retard effect. On the other hand, the coil contact breaker requires to be moved through a very much larger arc, and as both movements must be governed simultaneously by one lever, a special mechan-

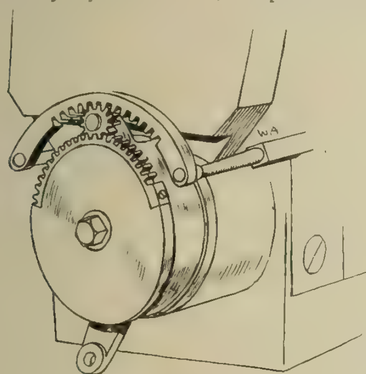


Fig. X.

ism is required. In the Hall E.O.A. magneto the coil contact breaker and the distributor are moved through a helical slide by a lever, the other end of which operates the magneto contact breaker in the usual way. The C.A.V. dual ignition machine has the ingenious arrangement illustrated in Figs. X and XI., the former being a sketch of the complete mechanism, and the latter a sketch of the inner side of the contact breaker cover. The magneto contact breaker is mounted in the usual way, and is advanced or retarded by a lever on the side of its case which carries the actuating cams. The coil contact breaker consists of a carbon brush, borne by the magneto contact breaker body, which rubs against a modification of the ordinary high tension distributor, which forms the lid of the cover. This lid carries on its edge a small rack, in which runs a small pinion mounted on the walls of the contact breaker case, this pinion also meshing with a second rack fixed to the body of the machine. The coil contact breaker is by this means always advanced twice as far as that of the magneto. In order to avoid the use of

Fig. XI.

very large collecting plates in the distributor to accommodate the large degree of coil contact breaker advance, the body of the distributor is connected to the contact breaker casing, so that when the latter is moved the former is also rotated in the opposite direction.

Amongst the neatest devices shown at Olympia was the Simms magneto starter. This consists of a small low tension coil and a trembler, carried on the dashboard by means of which an intermittent current can be sent through the low tension winding of the armature, this inducing a high tension current, and producing a spark at the plug. In order to prevent the battery current running away to earth when the contact breaker points are closed, a distributor is arranged on the lid of the contact breaker cover, by which current is prevented from passing unless the points are open. It has been found that four and six cylinder engines come to rest ninety-nine times out of a hundred with the points open, hence multi-cylinder engines can be started on the switch with a considerable degree of certainty. A six or eight volt. battery is used.

The new Brooks magneto is both ingenious and effective. Its principal feature is the employment of movable pole pieces, which, mounted on a sleeve connected to the contact breaker cover, ensure that the circuit is broken when the field is at its maximum, the action being similar to that obtained by advancing the armature with the contact breaker. On the same machine the double-point contact breaker is arranged in a most ingenious manner, being mounted on a sliding block, which can be withdrawn for inspection without disturbing any other part of the machine. In view of the fact that when in position on the engine the magneto is rarely easy to get at, and still less so to inspect, this contrivance must certainly be regarded as a decided improvement.

It seems probable that in the early future that this magneto will find itself displaced, or at any rate threatened, by the lighting dynamo which, now that it has been reduced to such a practical state, is opposed by no insuperable objections from being adapted to serve also for ignition purposes. A great number of these dynamos were exhibited at Olympia, and whilst some were undoubtedly good, others had been designed with two optimistic a view of the conditions under which these small generators have to work. They may be divided into two genera, those which employ permanent magnets, and those which use electromagnets, and these in their turn can be sub-divided into four species, viz., (i) those which cut in and cut out electrically, (ii) those which cut in and cut out mechanically, (iii) those which combine the above ideas, and (iv) those which are entirely automatic. In the permanent magnet type the general rule is to allow the machine to be driven at any speed above a certain minimum, as the field is constant, and the output consequently does not at any time rise very high. In the excited magnet type the rise is, of course, considerably higher, hence a number of machines are provided with a slipping driving pulley, in which at a predetermined speed centrifugal action allows the armature shaft to slip and lag behind the driving shaft. In the C.A.V. system there are neither electric nor mechanical cutouts. Owing to the special arrangement of short-circuited coils on the armature, and the provision of subsidiary field magnets, as the speed increases above the normal, a cross-magnetisation effect is obtained, which weakens the main field and reduces the pressure of the output, which in consequence never appreciably rises above its normal. Below the speed at which the dynamo generates current at a higher voltage than that of the battery circuit, the dynamo is allowed to be driven as a motor by the battery, but as a free wheel is mounted upon the armature shaft, and there is consequently no tendency for the motor to drive the engine, the amount of current used is practically negligible.

An improvement for which racing ex-

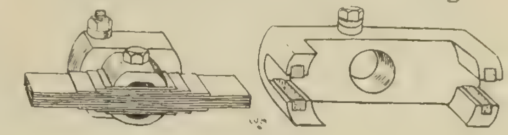


Fig. XII.

perience is more than anything directly responsible, is an elastic drive for the magneto, which is used this year on a number of cars. In the Rover models the magneto coupling consists of a leather disc, whilst in the Vauxhall a hard rubber buffer is used. The Bosch coupling illustrated in Fig. XII. is also worthy of note, a small stiff laminated spring being used.

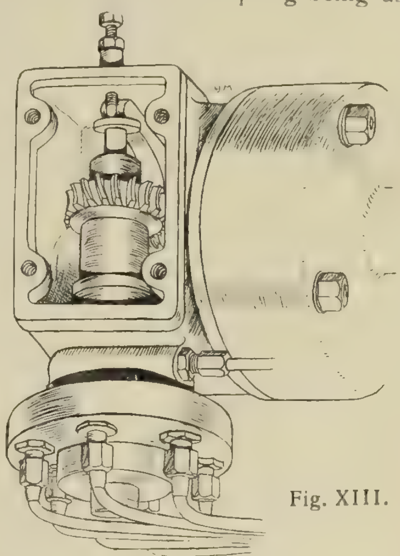


Fig. XIII.

The lubricating pump on the Valveless chassis, illustrated in Fig. XIII., is an excellent example of careful attention to detail, as well as an instance of originality. A small upright plunger pump is operated by a cam, actuated by a 10 to 1 gear from the engine. The stroke of the plunger is controlled from outside the pump casing by a set-screw, which decides the point to which it can fall on the return stroke. At the top of the pump is a six-way oil distributor, leading to different parts of the engine and chassis, the body of the pump being for this purpose slowly rotated by a worm reduction gear. The whole affair is of quite small proportions, and, we believe, is most effective in working.

On the same car a very ingenious trip pedal is used to operate the compression release valve. This is shown in Fig. XIV. The pedal proper is loosely jointed to the valve-stem, and is held against the action of the valve spring by a small knife-edge,

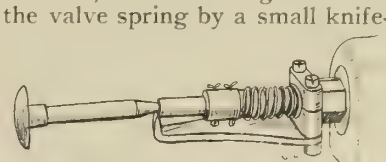


Fig. XIV.

which fits into the groove turned in its stem. To allow the valve to return to its seating a slight touch of the foot is all that is necessary to release the pedal, whilst to open the valve the foot presses the pedal in until it engages once more with the ledge.

MODERN DEVELOPMENT OF BRITISH UTILITY MOTOR VEHICLES.

A Review of Progress since the introduction of the Heavy Motor Car Order.

ALTHOUGH much valuable work has been done in the more distant past in connection with self-propelled vehicles for business purposes, and though automobiles have been used for business ever since the motor car was practically legalised on British highways in November, 1896, the real development of these specialised branches dates from the day on which the Heavy Motor Car Order was brought into force—March 31st, 1905. Before that date the weight limits imposed by law had not been such as to render road motor transport, of the heavier order, profitable, and the sporadic attempts to work under the then existing regulations had not in general proved successful, while the heavy demand for private cars had left the lighter branches of industrial motor work practically untouched.

The advent of the Heavy Motor Car Order, however, very nearly coincided with the time when economic conditions turned the attention of manufacturers towards the possibilities of the lighter types of industrial motor vehicles, so that the development of the both heavy and light utility motors may, for all practical considerations, be regarded as contemporaneous. Since 1905 that development has been so steady and continuous that it is not possible to take any fraction of the elapsed period by itself, and therefore, in taking a broad view of commercial automobile development, it is necessary to consider its history of the past five years.

That history has been one of development for many varieties of use, and so, not only do we have the two broad divisions of steam and internal combustion vehicles, but we have to consider self-propelled vehicles as mutually divergent as the motor cab and the steam tractor.

Taking 1905 as our commencement we find the designs of the few existing utility vehicles naturally influenced by prior and existing types, and thus in the several departments the traction engine, and the private owner's steam and petrol car have all left their mark in the industrial motor world of to-day. Broadly speaking, the best results have been obtained from those machines whose general design most resembled vehicles in previous use. Thus the traction engine type has survived with relatively little alteration, while the pleasure car type, with suitable modifications, has also come into very great use. On the other hand, the majority of the really original designs have failed entirely, though a few have survived to a limited extent. This is entirely in accordance with the usual experience in engineering, where it is a curious paradox that those who try to be the most original generally fail to produce any really original results. The reason for this is that those who start with a well-tried machine and modify it to suit the new conditions start with all the possible knowledge on the subject, and all their modifications are improvements, while those who start with the intention of making an entirely original machine, with no reference to previous experience, usually spend their whole time making the mistakes which have already been met in a previous generation.

Steam—Tractors.

Taking the steam vehicle as being the oldest, the influence of traction engine practice has on the whole come out predominant, although in many steam lorry designs such car features as Ackerman steering and final transmission by side chains are retained.

The steam tractor as we know it today is, in fact, nothing more than a miniature traction engine built to come within the 5 ton weight limit of the Heavy Motor Car Order. Since its introduction, however, it has shown distinct tendencies of its own in general, and several interesting features have been introduced in particular designs.

Tractors have been built to work with high-pressure steam in one or two cylinders, and also compound. The double cylinder non-compound never attained to favour, and the single cylinder design, in which extreme simplicity was the keynote, has not been so widely used of late. The compound engine is undoubtedly the type in general favour, for reasons of economy and to save having to carry such large tanks as might otherwise be necessary, and the successful employment of feed water heaters has also materially contributed towards steaming efficiency. In the main details ordinary small steam engine practice is followed, but one of the most radical departures from ordinary traction engine design was brought out by a leading firm some four years ago in the shape of a compound tractor with three cylinders, the high pressure being arranged tandem with one of the low pressure cylinders, and in this design the makers were not the only firm to borrow from automobile usage so far as to enclose the motion in an oil-tight casing, which enabled splash lubrication to be used.

Like traction engines, too, tractors are at present gear driven, and, generally speaking—whether three or four shaft tractors—have the balance gear contained within the massive back axle. Consequently the drive is taken by two spur pinions on the first countershaft to large gears on this live axle. One firm, however, of long experience places the balance gear on the first countershaft, and through two pinions—one on each end of the shaft—transmits the drive to gears secured to each of the driving wheels which revolve on a solid axle. This form of construction has left the balance gear more accessible, and has certainly resulted in a very fine machine.

The tractor, which has the advantage of being detachable from its load so that it can continue to work while loading is in progress, also has the advantage of large back wheels, but these large wheels, as a factor for smooth running, are somewhat discounted by the very short springs that manufacturers of tractors have borrowed from traction engine design. Under the circumstances these short springs were inevitable, for along with them the traction engine form of transmission was also borrowed. In this transmission the play between the gears on the spring-supported gear shaft and road-supported back wheels of a machine, whether it be of the three or four-shaft variety, is taken up merely by lengthened teeth, so spring action must necessarily be limited. The higher speeds allowed to tractors under the Heavy Motor Car Order, however, have led to a growing feeling among some manufacturers that things might be somewhat improved in these respects, and consequently in some designs attempts have been made to improve the springing, or to provide a gear drive in which the spring action of the vehicle shall be taken up in some more scientific manner than by merely allowing the teeth to slide more or less deeply into mesh according to the play of the springs with road inequalities.

This last is effected by mounting the big gear on a sort of enlarged Oldham coupling instead of securing it directly to the axle. Though the old system is crude, it has, all the same, proved reasonably effective, but it is possible that the tractor of the future may have more refinements embodied in its design, and be a more comfortably running vehicle than it is at present, although production costs must always prove a difficult obstacle, for the present prices of tractors are far from high, especially when the work involved is considered.

Some considerable stir was made when the Pedrail first made its appearance. This invention, which was confined to the back wheels of the tractor, consisted of a series of pads, very like elephants' feet, placed at regular intervals around the circumference of the wheel, from which they projected radially. These pads were actuated by a mechanism, somewhat similar in principle to that used for feathering the floats of a paddle wheel, so that, as they came near the ground, the soles of the feet, so to speak, were placed parallel with the ground. As the feet were each provided with a controlled universal movement they accommodated themselves to inequalities of the ground in a remarkable manner. The invention attracted much attention, especially from the military authorities of this and other countries, for the purpose of its design was thoroughly in accord with military requirements, but although tractors fitted with this device are at present in work, the large number of parts required, and the severe stresses and wear on the joints are disadvantages.

Another invention, quite as ingenious its way, and brought out with the same purpose—to travel over rough country—was the chain tractor, but as this more recent device will probably be fresher in the minds of our readers, we will not enlarge on it here.

Steam Waggons and Lorries.

In the main, steam waggons are all of the heavier class, and their design has followed along one of two lines: one school has taken the general arrangement of the tractor, and modified it to their requirements, while the other has adapted the design of the motor car to its peculiar wants, although of course in such a rough classification it is impossible to draw a hard and fast line, tractor and car features overlapping in many cases.

The former class has, if possible, the more nearly approximated to the tractor owing to the fact that many well-known tractor making concerns have taken up the manufacture of the steam waggon, and thus we have a vehicle the front part of which is virtually a tractor. Of course this involves a locomotive type of boiler carrying the engine on top, but at the smoke-box end the channel frames of the waggon are secured in such a way as to allow for expansion of the boiler. These frames are carried at the back by the driving wheels, the altered position of which, as compared with those on the tractor, necessitating a modification in the transmission, as a gear drive is obviously impossible. Consequently the drive from the engine is taken by a two-speed countershaft, whence it is passed to the balance gear live axle by a long chain drive. It is perhaps to many a matter of surprise that an open chain should run well on work of this kind, but in practice it appears to be thoroughly successful.

To comply with other requirements of this design, the hornplates, footplate, and tender also have to be modified, and the latter is in fact reduced to a bunker.

Although the locomotive type of boiler is generally used with an engine placed above it, some makers employ it in conjunction with an engine placed under the frame, between the front and back wheels, which is the usual alternative. In one of these designs special provision is made for retaining a comparatively large body of water above the roof of the firebox, so as to ensure that it shall be covered with water even when descending the steepest hills, as the ordinary locomotive type of boiler is somewhat at a disadvantage in this direction, although the difficulty can usually be got over by the driver by a careful management of steam and water supply. Another interesting development is the double locomotive type placed across the frame, for it has proved most successful.

In boilers the locomotive type, sometimes slightly modified, appears to be coming into increasing favour. This tendency may be in part due to the difficulties that have been caused in many of the vertical fire tube designs by contraction and expansion, but at least one well-known firm has got over this difficulty by employing a sort of compound fire tube of steel and copper, the one material being placed concentrically around the other.

If anything, judging by their increasing numbers, the balance in advantage seems to lie in having the engine on top of the boiler, for besides being easily accessible, and enjoying an absence of long steam pipes with their consequent condensation losses, the engine motion is well away from the dirt of the road, while another, though small, point is that the general run of enginemen are more accustomed to the handling of this type.

Generally speaking, however, the vertical boiler is used with those cars that have their engines beneath the frame, and these generators are usually of the fire tube type, although one old-established firm entered a vehicle with a water tube boiler in the Commercial Vehicle Trials of 1907, and obtained excellent results. Nevertheless this type does not appear to have been developed since then.

The principle of superheating the steam by the gases in the smoke-box is, however, far from unknown in heavy motor practice, for the practice of placing superheating coils in the smoke-box has been common for a considerable time past, especially in boilers of the vertical fire tube type. In view of the way in which superheating bids fair to come into favour for locomotive work, however, it is possible that we may see its further development in the locomotive type road waggon and tractor, but cost of production and weight constitute important factors in the problem, while in some machines which employ a pivoted baffle plate in the smoke-box (to increase the draught by shutting off or opening up the lower rows of flue tubes as required), the question of space is also an important item for consideration. In drawing any analogy, too, from locomotive practice it must be remembered that the draught of the exhaust on the fire leads to higher smoke-box temperatures in the locomotive than is the case on the road vehicle.

For the ordinary non-locomotive type of waggon with engine below the frame, side chain or spur gearing is employed, and as a rule the engine motion and transmission is more completely encased. A comparatively high speed engine is used as a rule, but, if we except a somewhat freakish design with a very large single cylinder engine turning the back axle by means of a species of ratchet gear, the most notable departure that has appeared is a design brought out by a Scotch engineering firm in 1906. In this a large slow speed engine is used driving by chain from the crank shaft direct to the back axle, and as a fairly high pressure of steam is employed, cam actuated poppet valves are fitted. This machine has several other notable points, among which is the steering, for the axis of Ackerman steering pivots are in the centre lines of the wheels, the somewhat flat cast steel spokes of which are bellied outwards accordingly. In heavy vehicles which require more effort to steer them than light machines, this is a feature that might well demand more attention than it obtains. At present, the writer believes that only one other important firm in this country employs this principle, which is to be found alike in their petrol and steam vehicles, and in this case the steering pivot is contained within the hub.

Transmission arrangements usually provide for two, rarely for three, speeds on the ordinary sliding gear principle, and frequently safety devices are fitted to prevent more than one gear being brought into mesh simultaneously, but though they serve their purposes well, they are necessarily crude compared with ordinary automobile change speed mechanism, and do not here require elaboration.

Besides the two broad classes already mentioned, both working with saturated steam up to pressures of 250 lbs. per sq. in., flash boilered vehicles, usually with single acting poppet valve engines have been employed, but for the transport of goods have not been widely used. These machines are usually of the 3-ton type, which is suitable for omnibus work, and for this purpose have met with some measure of adoption, especially in London.

Undoubtedly these heavy vehicles have suffered in the past from lack of experience on which to base their working, and this has especially been evidenced by the excessive speeds at which they have frequently been driven. It is a mistake to endanger the advantage that mechanical traction undoubtedly possesses over the horse in this way. It is quite possible to make these heavy hard-tyred machines run up to 12 miles an hour, as was shown at the R.A.S.E. trials this summer, but such speeds are not economical. Suggestions have frequently been made that the governors should be encased, so as to be beyond the control of drivers, but men are nowadays so cunning that it is extremely difficult to devise a means of encasing any mechanism so that it cannot be tampered with. This fact was very much emphasised in the earlier days of the motor omnibus, when the inexperience of the drivers and rivalry of the several companies then on the road led to excessive speeds which the companies tried to prevent, with only partial success, by locking up the governors

Such vehicles as those with which we have been dealing invariably run on hard tyres and are therefore confined by law to a speed of five miles an hour. A lighter class, to run on hard tyres at eight miles an hour within the three-ton weight limit, has not met with much success, but fitted with rubber tyres, so as to obtain the full twelve miles an hour allowed by law, the three-tonners are making considerable headway.

With regard to the future of the steam tractor and lorry there does not appear much immediate likelihood of radical change, so long as there are no alterations in the existing law, and, even should such alterations be introduced, it is improbable that they would be of a very sweeping nature all at once, especially now that there are such large numbers of heavy vehicles already in existence.

Internal Combustion Vehicles.

The departure that has caused more stir than any other during the last five years among those interested in the internal combustion engine is undoubtedly the slide valve, but this has not as yet been put on the market on a practical scale in the sphere of the business motor. It will, however, undoubtedly have to be reckoned with in the future, but as the *pros* and *cons* attached thereto have already been discussed at such length in the columns of *The Automobile Engineer* it is unnecessary for the writer to discuss the matter further, for readers are in a position to discount for special requirements and judge for themselves. But it may be said that the sliding sleeve has already been tentatively in use on industrial vehicles in this country and France.

Piston and rotary valves—the latter both of the sleeve and disc types—are also contingencies to be reckoned with, but as yet experience with them in com-

mercial work is almost entirely lacking.

The difference between private car and commercial work is very real. Many people fail to recognise what comparatively easy working conditions the average private car enjoys.

For ordinary engineering work it is necessary that an engine should be capable of working at full power for nine or ten hours a day for years without either materially losing power or requiring excessive repairs. It is nothing at all uncommon for such engines to work regularly every week from Monday morning to Saturday night, with no stop at all, and occasionally engines have worked for more than six months, night and day, without stopping. Further, this sort of work has to be done for years.

While industrial motors do not have to run the long hours which stationary and marine engines do, they have to do work much more like this than like the ordinary pleasure car work, and, therefore, the advantages of the slide valve seem very doubtful. In pleasure car work silence is so important that even if it entails heavier repairs after some years' use, or entails a greater oil consumption, it may be worth the extra cost in running, but makers of utility vehicles can obtain such good results from poppet valves that they are likely to leave experiment to others.

Some phases in industrial working requirements make specially severe demands on certain types. Take the case of motor omnibuses, for instance. In 1906 the London General Omnibus Co. collected certain data of everyday working traffic conditions in the Metropolis, and as a result of their researches it was found that a motor omnibus on a double shift day of eighteen to nineteen hours had to be stopped 690 times, while the brake was applied 2,530 times, and the gears changed 1,035 times. Since then traffic conditions have not grown easier, although the drivers are now certainly far better trained, but the foregoing figures give some idea of the severity of the working conditions.

In general design commercial motors of the internal combustion type have practically followed the lines of the pleasure car with modifications. The early ones were, in fact, simple pleasure car chassis fitted with a van body, and were generally unsatisfactory, as might have been expected. The essential point in a pleasure car is to make it as light as possible consistent with enough durability to stand the work it is usually put to, and this consists of extremely short hours of work, of which only a very small part is at anything like full power. Thus, when it is said that various working parts of a car are wonderfully small for the work they stand people often omit to remark that they don't stand; at all events, in the way that the word is usually understood by engineers. That is to say, that they will not stand continuous full power work for the usual working hours of the day. Commercial vehicles, however, will have to stand this and yet run with a moderate amount of repairs, and hence the development of this type of conveyance has largely been in the direction of modifying the pleasure car into a distinct machine. In fact, as the ordinary commercial vehicle will have to do something like ten times the number of hours' hard work that the pleasure car does in a year it is obvious that there must be an enormously greater margin of wear and safety in all the parts. In the case of the engines it means that lower speeds must be run and that the sizes of the cylinders for a given power must be larger than in the pleasure car in order that there should be margin for the development of the full power for long stretches without excessive wear.

The history of the last two years is very largely the recognition of these points, and, therefore, the elimination of defects. It naturally follows from this that the tendency of the trade is to specialisation, and that, therefore, industrial motors are more and more being made either in works devoted entirely to them, or else in a special department of their own. There is no doubt that this tendency is right and that works which devote themselves to commercial motors only should be able to construct them better and cheaper than those which produce pleasure cars or anything else as well.

Use of Cheap Fuels.

Before concluding we must not omit to mention the attempts that have been made to use cheaper fuels. One of these, in the shape of a single cylindered machine, using heavy oil, with all its details reduced to their simplest elements, promises particularly well, for in this design the builders, rightly considering the lorry as a business proposition, have done everything possible to reduce cost alike in construction and working.

A good deal of experimental work has also been done with producer gas in this country and France, and some of the consumption costs have been remarkably low. He would be a bold man who would foretell exactly what this source of power has in store for the automobile of the future: it seems to have almost boundless possibilities, but at present is still in the experimental stages for transport work.

Lorries, generally speaking, follow the same lines as the omnibus and the same tendencies are evident. Considering the immense advantages in principle which the internal combustion engine has over steam, it is rather curious that the heavy lorry with an internal combustion engine has not made greater headway relatively to steam. It seems probable that this is largely due to the great experience which the traction engine makers started with as to transmission, gear, etc., which enabled them to avoid some of the errors of the makers of petrol lorries. Although the latter have gained a great deal of experience and eliminated many of their weak points, it still seems possible that they might learn something from the traction engine maker. One point of difference in the practice of the two schools is the steering gear, the traction engine maker generally using the central pivot and the petrol lorry makers the Ackerman axle. The former appears to have many advantages for this class of work, being much simpler in construction and allowing of the wheel to be turned much more easily when the vehicle is standing, a matter of some importance in manœuvring in a confined space.

In transmission gears it seems curious that the live axle is preferred both in the lighter vehicles, such as the taxis, and also in the heaviest, such as the traction engine, while the medium weight vehicles often have side chains.

There are signs, however, that the live axle is increasing in popularity. Worm transmission has been used for years in this work, as also the bevel, but with bevel live axle transmission sufficient reduction is not easy without a reducing gear. Moreover, side chains, despite their obvious disadvantages, allow of the axle being cranked and so permit a lower body position.

The Motor Omnibus.

The omnibus has not developed in the last two years to any great extent, in fact, the number of motor omnibuses running in London in September, 1910, was slightly less than in the same month of 1908. The alteration in technical details has naturally not been very marked, though there is now a great improvement in detail, and a general tendency to quietness, but the most outstanding influence in this quarter has been the police regulation issued some two years ago, limiting the weight of the vehicle complete with body to $3\frac{1}{2}$ tons. As in the cab business so in the motor omnibus industry the regulations of the Metropolitan police have been a vexed question (for provincial regulations are largely modelled on those of London). It cannot be denied that in its earlier days the motor omnibus was severely handicapped by the police. The inspectors appointed to look after the licensing of these vehicles had had no engineering training, and consequently the omnibus proprietors suffered from their ignorance, and from an official and absurd tendency towards red tape. The writer has known a 'bus refused a license because the body was $\frac{1}{8}$in. longer than the regulation dimension, and another car stopped because of a broken lamp glass, while every omnibus engineer can doubtless recall many more such instances. It must be admitted, however, that the weight limitation of $3\frac{1}{2}$ tons with double decked body has already exercised a beneficial influence, and has evolved a quieter type of 'bus, that, as time goes on, will probably prove considerably lighter on repairs than its more cumbersome predecessors. The necessity to cut down weight on these lighter types has led to the adoption of propeller shaft and live axle bevel drive, and another interesting feature in the shape of a gear box driven by inverted tooth chains has also been introduced owing to the insistent demands for quieter running—an object that has certainly been obtained to a marked degree in the new models.

Going back to the earlier history of about four years ago probably the most notable features were the adoption of inverted tooth side chain drive to the rear wheels, and experimental work with various lubrication systems with a view to reducing smoke. In fact, it may be said that industrial motor requirements have done more than anything to improve lubrication arrangements in general. On the other hand, cooling arrangements in some cases seem far from adequate, for it is not at all uncommon to see the water boiling furiously. Though the difficulties are admittedly far greater than in the case of the private car, there seems

no reason why this detail should be improved, except perhaps the severe weight restrictions, but even these do not look insurmountable.

There is no doubt that the problem of constructing a really satisfactory motor omnibus is as difficult a one as any there is in motor engineering, for they have to be very light in proportion to the load carried and, at the same time, stand very hard work, while it is extremely desirable in the public interest that they should not be noisy. In the petrol omnibus the latter is the point which appears to require more attention than any other. For this reason it seems doubtful whether the sliding gear adopted by the pleasure car is the best for the purpose of a motor omnibus, judging by the figures we have already given. It is not easy to make hardened steel wheels run really quietly, even when new, and with wear they rapidly get worse. Some system of gears always in mesh, or possibly chains, might be desirable. It may be remembered that about three and a half years ago some motor omnibuses with chain drive throughout and epicyclic transmission were put on the streets of London, and proved remarkably quiet, but financial complications led to their withdrawal.

Both in upkeep and efficiency the gear box has proved one of the most serious problems, and many attempts, as for example, the use of hydraulic or pneumatic transmission systems, have been devised, but none have yet proved themselves commercial successes, probably owing in part to the fact that such mechanisms involve work of very great accuracy, and are consequently expensive to manufacture.

(It is the gear box on the motor omnibus, too, that seems to point to three-point suspension as being most correct in principle, and this also applies to the engine—for the number of four-armed gear boxes and crank cases that had their supporting arms broken was enormous, especially in the early days.)

With a view to doing away with gear box noise and the wear and tear of changing gears the petrol-electric principle seems to offer possibilities, more especially in connection with the heavier classes of passenger-carrying vehicles, but its use has not spread to the extent which was hoped and anticipated by its supporters some four or five years ago. At that time those who acknowledged the possibilities of any such system were divided as to the most promising method of approaching the problem and, broadly speaking, at the present moment we are not much nearer a definite decision.

Early systems, as a rule, involved the use of accumulators as an *essential* portion of the installation, but it has at last been realised that nothing short of an epoch-making development in the manufacture of secondary batteries will suffice to justify their use to this extent. It is, of course, possible to use accumulators, not in such a way that the whole of the power passes through them, but as an auxiliary serving to increase engine power when necessary, and so included as to remain almost fully charged by absorbing a portion of the power when that given out by the engine is in excess of requirements, or when a braking effect is actually beneficial. This system, which in an early form was introduced into this country under the name of the "Gearless," or Auto-Mixte, has been developed and improved by the Daimler Co., and is now embodied in the Daimler K.L.P. petrol-electric omnibus.

Among those who have held that accumulators should find no place on a petrol-electric vehicle, diverse opinions have been offered as to whether the electric generator should give a continuous or an alternating current. The principal advocate of the latter has recently given his attention more particularly to other spheres in which electric transmission appears to offer great possibilities. Meanwhile, certain supporters of the continuous current system have brought their experiments to a close, whilst others have made steady, if rather gradual, headway. Among the latter must be mentioned the one who was responsible for the electrical equipment of the only petrol-electric vehicles which may be said to have given anything approaching permanent service in the London streets.

Another system, which would appear on paper to be liable to very severe criticism, seeing that the electric motors with their commutators are situated in the rear wheels and are therefore unsprung, has survived some systems which, at first sight, appeared very promising, and good reports come in from certain quarters where it is being employed on public-service vehicles.

Generally speaking, the tendency appears to be for the designers of petrol-electric systems to attempt less than they have done in the past, and to endeavour to replace clutch and gear box only, and to drive to a differential back axle. This sacrifices one of the possible benefits of the system, in so far as it does not reduce mechanical gearing to a minimum. At the same time, it assists towards simplicity and reduction of weight.

As matters at present stand, the principal theoretical objections to electric transmission on the grounds of its weight and its inefficiency at varying speeds (both conducing to excessive consumption of fuel) need not be regarded as very serious obstacles in the way of development, with the sole exception that even a slight increase in weight makes it difficult for the designer to conform to the Scotland Yard Regulations for motor omnibuses in London. The point on which the chief doubt still exists is whether the electrical machinery, when its weight is reduced to the extent which is necessary in order to compete with mechanically gear-driven vehicles, can claim that absolute immunity from breakdown, which alone can lead to its extensive employment. A practical difficulty, too, is that operating concerns are chary of employing any system which required additional special knowledge on the part of their drivers or mechanics.

The most recent development is a transmission, which should perhaps hardly be included under this heading, since the drive is only partially through an electric medium. This system is at the present moment being applied to heavy rail motor coaches, as well as to motor vehicles.

The foregoing sufficiently outlines the general constructional tendencies in each of the leading sections of the industrial automobile world, but in considering the results given by such mechanical developments it is quite impossible to say how much is due to improved mechanism, and how much to increased experience of the user. Five years ago but very little real experience in working business automobiles was available as data, and consequently the industry suffered severely, for users—master and man alike—had little knowledge on which to work. Nowadays the accumulated experience of half a decade, besides improvements in materials and fittings, such as ignition and carburation accessories, have told their tale, and while the machines are more intelligently and effectively worked, they are better able to withstand abuse, and in some cases more foolproof. Such is the irony of development.

In nothing are these improved conditions more clearly evidenced than in the depreciation charges. Four years ago it was generally admitted by those best qualified to express an opinion, that, while depreciation was more or less speculative, it was not safe to write off less than 20-25 per cent. for petrol gear-driven machines, and 15-20 per cent. for steam waggons. In the case of the petrol vehicle the depreciation estimate was somewhat influenced by the results hitherto shown by the much abused and long-suffering London motor omnibuses, and quite possibly the view on the steam vehicle was influenced to a lesser extent by the same cause. By the present time, however, experience has taught us that under average conditions an annual depreciation charge of 10-12½ per cent. is quite ample for a steam vehicle, while 12½-17½ per cent. sufficiently covers the petrol machine under normal good conditions.

Although the generalisation of three or four years ago—to the effect that up to three tons useful load petrol is preferable; above that capacity steam—still holds good on the whole, it is less true than formerly. This is so for two reasons. In the first place some makers of steam waggons have taken up the manufacture of conveyances of smaller capacity than their standard five tonners, and these smaller machines are usually being run on rubber tyres. Thus the steam makers threaten to invade the domain of the petrol waggon's supremacy. In the second place, constant improvements in detail have enabled the heavier petrol vehicles to close up the gap in efficiency between themselves and steam. These improvements in detail can be seen reflected in the amounts in the shape of decreased fuel and lubrication items, and fewer replacements, and breakdown and emergency charges.

The Motor Cab.

We have already expressed the opinion that the industrial motor vehicle will tend more and more to specialised design and manufacture, but an exception to this tendency at present is to be found in the taxi cab. This is undoubtedly the vehicle which has developed most in the last four years, for before that they had not been put on the streets on a large scale, although there had been much experimental working. Naturally, therefore, they were simply pleasure car chassis slightly altered to come within the police regulations, and even now there is very little difference in design between

the majority of them and the ordinary pleasure car. The principal alteration required, when the early police regulations were published, was in the steering, which had to give such a lock that the cab could be turned in a circle of 25 ft. diameter, but though such steering locks are now provided on motor cabs, they are very far from perfect in action and offer an interesting field to the inventive engineer.

Another outstanding feature is to be seen in the provision of much larger engines and the replacement of the original two-cylinder type by four-cylinder cabs. The early taxis had two cylinders, 75 by 120 millimetres (3in. by 4¾in.) or thereabouts. The latter vehicles have four cylinders, usually about 80 by 120 millimetres (3⅛in. by 4¾in.), and even in some cases 83 by 127 (3¼in. by 5in.).

This increase of power is due to several causes, one of which is, doubtless, the necessary of providing an ample margin to meet any conditions, for the motor cab with its vastly increased range of action has revolutionised the ideas of habitual cab users, and so is now liable to be called on to go anywhere. But the most potent factor in this development of power probably lies in the fact that the really lucrative customer—the person who uses the cab for long journeys across country—nowadays simply refuses to look at a two cylinder cab, and consequently certain cab owning companies have forced each others hands and embarked on a race in luxury in their anxiety to secure this class of traffic. Doubtless an ample margin of power is most necessary, for if commercial vehicles of any kind are to be a thorough success it is necessary that they should be absolutely certain of their performances. The larger engines also render changes of gear less frequent and therefore save wear in the gear box, and they also allow of the power ordinarily required being obtained at lower revolutions, which again saves wear in the transmission, but it remains yet to be seen how far the limits of expense in this direction can be carried. Comfort and even luxury are most desirable, but in the cab business these properties must show a profit.

As far as London traffic goes engines have probably reached the limit of their dimensions, but in many of the towns in the North it seems probable that still more power will be required, as there is a far greater amount of hill work to be done than in London, and also the surface of the road is often not so good. In the North, also, there should be more power on the low gear, though this, of course, can be obtained by lowering the gearing in the back axle.

The taxi has been in use on a large scale such a short time that it is impossible to foretell what will be the permanent type, or whether the manufacture will remain in the hands of the pleasure car maker or be a special one for makers doing nothing else. There can be no doubt that the use of what is practically a pleasure car chassis for continuous hard work entails a great deal of wear and tear, and that this could largely be avoided by making all the parts, especially the transmission gear, more substantial. On the other hand, in all vehicles there comes a point when the increased durability gained by this entails so much weight that there is no saving in the running cost owing to the increased petrol consumption, and still more the extra wear of the tyres; let alone the limitations of police regulations. Where pneumatic tyres are used the question of weight is especially important, and as these are universal in the taxi it is quite possible that it will pay to put up with a considerable amount of repairs to the transmission, etc., rather than increase the weight. Every endeavour should, of course, be made to increase the size of those parts subject to wear without increasing the weight, but in the tyre question the increasing engine power also demands at least as much consideration as weight.

A YEAR'S DEVELOPMENT OF THE AEROPLANE.

The Progress made in design and construction during 1910.

By. W. G. Aston.

IT may be said, though in some cases with a little injustice to their designers, that all machines are capable of having their genealogy traced back to four main parent types, viz., the Wright and Farman Voisin biplanes, and the Antoinette and Bleriot monoplanes, and it is certainly a singular fact that during the past year, in spite of the large volume of experimental work which has been carried through by the manufacturers concerned, none of these machines, with the exception of the Wright, has undergone any appreciable change either in construction or in design. From this, taken in conjunction with the fact that practically all the prizes which have been competed for at meetings during the past season have been carried off by one or other of these makes, it might well be argued that a certain degree of finality has been achieved, or at least is within sight. There are, however, cogent reasons for this apparent stagnation on the part of the principal constructors, in comparison with whom it cannot for a moment be denied that the smaller aeroplane-producing firms have been making considerable strides towards the perfection of the machine. The deplorable fact of the matter is that in spite of the vast sums which have been, as prize money, won by these firms, and the large number of machines which have been made and sold at a big profit during the last season, the heavy financial losses on experimental work which has been in some cases going forward for years, have not yet been recouped. The only possible step for the manufacturer who has already established a reputation for, and installed a plant for the production of, a particular type of machine is, obviously, to go on making that machine so long as there be a sufficient demand for it. It is necessary to understand this fact in considering the progress which has been made in a year that will in history unquestionably be distinguished by the extraordinary aeronautical achievements which it has witnessed.

I have already suggested that viewed from a certain standpoint the smaller manufacturers who are quite naturally building up their fabrics upon foundations already prepared for them, have opportunities of developing their designs with more freedom than financial exigencies permit the pioneer-constructors to enjoy, but on the other hand the clientele who provide the smaller fry with the sinews of war very largely hamper this development by enforcing its restriction to details. The aeroplane builder is called upon to make a machine sufficiently resembling a well-known and successful prototype to warrant some optimism as to its flying capabilities; it is, therefore, only to small details that he can apply such originality as he is possessed of, and in consequence it is only in details that improvement has been made—to judge from a careful inspection of the 1909 and 1910 Paris Shows, which in this regard may well be taken as a representative criteria. It will be readily seen that in view of this restricted field of development it may well be that designers, as a body, are one and all on the wrong track, if not altogether, at least to a certain extent. Such excellent ends have been achieved, however, that one can scarcely conclude that the means whereby they have been gained are wrong, but there is nevertheless every indication that freed from the deplorable influence of unnecessary "Flying Meetings," the year 1911 will see some very radical changes in aeroplane design.

It is entirely erroneous to suppose that during the past season there has been a struggle for existence between the monoplane and the biplane; there is no reason whatever why the two types should not continue to be co-existent, for each has its own distinct sphere of utility which every day is becoming more clearly defined. Each system has certain constructional and other advantages over the other, those of the monoplane being principally :—

(1) Minimum of head resistance.

(2) High flying speed, involving a certain amount of natural stability and capacity for facing winds of some strength.

(3) Concentration of weight about a central point and consequent reduction in moments of inertia.

(4) Protection of pilot.

Those of the biplane are :—

(1) Lightness of construction due to the girder formation of the planes.

(2) Employment of large supporting surfaces, affording large weight-carrying capacity.

(3) Arrangement of planes permits the use of high aspect ratio without impairing strength of construction.

The monoplane is thus pre-eminently a speed machine and the biplane a weight-carrier; with the exception of the Roe Triplane, no machines using more than two

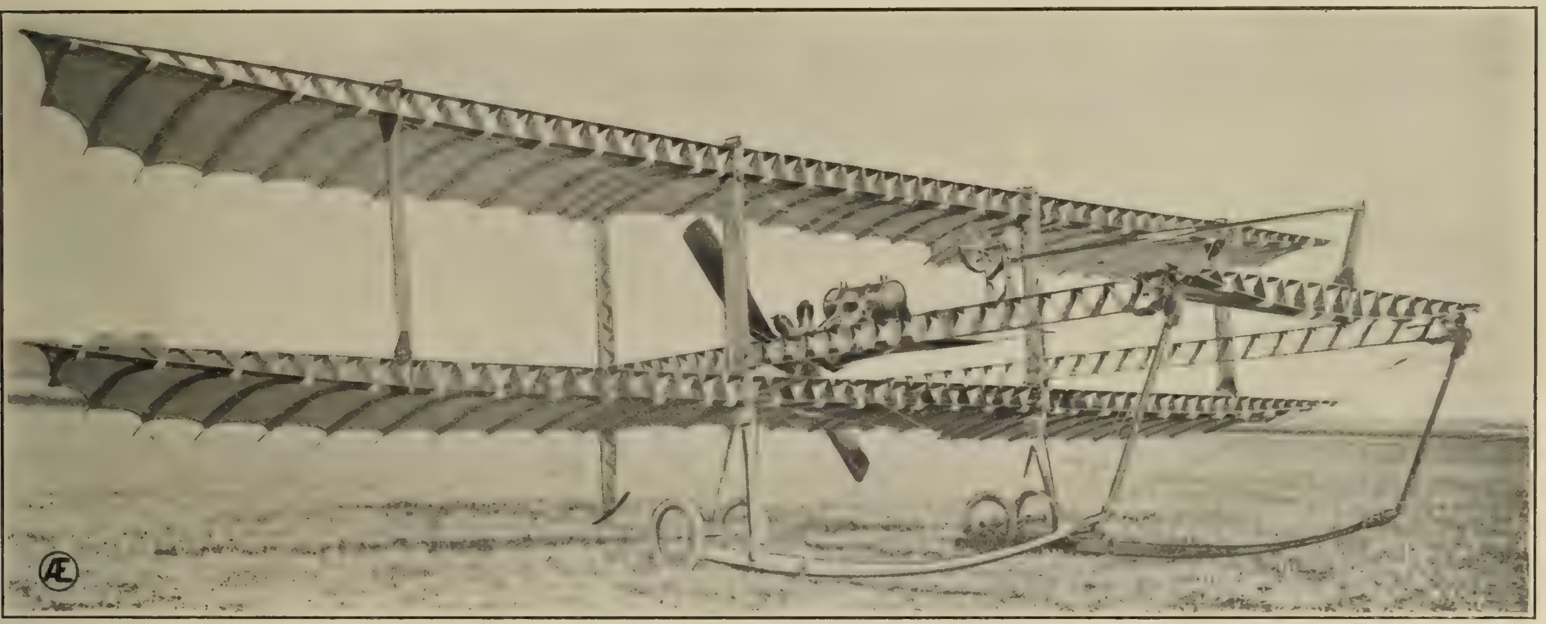
planes have yet appeared, and this fact is no doubt due to the constructional difficulties which present themselves in the production of a multiplane on lines similar to those of the present type of biplane. There is, however, every reason to suppose that in view of its advantages in certain directions a machine having a large number of planes will materialise in the near future.

Apart from this question, 1910 has been undoubtedly a "monoplane year," this type of machine having demonstrated its superiority over the biplane in respect of speed just as effectively as the latter has proved its superiority in point of weight carrying. With regard to distance traversed on a given supply of fuel and ability to rise to a height, there is nothing to choose between the two types; as far as the latter point is concerned, both monoplanes and biplanes have successfully established new altitude and duration records. Here again, however, there can be no true competition between the two, for whilst on the one hand the monoplane can rise to a given height with more rapidity than a biplane, the latter holds an equal advantage in duration work by reason of its greater fuel carrying capacity and of the fact that with a given supply of fuel it is capable of carrying a prescribed weight a greater distance, though at a less speed, than can be done with a monoplane. Already, then, we see the lines along which both systems may be developed, the monoplane as a purely sporting vehicle, the biplane as a passenger-carrying machine, which therefore commends itself to military and possibly commercial purposes.

To return to the parent types already mentioned, we find that such change as has taken place in them has been made with a view to imparting a certain amount of *natural stability* to the machine, and by this term I wish to imply something widely different from what is suggested by the "*automatic stability*." An automatically stable aeroplane is one, I take it, which is capable of maintaining both its lateral and longitudinal balance in the face of all kinds of atmospheric variations, such as side gusts, etc.; but a naturally stable one is merely expected to be capable of tending to right itself when uncontrolled in still air. Although much research has been done in the problem of stability, theoretically it is a question which is still in a contentious state. Broadly speaking, it may be stated that when lines drawn at right angles to the chords of the curves of the supporting surfaces converge to a point or points *above* the machine, the machine will then tend to right itself longitudinally by virtue of its pendulum-like state, whereas if such lines converge to a point *below* the machine, it will conform only to conditions of unstable equilibrium. In this elementary theorem it will be seen that no notice is taken of the relative positions of the centres of pressure and gravity, but it may be said that in no existing machines are these points sufficiently far apart to upset the general truth of the statement. The same reasoning may be applied to the question of natural lateral stability, the use of a dihedral angle between the wings involving exactly similar conditions of stable equilibrium.

The Farman machine may be justly described as a naturally stable one, and this to a very great extent accounts for the success with which it has met. Several Farman machines have been turned out with monoplane tails in place of the biplane type previously used, but beyond a very slight increase of speed, due to the small decrease in head resistance, no advantage is traceable to its use, and in consequence most of the new Farman machines will be exactly similar to the previous type. Another slight alteration which was made in certain machines, which we believe, however, were principally designed for speed purposes, was the cutting away of the lower main plane at each side, so that it was considerably shorter in spread than the upper plane. This plan does not appear to have produced any valuable advantages, and it may possibly have been a positive disadvantage by reason of the fact that with this arrangement the centre of pressure was rendered even more remote from the centre of gravity. In any case, however, the efficiency of the machine can only have been slightly affected, for it was on this type of Farman biplane that the London to Manchester flight was made in the summer of 1910.

As for the Antoinette, no great sign of modification or improvement is discernible in the new type. The machine is undoubtedly one of the best types of monoplane existant, if we are to judge by the performances which have been put up by it, when flying under extremely adverse conditions. It will be understood that in this machine, in which the tail or empennage is formed of a very large flat V-shaped plane, terminating in a pair of quite small movable flaps, quite a large alteration of the angle of these flaps can be made without seriously affecting what we may call the average inclination of the whole tail. Hence it might easily happen that perpendiculars from the planes themselves might easily, if produced, meet below the machine, without, however, giving a true indication of the machine's stability. On the type XI. Bleriot monoplane, it may be said that conditions exist which are quite comparable if not analogous to those which characterise the Wright biplane, but in no case can the stability of the whole machine be so violently nullified, as the central portion of the tail is rigid. In ordinary flying, the Bleriot monoplane might be described as being a machine of neutral stability, as if the engine is pulling well, so that the tail plane is well "up," all the essential conditions of stability are fulfilled. It is not surprising to find, however, that an empennage of the Antoinette type is now being fitted to some of these machines.



The Paulhan Biplane.

It would be idle to deny that some valuable lessons have been learned from the distressing fatalities which have characterised so many of the past year's flying meetings, but in most cases the accidents could have been easily prevented by the exercise of a very small amount of judicious forethought. To take a case in point, at least two fatalities can be traced as being directly due to installing a high powered engine in a construction which was barely capable of withstanding the stresses of a motor of half the power. One would have thought that even the most "rule of thumb" aeroplane designer could not have forgotten the important fact that head resistance is a function of the square of the velocity, yet both Le Blon and Delagrange did not hesitate to employ a 50 h.p. engine on a machine which was by no means any too

strong for a 25 h.p. The mere keeping down of engine power is, however, no longer a safeguard, if the precipitous "vols-planés" in which so much importance is mistakenly placed, are to be indulged in. This action has proved a fruitful cause of accidents, but it requires no great faculty of perception to see that these were inevitable, when a machine designed for a flying speed of not more than fifty miles an hour is allowed to dive earthwards at a velocity probably half as much again, if not more. It is not so much the stress thrown upon the planes during the descent which causes the fatal rupture, for during the descent the head resistance of the machine is at its minimum, but at the termination of the glide the tail plane is called upon to stand the enormous effort of diverting the path of the whole machine from practically a vertical one into a nearly horizontal, and the immense stress thus produced has proved far too heavy for a deplorably large number of machines.

Regarding now the construction of the machine itself, very few improvements appear to have been made. There is hardly any doubt that, both from the dynamic point of view and also that of utility, the boat-shaped fuselage is infinitely superior to the popular Bleriot type of frame, which consists of a simple girder formation. Not only is its head resistance considerably less, and its strength and resistance to shocks immensely greater, but it also provides the pilot with the necessary modicum of comfort and protection, and it is clear that only on account of its cost has it failed to become universal practice.

It must be admitted that the construction of sustaining planes of a biplane and monoplane type is decidedly imperfect at present, and considerable development may therefore be looked for in this respect. The disadvantages of the present system of wing construction are principally that the framework, being practically rigid in all directions, no provision is made for the stretching and contracting of the covering fabric, which is brought about by changes in atmospheric conditions. The Antoinette wings, for instance, are probably three or four times as strong as any others used on a monoplane, yet they were not able, in the case of Wachter's fatality, to resist the load imposed upon them by a sudden change from pouring rain to brilliant sunshine. It is true that the construction used on most biplanes is less susceptible to this kind of strain than that of the monoplane, at all events no biplane accidents are traceable to this cause. The Paulhan biplane, shown at the Paris Show, possesses, however, great advantages in this respect, apart from any other claims which may be made for it. In this case only one wing spar is used, which forms the leading edge of the plane, and from this, solid sticks, forming ribs, are extended backwards, the fabric cover being provided with pockets in which these ribs are housed. The front spar consists of two broad, thin, ash planks, held apart by a series of wooden distance-pieces forming a series of triangular cells which resemble a number of continuous capital W's. Should the fabric contract from any cause, this can only have the effect of throwing an increased compressional stress on the ribs, and a bending stress on the front spar, both of which stresses are from the situation and construction of the members well capable of being withstood. It is interesting to note that the majority of the critics at the Paris Show deemed this machine as being utterly impracticable as a flyer, but it has since demonstrated its flying capability beyond the possibility of dispute. It is not probable that the highly individual form of construction used in it will become universal, but it possesses so many advantages over the construction hitherto favoured, that it cannot fail to have a beneficial influence upon aeroplane design during the next year.

Perhaps the most notable development in design and construction has taken place with regard to the landing chassis of both monoplane and biplanes. At the beginning of the present year these were, for the most part, only suitable for running over the smoothest of flying grounds, and, in many cases, they were not even strong enough for this purpose. Now, however, the aeroplane is rightly looked upon as a vehicle for cross country locomotion, which must be able to land and start from anywhere. Hence, considerable ingenuity has been expended upon making the running gear not only light and of small head resistance, but capable of passing with a minimum of friction and likelihood of breakage over the roughest of ground. In this respect no one has done more than Mr. Cody. He has had two disadvantages to contend with, namely, the great weight of his machine, and its consequent high speed, and, worse still, the exceptional unevenness of the ground over which he flies. Nevertheless, he has been able to build a running chassis which is able to ignore such things as ditches, thick grass, and even trunks of trees, and his example has had a considerable effect upon the design of other machines. Much, however, remains to be done in this department of construction, as it is evident that unless a machine can with certainty and rapidity be made to rise off ground, which can only be selected by a cursory inspection from above, it can have little value either as a sporting or military vehicle, and it is probably to the latter qualification that development in the immediate future will owe its inception.

Another phase to which several manufacturers are now wisely turning their attention is the ability of a machine readily to take to pieces, either for housing or transport. At present the owning and running of an aeroplane imposes a stress on the purse which only a few rich men can sustain, and few men possess such qualities as must accelerate the development and perfection of the machine. If an aeroplane can be made with a sufficiently effective running gear to enable a specially prepared flying ground of great area to be dispensed with, and if it can be made to take to pieces so quickly and easily that a large hangar is no longer required, it must then appeal to a far larger section of the public than it does at present, although it is to be feared that even so, its maintenance must be a very costly matter, small, however, compared to that which it is to-day. Further, efforts must be made to render it possible to use an aeroplane without the necessity of employing a small army of mechanics and repairers, the work of whom, however, chiefly lies in maintaining the engine in order, and in making good damage.

Turning from the aeroplane itself to its engine, it cannot be denied that development of the power plant has been less than revolutionary. Although other engines have put up many notable performances, the engine which has established itself already as the favourite pattern is the rotating Gnome type, which has already been followed by a considerable number of imitators, some good but most of them indifferent. At present the greatest difficulty to contend with in an engine of the rotating cylinder type is (apart from the fact that its efficiency is seriously impaired by its considerable head resistance, both in rotation and forward motion) the question of lubrication, but it must be remembered that it is this very problem which prevented this type of motor from coming into general use years ago. Unfortunately, in an aeroplane, the question of fuel-carrying capacity is of more urgent importance than in any other type of vehicle, and no pains can be too great to be spent on attempts to lessen the immoderate oil consumption of the rotating engine, and it may be said that a solution or even part solution of this problem would not only result in the effective radius of action of the machine being very greatly increased, but would also help in largely reducing the running expenses to enable it to be brought within the reach of a large number of men who at present do not see their way to sinking considerable capital in castor oil, the consumption of which is in the majority of cases not very different from that of petrol.

Finally, 1910 has seen the development of machines suitable only for the giving of acrobatic performances, into types for which a useful if not an immediately commercial future is beginning to dawn. It has seen the production of an aeroplane which, in ordinary weather, can be relied upon to fly, and do so safely under the control of almost any level-headed person. It has seen that even with the imperfect type of engine which is now being used, flights of very considerable duration and distance are possible, and that altogether useless altitudes can be achieved. It has seen, too, that as a means of locomotion the aeroplane can be made wonderfully efficient. What we may expect to find in the coming year is more attention being paid to that most important consideration, economy, and to the scarcely less important ones of comparative safety and its capability of travelling from point to point, irrespective of the locality of specially prepared aerodromes.

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